MODELING, OPTIMIZATION AND DESIGN OF A SOLAR THERMAL ENERGY TRANSPORT SYSTEM FOR HYBRID COOKING APPLICATION

BY

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Dedicated to my parents and sisters...
Abstract

Cooking is an integral part of each and every human being as food is one of the basic necessities for living. Commonly used sources of energy for cooking are firewood, crop residue, cow dung, kerosene, electricity, liquefied petroleum gas (LPG), biogas etc. Half of the world’s population is exposed to indoor air pollution, mainly the result of burning solid fuels for cooking and heating. Wood cut for cooking purpose contributes to the 16 million hectares (above 4% of total area of India) of forest destroyed annually. The World Health Organization (WHO) reports that in 23 countries 10% of deaths are due to just two environmental risk factors: unsafe water, including poor sanitation and hygiene; and indoor air pollution due to solid fuel usage for cooking. In under-developed countries, women have to walk 2kms on average and spend significant amount of time for collecting the firewood for cooking. The cooking energy demand in rural areas of developing countries is largely met with bio-fuels such as fuel wood, charcoal, agricultural residues and dung cakes, whereas LPG or electricity is predominantly used in urban areas.

India has abandon amount of solar energy in most of the regions making it most ideal place for harvesting solar energy. With almost 300 sunny days each year, one can confidently relay on this source of energy. India’s geographical location is in such a way that theoretically it receives $5 \times 10^{15}$ kWh/year of solar energy. Solar cooking is the simplest, safest, environmental friendly and most convenient way to cook. It is a blessing for those who cook using firewood or cow dung, who walk for miles to collect wood, who suffer from indoor air pollution. Hence solar cooking is going to play major role in solving future energy problem.

Solar based cooking has never been a strong contender in the commercial market or even close to being a preferred method of cooking. They have been relegated to demonstration appliances to show case the solar based concepts. In this mode, cooking is no longer a time independent activity that can be performed at any time of day. One is forced to cook only at certain times when there is sufficient insolation. The geography of the cooking activity also shifts away from the kitchen. The kitchen is no longer the hearth of the home as the actual cooking activity shifts to the roof tops or high insolation platforms. This further adds to the inconvenience apart from being unable to cook at night or during cloudy conditions or during most of the winter days. Another issue of significant inconvenience is the general social structure in most families of the developing countries wherein the cooking activity is carried out by the senior ladies of the home.
Abstract

They are generally not athletic enough to be moving to and from the kitchen and the roof top to carry out the cooking exercise. As the solar cookers are enclosed spaces, interactive cooking is not possible let alone having any control on the rate of cooking. These are some of the more significant issues in the social psyche that has abundantly impeded the acceptance of solar thermal based cooking appliances. These issues and problems are in fact the motivating factors for this thesis. Based on these motivating factors, this thesis aims to propose solutions keeping the following points as the major constraints.

- cooking should be performed in the kitchen.
- one should be able to perform the cooking activity independent of the time of day or insolation.
- the cooking activity should be interactive
- the time taken for cooking should be comparable with the conventional methods in vogue.
- there should be a reduction in the use of conventional energy.

Using the constraints and the motivating factors discussed above as the central theme, this thesis proposes a method to transfer solar thermal energy to the kitchen and act as a supplement to the conventional source of energy like the LPG or other sources that are traditionally being used in the households. The method proposed is in fact a hybrid scenario wherein the solar thermal is used to supplement the traditional source. Solar photovoltaic cells are also used to power the electronics and apparatus proposed in this thesis. This thesis addresses in detail the issues in analysis, modeling, designing and fabrication of the proposed hybrid solar cooking topology.

The main goal of the proposed system is to transfer heat from sun to the cooking load that is located in the kitchen. The topology includes an additional feature for storing the energy in a buffer. The heat is first transferred from the solar thermal collector to a heat storage tank (that acts as the buffer) by circulating the heat transfer fluid at a specific flow rate that is controlled by a pump. The stored heat energy that is collected in the buffer is directed into the kitchen by circulating the heat transfer fluid into the heat exchanger, located in the kitchen. This is accomplished by controlling the flow rate using another pump.
Abstract

The solar thermal collector raises the temperature of the thermic fluid. The collector can be of a concentrating type in order to attain high temperatures for cooking. Concentrating collector like linear parabolic collector or parabolic dish collector is used to convert solar energy into heat energy. Absorption of energy from the incident solar insolation is optimized by varying the flow rate of circulating thermic fluid using a pump. This pump is energized from a set of photovoltaic panels (PV cell) which convert solar energy into electrical energy. The energy absorbed from the solar thermal collector is stored in a buffer tank which is thermally insulated. Whenever cooking has to be carried out, the high temperature fluid from the buffer tank is circulated through a heat exchanger that is located in the kitchen. The rate of cooking can be varied by controlling both the flow rate of fluid from the buffer tank to heat exchanger and also by controlling the amount of energy drawn from the auxiliary source. If the available stored energy is not sufficient, the auxiliary source of energy is used for cooking in order to ensure that cooking is independent of time and solar insolation. In the proposed hybrid solar cooking system, the thesis addresses the issues involved in optimization of energy extracted from sun to storage tank and its subsequent transfer from the storage tank to the load.

The flow rate at which maximum energy is extracted from sun depends on many parameters. Solar insolation is one of the predominant parameters that affect the optimum flow rate. Insolation at any location varies with time on a daily basis (diurnal variations) and also with day on a yearly basis (seasonal variation). This implies that the flow rate of the fluid has to be varied appropriately to maximize the energy absorbed from sun.

In the proposed system, flow rate control plays a very significant role in maximizing the energy transfer from the collector to the load. The flow rate of the thermic fluid in the proposed system is very small on the order of $0.02\,\text{kg/s}$. It is very difficult to sense such low flows without disrupting the operating point of the system. Though there are many techniques to measure very low flow rates, they invariably disrupt the system in which flow rate has to be measured. Further, the low flow sensors are far too expensive to be included in the system. A reliable, accurate and inexpensive flow measuring technique has been proposed in this thesis which is non-disruptive and uses a null-deflection technique. The proposed measuring method compensates the pressure drop across the flow meter using a compensating pump. The analysis, modeling, design and fabrication of this novel flow meter are addressed.

The design and implementation of different subsystems that involves the selection and
Abstract
design of solar concentrating collector and tracking are explained. Finally, it is essential
to know the economic viability of the proposed system that is designed and implemented.
To understand the economics, the life cycle cost analysis of the proposed system is pre-
sented in this thesis.

The major contributions of this thesis are:

• **Energy transport:** Major challenge in energy transport is to bring heat energy
obtained from the sun to the kitchen for cooking. Energy transferred from solar
insolation to the cooking load has to be optimized to maximize the overall efficiency.
This can be split into two parts, (a) optimizing efficiency of energy transferred from
the collector to the energy buffer tank, (b) optimizing efficiency of energy transferred
from the buffer tank to the load. The optimization is performed by means of a
maximum power point tracking (MPPT) algorithm for a specific performance index.

• **Modeling of the cooking system:** There are several domains that exist in the
solar cooking system such as electrical domain, thermal domain, and hydraulic
domain. The analysis of power/energy flow across all these domains presents a
challenging task in developing a model of the hybrid cooking system. A bond graph
modeling approach is used for developing the mathematical model of the proposed
hybrid cooking system. The power/energy flow across different domains can be
seamlessly integrated using the bond graph modeling approach. In this approach,
the various physical variables in the multi-domain environment are uniformly de-
dined as generalized power variables such as effort and flow. The fundamental prin-
ciple of conservation of power/energy is used in describing the flow of power/energy
across different domains and thus constructing the dynamic model of the cooking
system. This model is validated through experimentation and simulation.

• **Flow measurement:** A novel method of low fluid mass flow measurement by
compensating the pressure drop across the ends of measuring unit using a com-
pensating pump has been proposed. The pressure drop due to flow is balanced by
feedback control loop. This is a null-deflection type of measurement. As insertion
of such a measuring unit does not affect the functioning of the systems, this is also
a non-disruptive flow measurement method. This allows the measurement of very
low flow rate at very low resolution. Implementation and design of such a unit are
discussed. The system is modeled using bond graph technique and then simulated.
The flow meter is fabricated and the model is experimentally validated.
• **Design Toolbox**: Design of hybrid cooking system involves design of multi domain systems. The design becomes much more complex if the energy source to operate the system is hybrid solar based. The energy budget has to be evaluated considering the worst case conditions for the availability of the solar energy. The design toolbox helps in assessing the user requirement and help designing the cooking system to fulfill the user requirement. A detailed toolbox is proposed to be developed that can be used in designing/selecting sub-systems like collector, concentrator, tracking system, buffer tank, heat exchanger, PV panel, batteries etc. The toolbox can also be used for performing life cycle costing.
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Contents

Abstract v
Acknowledgements xi
List of Figures xvii
List of Tables xxii
Symbols xxiii

1 Solar Cooking: An Overview 1-1
   1.1 Introduction ........................................ 1-1
   1.2 An Indian perspective ................................. 1-1
   1.3 Existing cooking sources ............................. 1-3
   1.4 Solar cooking .......................................... 1-5
       1.4.1 Existing solar cooking methods and their disadvantages 1-6
       1.4.1.1 Box type cooker ................................ 1-6
       1.4.1.2 Panel cooker .................................. 1-7
       1.4.1.3 Parabolic cooker ............................... 1-8
       1.4.1.4 Funnel cooker .................................. 1-8
       1.4.1.5 Scheffler cooker ............................... 1-9
       1.4.1.6 Other Types ................................... 1-9
   1.5 Modeling techniques for solar systems .............. 1-10
   1.6 Scope of the thesis .................................. 1-11

2 Maximum power point tracking 2-1
   2.1 Introduction .......................................... 2-1
   2.2 Solar Cooking System ................................ 2-3
       2.2.1 System Description ................................ 2-3
       2.2.2 System operation .................................. 2-4
   2.3 Energy Optimization .................................. 2-4
       2.3.1 Solar Heating .................................... 2-4
       2.3.2 Thermal energy storage ............................ 2-8
       2.3.3 Maximum Power Point Tracking (MPPT) .......... 2-13
<table>
<thead>
<tr>
<th>Contents</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.3.4 Load-side optimization</td>
<td>2-18</td>
</tr>
<tr>
<td>2.3.5 Selection of pipe diameter</td>
<td>2-19</td>
</tr>
<tr>
<td>2.3.5.1 Pressure drop</td>
<td>2-20</td>
</tr>
<tr>
<td>2.3.5.2 Heat loss</td>
<td>2-22</td>
</tr>
<tr>
<td>2.4 Experimental Results</td>
<td>2-23</td>
</tr>
<tr>
<td>2.5 Conclusion</td>
<td>2-25</td>
</tr>
<tr>
<td>3 System model</td>
<td>3-1</td>
</tr>
<tr>
<td>3.1 Introduction</td>
<td>3-1</td>
</tr>
<tr>
<td>3.2 Solar Cooking System</td>
<td>3-3</td>
</tr>
<tr>
<td>3.3 Modeling and Simulation</td>
<td>3-4</td>
</tr>
<tr>
<td>3.3.1 Thermal Model</td>
<td>3-8</td>
</tr>
<tr>
<td>3.3.2 Thermal Capacitances</td>
<td>3-12</td>
</tr>
<tr>
<td>3.3.3 Hydraulic Domain Model</td>
<td>3-15</td>
</tr>
<tr>
<td>3.3.4 State Equations</td>
<td>3-17</td>
</tr>
<tr>
<td>3.4 Parameter Estimation</td>
<td>3-20</td>
</tr>
<tr>
<td>3.5 Results</td>
<td>3-23</td>
</tr>
<tr>
<td>3.6 Conclusion</td>
<td>3-27</td>
</tr>
<tr>
<td>4 Flow measurement</td>
<td>4-1</td>
</tr>
<tr>
<td>4.1 Introduction</td>
<td>4-1</td>
</tr>
<tr>
<td>4.2 The Principle of flow meter</td>
<td>4-3</td>
</tr>
<tr>
<td>4.3 Modeling</td>
<td>4-6</td>
</tr>
<tr>
<td>4.4 Design</td>
<td>4-12</td>
</tr>
<tr>
<td>4.5 Implementation</td>
<td>4-12</td>
</tr>
<tr>
<td>4.6 Effect of pressure transducer performance limits</td>
<td>4-15</td>
</tr>
<tr>
<td>4.7 Linearity</td>
<td>4-15</td>
</tr>
<tr>
<td>4.8 Experimental Results</td>
<td>4-17</td>
</tr>
<tr>
<td>4.8.1 Comparison</td>
<td>4-19</td>
</tr>
<tr>
<td>4.9 Integration with cooking system</td>
<td>4-20</td>
</tr>
<tr>
<td>4.10 Conclusion</td>
<td>4-21</td>
</tr>
<tr>
<td>5 Design of heat transport system</td>
<td>5-1</td>
</tr>
<tr>
<td>5.1 Introduction</td>
<td>5-1</td>
</tr>
<tr>
<td>5.2 Specifications</td>
<td>5-2</td>
</tr>
<tr>
<td>5.3 Estimation of Solar collector size</td>
<td>5-3</td>
</tr>
<tr>
<td>5.3.1 Estimation of solar insolation</td>
<td>5-3</td>
</tr>
<tr>
<td>5.3.2 Sizing of solar collector</td>
<td>5-5</td>
</tr>
<tr>
<td>5.3.2.1 Solar thermal collector</td>
<td>5-5</td>
</tr>
<tr>
<td>5.3.2.2 Sizing of PV panels</td>
<td>5-8</td>
</tr>
<tr>
<td>5.4 Sizing of the storage tank</td>
<td>5-11</td>
</tr>
<tr>
<td>5.5 Sizing of battery</td>
<td>5-13</td>
</tr>
<tr>
<td>5.6 Selection and sizing of pumps</td>
<td>5-15</td>
</tr>
<tr>
<td>5.6.1 Sizing of Pumps</td>
<td>5-15</td>
</tr>
<tr>
<td>5.6.2 Positive displacement pump</td>
<td>5-16</td>
</tr>
</tbody>
</table>
### Contents

5.6.2.1 Rotary pump ........................................... 5-16
5.6.2.2 Reciprocating pump ................................. 5-17
5.6.3 Dynamic pump ........................................... 5-17
5.6.3.1 Centrifugal pump .................................. 5-18
5.6.3.2 Axial flow pump: ................................. 5-18
5.6.4 Internal gear rotary pump ......................... 5-18
5.7 Heat exchanger ........................................... 5-20
5.8 Monitoring and control system .................... 5-21
5.8.1 Temperature measurement .......................... 5-21
5.8.1.1 Different types of sensors ..................... 5-22
5.8.2 Flow measurement .................................. 5-23
5.8.3 Control system ...................................... 5-23
5.8.3.1 Tracking of Solar Collector .................. 5-23
5.8.3.2 Power Supply for pumps ...................... 5-26
5.9 Experimental Results ................................. 5-28
5.9.1 Experimental setup .................................. 5-28
5.9.2 Characteristic of Storage tank .................... 5-29
5.9.3 Characteristic of pumps ............................ 5-30
5.9.4 Performance with constant input power .......... 5-31
5.9.5 Performance with daily insolation variation .... 5-36
5.10 Conclusion .............................................. 5-42

6 Life Cycle Costing ..................................... 6-1
6.1 Introduction ............................................ 6-1
6.2 LCC Parameters ........................................ 6-2
6.2.1 Capital costs ....................................... 6-3
6.2.2 Replacement costs .................................. 6-5
6.2.3 Maintenance and running costs ................... 6-5
6.2.4 Salvage .............................................. 6-6
6.2.5 Carbon credit ....................................... 6-6
6.2.6 Payback period ..................................... 6-7
6.3 Life Cycle Cost Analysis ......................... 6-7
6.4 Bill of materials ....................................... 6-10
6.5 Design Toolbox ......................................... 6-11
6.5.1 Module 1 ............................................. 6-12
6.5.2 Module 2 ............................................. 6-14
6.5.3 Module 3 ............................................. 6-15
6.5.4 Module 4 ............................................. 6-15
6.5.5 Module 5 ............................................. 6-16
6.5.6 Module 6 ............................................. 6-17
6.5.7 Module 7 ............................................. 6-18
6.5.8 Module 8 ............................................. 6-19
6.6 Toolbox structure ..................................... 6-22
6.7 Sample design .......................................... 6-23
# List of Figures

1.1 Block diagram of solar cooking system ........................................ 1-14

2.1 Block diagram of solar cooking system ........................................ 2-3
2.2 Energy flow in solar collector .................................................... 2-5
2.3 Variation in useful energy gain with flow rate ................................ 2-7
2.4 Effect of inlet temperature on collector efficiency .......................... 2-8
2.5 Solar collector with storage ......................................................... 2-9
2.6 Pumping power ............................................................................ 2-11
2.7 Effective Collector power .............................................................. 2-12
2.8 Effective collector efficiency for different inlet temperatures .......... 2-13
2.9 Effective Collector efficiency for different pump constants ............... 2-13
2.10 Effective Collector efficiency for different insulations ..................... 2-14
2.11 Control block diagram for MPPT .................................................. 2-15
2.12 Flowchart for MPPT algorithm ...................................................... 2-17
2.13 Block diagram for energy transfer from storage to load .................. 2-18
2.14 Effect of diameter on the performance .......................................... 2-21
2.15 Variation in optimal diameter with fluid temperature ..................... 2-21
2.16 Block diagram of the experimental setup ....................................... 2-24
2.17 Experimental result showing MPP .................................................. 2-25

3.1 Block diagram of the system .......................................................... 3-4
3.2 Energy flow diagram of the system ................................................ 3-4
3.3 Block diagram showing bond graph elements .................................. 3-5
3.4 Thermal domain Bond graph model ................................................. 3-6
3.5 Hydraulic domain Bond graph model .............................................. 3-7
3.6 Model of the capacitor element \( C_{coll} \) ........................................ 3-13
3.7 Model of the capacitor element \( C_{c1} \) ............................................ 3-14
3.8 Accurate model of the capacitor element \( C_{c1} \) ............................... 3-14
3.9 Block diagram of experimental lab prototype ................................... 3-20
3.10 Block diagram of parameter estimator .......................................... 3-22
3.11 Simulink model of the system ...................................................... 3-25
3.12 Comparison of simulation result with experimental result ............... 3-26
3.13 Variation in effective collector efficiency with flow rate ................ 3-26
3.14 Variation in overall efficiency with flow rate ................................ 3-27
3.15 Variation in temperature ............................................................ 3-27
<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1</td>
<td>Block diagram of proposed flow meter</td>
<td>4-5</td>
</tr>
<tr>
<td>4.2</td>
<td>Experimental flow measurement system</td>
<td>4-6</td>
</tr>
<tr>
<td>4.3</td>
<td>Bond graph model of the system</td>
<td>4-7</td>
</tr>
<tr>
<td>4.4</td>
<td>Control Block of the system</td>
<td>4-9</td>
</tr>
<tr>
<td>4.5</td>
<td>Simulink model of the system</td>
<td>4-10</td>
</tr>
<tr>
<td>4.6</td>
<td>Simulation results: $P_1 - P_2$ Vs Time and flow rate Vs Time</td>
<td>4-10</td>
</tr>
<tr>
<td>4.7</td>
<td>Simulation result for Compensating pressure</td>
<td>4-11</td>
</tr>
<tr>
<td>4.8</td>
<td>Simulation result for output voltage</td>
<td>4-11</td>
</tr>
<tr>
<td>4.9</td>
<td>Engineering drawing of flow measurement system</td>
<td>4-13</td>
</tr>
<tr>
<td>4.10</td>
<td>Circuit diagram of the controller</td>
<td>4-14</td>
</tr>
<tr>
<td>4.11</td>
<td>Block diagram of the experimental setup</td>
<td>4-14</td>
</tr>
<tr>
<td>4.12</td>
<td>Experimental result for voltage Vs flow rate</td>
<td>4-18</td>
</tr>
<tr>
<td>4.13</td>
<td>Transient response of the controller</td>
<td>4-19</td>
</tr>
<tr>
<td>4.14</td>
<td>Experimental setup with flow meter</td>
<td>4-21</td>
</tr>
<tr>
<td>5.1</td>
<td>Block diagram of the experimental setup</td>
<td>5-2</td>
</tr>
<tr>
<td>5.2</td>
<td>Linear parabolic collector</td>
<td>5-6</td>
</tr>
<tr>
<td>5.3</td>
<td>Paraboloid collector</td>
<td>5-7</td>
</tr>
<tr>
<td>5.4</td>
<td>Internal gear rotary pump</td>
<td>5-19</td>
</tr>
<tr>
<td>5.5</td>
<td>Circuit diagram of thermocouple amplifier board</td>
<td>5-23</td>
</tr>
<tr>
<td>5.6</td>
<td>Control block of the tracking system</td>
<td>5-24</td>
</tr>
<tr>
<td>5.7</td>
<td>Pulses and switching states</td>
<td>5-25</td>
</tr>
<tr>
<td>5.8</td>
<td>Circuit diagram of the power converter</td>
<td>5-26</td>
</tr>
<tr>
<td>5.9</td>
<td>Control block diagram for MPPT</td>
<td>5-27</td>
</tr>
<tr>
<td>5.10</td>
<td>Chopper drive circuit</td>
<td>5-27</td>
</tr>
<tr>
<td>5.11</td>
<td>Circuit diagram of the power supply for pump</td>
<td>5-28</td>
</tr>
<tr>
<td>5.12</td>
<td>Decay of storage temperature</td>
<td>5-30</td>
</tr>
<tr>
<td>5.13</td>
<td>Pumping power of pump-1</td>
<td>5-31</td>
</tr>
<tr>
<td>5.14</td>
<td>Pumping power of pump-2</td>
<td>5-31</td>
</tr>
<tr>
<td>5.15</td>
<td>Collector power and pumping power</td>
<td>5-32</td>
</tr>
<tr>
<td>5.16</td>
<td>Raise in storage tank temperature</td>
<td>5-33</td>
</tr>
<tr>
<td>5.17</td>
<td>Load power and pumping power</td>
<td>5-34</td>
</tr>
<tr>
<td>5.18</td>
<td>Variation in storage tank temperature during loading</td>
<td>5-34</td>
</tr>
<tr>
<td>5.19</td>
<td>Load power and pumping power</td>
<td>5-35</td>
</tr>
<tr>
<td>5.20</td>
<td>Variation in storage tank temperature during loading</td>
<td>5-35</td>
</tr>
<tr>
<td>5.21</td>
<td>Input power, collector power and pumping power</td>
<td>5-36</td>
</tr>
<tr>
<td>5.22</td>
<td>Variation in storage tank temperature during heating</td>
<td>5-37</td>
</tr>
<tr>
<td>5.23</td>
<td>Input power, collector power and pumping power</td>
<td>5-38</td>
</tr>
<tr>
<td>5.24</td>
<td>Variation in storage tank temperature during heating</td>
<td>5-39</td>
</tr>
<tr>
<td>5.25</td>
<td>Collector efficiency during heating</td>
<td>5-40</td>
</tr>
<tr>
<td>5.26</td>
<td>Variation in power</td>
<td>5-41</td>
</tr>
<tr>
<td>5.27</td>
<td>Variation in storage tank temperature during heating and loading</td>
<td>5-42</td>
</tr>
<tr>
<td>6.1</td>
<td>Amount of saving over the life span</td>
<td>6-7</td>
</tr>
<tr>
<td>6.2</td>
<td>Cash Flow diagram</td>
<td>6-8</td>
</tr>
</tbody>
</table>
List of Figures

6.3 Cost of unit energy ..................................................... 6-10

E.1 Photograph of the lab prototype ........................................ E-1
E.2 Photograph of the low flow measuring system ......................... E-2
E.3 Photograph of the setup for calibration of the flow meter ............. E-3
E.4 Photograph of the paraboloid collector .................................. E-4
E.5 Photograph of the experimental setup .................................... E-4
E.6 Photograph of the heat exchanger in kitchen ............................ E-5
E.7 Thermocouple, used for temperature measurement ...................... E-5
E.8 Photograph of the circulating pump ...................................... E-5
List of Tables

1.1 Cooking sources used in India ........................................ 1-2
2.1 Algorithm for MPPT .................................................. 2-16
3.1 Effort and Flow in different domains .............................. 3-5
3.2 Experimental results .................................................. 3-23
3.3 Simulation and Experimental results .............................. 3-24
3.4 Error percentage between simulation and experimental results 3-24
4.1 Comparison of existing flow meters ............................... 4-20
5.1 Different types of PV panels ....................................... 5-11
5.2 Different possible switching states ............................... 5-26
C.1 Temperature of storage tank ...................................... C-1
C.2 Characteristic of pump-1 .......................................... C-2
C.3 Characteristic of pump-2 .......................................... C-2
C.4 With constant input power ........................................ C-3
C.5 Both side circulation with constant power .................... C-4
C.6 Only load side circulation ........................................ C-5
C.7 With variable input power ........................................ C-6
C.8 Both side circulation with variable input power ............. C-7
C.9 With constant flow rate .......................................... C-8
C.10 Calibration of flow meter-1 ..................................... C-8
C.11 Calibration of flow meter-2 ..................................... C-9
## Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Name</th>
<th>Unit</th>
</tr>
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<tbody>
<tr>
<td>$A$</td>
<td>Surface area of the heat exchanger</td>
<td>$m^2$</td>
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<tr>
<td>$A_a$</td>
<td>Aperture area</td>
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<tr>
<td>$A_{coll}$</td>
<td>Area of collector</td>
<td>$m^2$</td>
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<td>$A_i$</td>
<td>Inner surface area of pipe</td>
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<td>$ALCC$</td>
<td>Annual life cycle cost</td>
<td>Rs</td>
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<td>Annual maintenance cost</td>
<td>Rs</td>
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<td>Cross sectional area of pipe</td>
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<td>Area of PV panel</td>
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<td>Receiver area</td>
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<td>Capacity of battery</td>
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<tr>
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<td>Thermal capacitance of heat storage tank</td>
<td>$J/K$</td>
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<tr>
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<td>Annual carbon credit</td>
<td>Rs</td>
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<tr>
<td>$C_{c1}$</td>
<td>Thermal capacitance of fluid outlet from heat storage tank</td>
<td>$J/K$</td>
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<tr>
<td>$C_{c2}$</td>
<td>Thermal capacitance of fluid outlet from heat exchanger</td>
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<td>Thermal capacitance of collector</td>
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<td>$C_{Load}$</td>
<td>Thermal capacitance of load</td>
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<td>$cost$</td>
<td>Cost of unit energy from solar</td>
<td>Rs/kWh</td>
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<td>$C_p$</td>
<td>Specific heat of the fluid</td>
<td>$J/kgK$</td>
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<td>Cost of the concentrating collector</td>
<td>Rs</td>
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<td>Cost of frame and mounting structure for collector</td>
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<tr>
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<td>Cost of PV panel structure, mounting and wiring</td>
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</tr>
<tr>
<td>$CP5$</td>
<td>Total cost of storage tank</td>
<td>Rs</td>
</tr>
<tr>
<td>$CP6$</td>
<td>Cost of heat transfer fluid</td>
<td>Rs</td>
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<td>Description</td>
<td>Unit</td>
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<tr>
<td>--------</td>
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<td>------</td>
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<tr>
<td>CP7</td>
<td>Cost of linear actuator</td>
<td>Rs</td>
</tr>
<tr>
<td>CP8</td>
<td>Cost of pump-I</td>
<td>Rs</td>
</tr>
<tr>
<td>CP9</td>
<td>Cost of pump-II</td>
<td>Rs</td>
</tr>
<tr>
<td>CP10</td>
<td>Cost of temperature sensor and conditioning circuit</td>
<td>Rs</td>
</tr>
<tr>
<td>CP11</td>
<td>Cost of heat exchanger</td>
<td>Rs</td>
</tr>
<tr>
<td>CP12</td>
<td>Cost of battery and charging circuit</td>
<td>Rs</td>
</tr>
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<td>CP13</td>
<td>Cost of flow meters</td>
<td>Rs</td>
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<tr>
<td>CP14</td>
<td>Cost of piping and insulation</td>
<td>Rs</td>
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<tr>
<td>CP15</td>
<td>Cost of electronics for control and monitoring purpose</td>
<td>Rs</td>
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<td>C_pf</td>
<td>Calorific value of the fuel</td>
<td>J/kgK</td>
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<tr>
<td>C_pL</td>
<td>Specific heat of load</td>
<td>J/kgK</td>
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<tr>
<td>CR</td>
<td>Concentration Ratio</td>
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</tr>
<tr>
<td>C_v</td>
<td>Volumetric heat capacity of copper</td>
<td>J/m³K</td>
</tr>
<tr>
<td>d</td>
<td>Diameter of pipe</td>
<td>m</td>
</tr>
<tr>
<td>D</td>
<td>Diameter of pipe</td>
<td>m</td>
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<td>Diameter of absorber area of paraboloid</td>
<td>m</td>
</tr>
<tr>
<td>D_c</td>
<td>Diameter of aperture area of paraboloid</td>
<td>m</td>
</tr>
<tr>
<td>D_i</td>
<td>Inner diameter</td>
<td>m</td>
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<td>D_o</td>
<td>Outer diameter</td>
<td>m</td>
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<td>DOD</td>
<td>Depth of discharge</td>
<td></td>
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<tr>
<td>D_no–sun</td>
<td>Number of no-sunny days</td>
<td>day</td>
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<td>Annual cost of conventional energy</td>
<td>Rs</td>
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<td>Effort in Bond n</td>
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<td>E_night</td>
<td>Amount of energy required for cooking during night</td>
<td>kWh/day</td>
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<td>Amount of energy required for cooking</td>
<td>kWh/day</td>
</tr>
<tr>
<td>E_storage</td>
<td>Amount of energy to be stored</td>
<td>kWh</td>
</tr>
<tr>
<td>f</td>
<td>Rate of inflation</td>
<td></td>
</tr>
<tr>
<td>f(n)</td>
<td>Flow in Bond n</td>
<td></td>
</tr>
<tr>
<td>F</td>
<td>Maximum force on linear actuator</td>
<td>N</td>
</tr>
<tr>
<td>F'</td>
<td>Collector efficiency factor</td>
<td></td>
</tr>
<tr>
<td>F_R</td>
<td>Heat removal factor</td>
<td></td>
</tr>
<tr>
<td>g</td>
<td>Earth’s gravitational constant</td>
<td>m/s²</td>
</tr>
<tr>
<td>G_b</td>
<td>Beam or Direct irradiation</td>
<td>W/m²</td>
</tr>
<tr>
<td>H</td>
<td>Pressure head</td>
<td>m</td>
</tr>
<tr>
<td>h₁</td>
<td>Time required to move collector from one end to another</td>
<td>hour</td>
</tr>
<tr>
<td>h₂</td>
<td>Time duration for which pump-I is operated in a day</td>
<td>hour</td>
</tr>
</tbody>
</table>
Symbols

\( h_3 \)  
Time duration for which pump-II is operated in a day \( \text{hour} \)

\( h_4 \)  
Time duration for which monitoring instruments are operated in a day \( \text{hour} \)

\( H_c \)  
Height difference between inlet and outlet of tank \( m \)

\( H_{c1} \)  
Height difference between inlet and outlet of \( C_1 \) \( m \)

\( H_{c2} \)  
Height difference between inlet and outlet of \( C_2 \) \( m \)

\( H_{coll} \)  
Height difference between inlet and outlet of collector \( m \)

\( H_{d1} \)  
Delivery head for pump-I \( m \)

\( H_{d2} \)  
Delivery head for pump-II \( m \)

\( H_f \)  
Friction head \( m \)

\( h_{fi} \)  
Heat transfer coefficient of absorber tube \( W/m^2 \degree C \)

\( \Pi_o \)  
Monthly averages of daily extra terrestrial global solar radiation \( Wh/m^2\text{day} \)

\( H_{tc} \)  
Solar insolation at earth's surface \( Wh/m^2\text{day} \)

\( i \)  
Rate of interest

\( i_a \)  
Armature current \( A \)

\( I_{aux} \)  
Equivalent input current from auxiliary source \( A \)

\( I_b \)  
Beam radiation \( W/m^2 \)

\( I_d \)  
Diffused radiation \( W/m^2 \)

\( I_{in} \)  
Equivalent input current from solar energy \( A \)

\( I_{max} \)  
Maximum current drawn from battery \( A \)

\( I_o \)  
Extra-terrestrial beam irradiance \( W/m^2 \)

\( I_{pump} \)  
Electric current drawn by pump \( A \)

\( I_{sc} \)  
Solar constant \( W/m^2 \)

\( k \)  
Thermal conductivity \( W/m \degree C \)

\( k_1 \)  
Thermal conductivity of pipe \( W/m \degree C \)

\( k_2 \)  
Thermal conductivity of thermal insulation \( W/m \degree C \)

\( K_1 \)  
Fraction of daily load required for night cooking

\( K_2 \)  
Fraction of daily load required during no-sunny days

\( K_e \)  
Motor back-emf constant

\( K_i \)  
Integral constant of controller

\( k_p \)  
Absolute roughness of pipe material \( m \)

\( K_p \)  
Proportional constant of controller

\( k_{pump} \)  
Pump constant

\( K_T \)  
Clearness index

\( l \)  
Length of pipe \( m \)
Symbols

\( L \)  
Length of the collector  
\( La \)  
Armature inductance  
\( LCC \)  
Life cycle cost  
\( Lh1 \)  
Hydraulic kinetic energy storage element on collector side  
\( Lh2 \)  
Hydraulic kinetic energy storage element on load side  
\( \dot{m} \)  
Mass flow rate  
\( ME \)  
Annual maintenance cost  
\( m_{LA} \)  
Mass of collector  
\( m_{storage} \)  
Mass of fluid in storage tank  
\( N \)  
Day of the year  
\( N1 \)  
Replacement period  
\( N2 \)  
Life cycle period  
\( P1 - P7 \)  
Hydraulic pressure  
\( P2 \)  
Hydraulic pressure head by \( \text{pump}1 \)  
\( P6 \)  
Hydraulic pressure head by \( \text{pump}2 \)  
\( PB \)  
Payback period  
\( P_c \)  
Compensating pressure  
\( P_{c1} \)  
Hydraulic pressure across \( C_{c1} \)  
\( P_{coll} \)  
Hydraulic pressure across collector  
\( P_{coll} \)  
Collected power  
\( P_{fl} \)  
Power flow from fluid to load  
\( P_{hx} \)  
Power input to heat exchanger  
\( P_{in} \)  
Power available at input of the collector  
\( P_L \)  
Power loss to ambient from receiver  
\( P_{LA} \)  
Power rating of linear actuator  
\( P_{loss} \)  
Power loss to ambient from pipe  
\( P_{mc} \)  
Power required monitoring and control circuits  
\( P_{min} \)  
Minimum pressure that can be measured using sensor  
\( P(n) \)  
Power in Bond \( n \)  
\( p(n) \)  
Pressure in Bond \( n \)  
\( P_{p1} \)  
Power drawn by pump-I  
\( P_{p2} \)  
Power drawn by pump-II  
\( \text{Power}_{pump1} \)  
Power required for pump-I  
\( \text{Power}_{pump2} \)  
Power required for pump-II  
\( P_{pipe} \)  
Hydraulic pressure drop across pipe

\( m \)  
\( H \)  
\( Rs \)  
\( Ns^2/m^5 \)  
\( kg/s \)  
\( Rs \)  
\( kg \)  
\( kg \)  
\( years \)  
\( years \)  
\( N/m^2 \)  
\( N/m^2 \)  
\( N/m^2 \)  
\( years \)  
\( N/m^2 \)  
\( N/m^2 \)  
\( W \)  
\( W \)  
\( W \)  
\( W \)  
\( W \)  
\( W \)  
\( W \)  
\( W \)  
\( W \)  
\( N/m^2 \)  
\( N/m^2 \)  
\( W \)  
\( W \)  
\( W \)  
\( W \)  
\( N/m^2 \)
### Symbols

- $P_{\text{pipe1}}$: Hydraulic pressure drop in collector side pipe $N/m^2$
- $P_{\text{pipe2}}$: Hydraulic pressure drop in load side pipe $N/m^2$
- $P_{\text{pump}}$: Power required by pump $W$
- $P_{\text{pump1}}$: Power required by pump-I $W$
- $P_{\text{pump2}}$: Power required by pump-II $W$
- $P_{\text{pv}}$: Peak power requirement of PV $W$
- $P_o$: Power required to circulate fluid $W$
- $P_r$: Power received at the receiver $W$
- $P_{R_2}$: Pressure drop across flow resistance $R_2$ $N/m^2$
- $P_{\text{s0}}$: Power taken out from storage tank $W$
- $P_u$: Useful power $W$
- $P_u'$: Useful power per unit length $W/m$
- $PW1$: Present worth of capital cost $Rs$
- $PW2 - PW6$: Present worth of replacement cost $Rs$
- $PW7$: Present worth of maintenance cost $Rs$
- $PW8$: Present worth of salvage $Rs$
- $PW9$: Present worth of carbon credit $Rs$
- $Q_1$: Maximum flow rate on collector side $m^3/s$
- $Q_2$: Maximum flow rate on load side $m^3/s$
- $\dot{q}$: Flow rate $m^3/s$
- $\dot{q}_1$: Flow rate on collector side $m^3/s$
- $\dot{q}_2$: Flow rate on load side $m^3/s$
- $Q_{\text{battery}}$: Amount of energy to be stored in battery $kWh$
- $Q_{c1}$: Volume of the fluid outlet from collector $m^3$
- $Q_{c2}$: Volume of the fluid outlet from heat exchanger $m^3$
- $Q_c$: Volume of the heat storage tank $m^3$
- $Q_{\text{coll}}$: Volume of the collector $m^3$
- $Q_L$: Volume of load $m^3$
- $\dot{q}_{\text{min}}$: Minimum flow rate $m^3/s$
- $Q_v$: Volume flow rate $m^3/s$
- $R1 - R5$: Replacement cost $Rs$
- $R_1$: Resistance of electrical heater $\Omega$
- $R_2$: Thermal resistance between collector and ambient $K/W$
- $R_3$: Heat transfer coefficient between collector and $C_{c1}$ $K/W$
- $R_3^*$: Thermal coefficient between collector and $C_{c1}$ $K/W$
- due to mass transfer
Symbols

- $R_4$: Thermal resistance between $C_{c1}$ and ambient $K/W$
- $R_5$: Thermal resistance between storage tank and ambient $K/W$
- $R_6$: Thermal resistance between $C_{c2}$ and ambient $K/W$
- $R_7$: Heat transfer coefficient between $C_{c2}$ and load $K/W$
- $R_8$: Thermal resistance between load and ambient $K/W$
- $R_9$: Equivalent Resistance of auxiliary source $\Omega$
- $R_a$: Heat transfer coefficient between $C_{c1}$ and storage tank $K/W$
- $R_b$: Heat transfer coefficient between storage tank and $C_{c2}$ $K/W$
- $R_c$: Tilt factor for beam radiation
- $R_{ce}$: Equivalent hydraulic resistance across heat storage tank $kg/m^4s$
- $R_{c2}$: Equivalent hydraulic resistance across $C_{c2}$ $kg/m^4s$
- $R_d$: Tilt factor for diffused radiation $K/W$
- $Re$: Reynold number
- $R_{ex}$: Equivalent hydraulic resistance across heat exchanger $kg/m^4s$
- $R_h$: Hydraulic flow resistance $kg/m^4s$
- $r_{LA}$: Radius of the collector $m$
- $Roil$: Cost of heat transfer fluid per unit volume $Rs/m^3$
- $R_{pipe}$: Hydraulic flow resistance by pipe $kg/m^4s$
- $RT2$: Cost of collector material for unit area $Rs/m^2$
- $RT3$: Cost of PV panel per peak W $Rs/W$
- $RT4$: Cost of conventional energy $Rs/kWh$
- $S$: Salvage value $Rs$
- $S_a$: Surface area of storage tank $m^2$
- $SA$: Surface area of collector $m^2$
- $\dot{S}(n)$: Entropy rate in Bond n $J/Ks$
- $ST$: Cost of storage tank material per unit area $Rs/m^2$
- $ST_{_insu}$: Cost of insulation per unit surface area $Rs/m^2$
- $ST_{_SA}$: Surface area of storage tank $m^{-1}$
- $SV$: Surface to volume ratio of storage tank $m^{-1}$
- $t$: Thickness of insulation $m$
- $T_a$: Ambient temperature $K$
- $T_{amb}$: Ambient temperature $K$
- $T_c$: Temperature of the heat storage tank $K$
- $T_{c1}$: Temperature of the thermal capacitance $C_{c1}$ $K$
- $T_{c2}$: Temperature of the thermal capacitance $C_{c2}$ $K$
- $T_{coll}$: Temperature of the collector $K$
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$T_f$</td>
<td>Temperature of fluid</td>
<td>$K$</td>
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<td>$T_{fi}$</td>
<td>Temperature of the inlet fluid</td>
<td>$K$</td>
</tr>
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<td>$T_{fo}$</td>
<td>Temperature of the outlet fluid</td>
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</tr>
<tr>
<td>$T_{hin}$</td>
<td>Temperature of the fluid inlet to heat exchanger</td>
<td>$K$</td>
</tr>
<tr>
<td>$T_{hout}$</td>
<td>Temperature of the fluid outlet to heat exchanger</td>
<td>$K$</td>
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<tr>
<td>$T_{LA}$</td>
<td>Maximum load torque at linear actuator</td>
<td>$Nm$</td>
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<td>$T_{Load}$</td>
<td>Temperature of the load</td>
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<tr>
<td>$T_{max}$</td>
<td>Maximum temperature of fluid in tank</td>
<td>$K$</td>
</tr>
<tr>
<td>$T_{min}$</td>
<td>Minimum temperature of fluid in tank</td>
<td>$K$</td>
</tr>
<tr>
<td>$T_{pipe}$</td>
<td>Temperature of pipe</td>
<td>$K$</td>
</tr>
<tr>
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<td>Temperature of the receiver</td>
<td>$K$</td>
</tr>
<tr>
<td>$T_s$</td>
<td>Temperature of storage tank</td>
<td>$K$</td>
</tr>
<tr>
<td>$T_{sin}$</td>
<td>Temperature of the fluid inlet to storage tank</td>
<td>$K$</td>
</tr>
<tr>
<td>$T_{sout}$</td>
<td>Temperature of the fluid outlet to storage tank</td>
<td>$K$</td>
</tr>
<tr>
<td>$u$</td>
<td>Velocity of fluid</td>
<td>$m/s$</td>
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<td>$U_L$</td>
<td>Overall heat loss coefficient</td>
<td>$W/m^2 \degree C$</td>
</tr>
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<td>$U_m$</td>
<td>Overall thermal heat conductivity of heat exchanger</td>
<td>$W/m^2 \degree C$</td>
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<td>$V$</td>
<td>Battery voltage</td>
<td>$V$</td>
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<td>$V_c$</td>
<td>Control voltage</td>
<td>$V$</td>
</tr>
<tr>
<td>$V_f$</td>
<td>Average fuel consumption per day</td>
<td></td>
</tr>
<tr>
<td>$V_{in}$</td>
<td>Input Voltage equivalent to solar energy</td>
<td>$V$</td>
</tr>
<tr>
<td>$V_{aux}$</td>
<td>Input Voltage equivalent to auxiliary source</td>
<td>$V$</td>
</tr>
<tr>
<td>$V_{oil}$</td>
<td>Volume of heat transfer fluid</td>
<td>$m^3$</td>
</tr>
<tr>
<td>$V_{pump}$</td>
<td>Voltage applied to pump</td>
<td>$V$</td>
</tr>
<tr>
<td>$V_{storage}$</td>
<td>Volume of fluid in storage tank</td>
<td>$m^3$</td>
</tr>
<tr>
<td>$W$</td>
<td>Width of the collector</td>
<td>$m$</td>
</tr>
<tr>
<td>$Wh_1$</td>
<td>Energy required from PV during daytime</td>
<td>$Wh$</td>
</tr>
<tr>
<td>$Wh_2$</td>
<td>Energy required from PV during night</td>
<td>$Wh$</td>
</tr>
<tr>
<td>$Wh_{battery}$</td>
<td>Total energy required to store in battery</td>
<td>$Wh$</td>
</tr>
<tr>
<td>$Wh_{total}$</td>
<td>Total energy required from PV</td>
<td>$Wh$</td>
</tr>
<tr>
<td>$x$</td>
<td>Charge/discharge rate indicator</td>
<td></td>
</tr>
<tr>
<td>$\delta$</td>
<td>Declination angle</td>
<td>$\degree$</td>
</tr>
<tr>
<td>$\Delta T_m$</td>
<td>Mean temperature difference between load and fluid</td>
<td>$K$</td>
</tr>
<tr>
<td>$\epsilon_o$</td>
<td>Emissivity of pipe surface</td>
<td></td>
</tr>
<tr>
<td>$\phi$</td>
<td>Latitude angle</td>
<td>$\degree$</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>--------------------------------------------------</td>
<td>---------------</td>
</tr>
<tr>
<td>$\eta_1$</td>
<td>Collector efficiency</td>
<td></td>
</tr>
<tr>
<td>$\eta_B$</td>
<td>Overall battery efficiency</td>
<td></td>
</tr>
<tr>
<td>$\eta_{battery}$</td>
<td>Battery efficiency</td>
<td></td>
</tr>
<tr>
<td>$\eta_c$</td>
<td>Collector efficiency</td>
<td></td>
</tr>
<tr>
<td>$\eta_{coll}$</td>
<td>Effective collector efficiency</td>
<td></td>
</tr>
<tr>
<td>$\eta_o$</td>
<td>Optical efficiency</td>
<td></td>
</tr>
<tr>
<td>$\eta_{pump}$</td>
<td>Efficiency of pump</td>
<td></td>
</tr>
<tr>
<td>$\eta_{PV}$</td>
<td>Efficiency of PV panel</td>
<td></td>
</tr>
<tr>
<td>$\eta_{storage}$</td>
<td>Efficiency of storage system</td>
<td></td>
</tr>
<tr>
<td>$\eta_{Total}$</td>
<td>Effective system efficiency</td>
<td></td>
</tr>
<tr>
<td>$\lambda$</td>
<td>Darcy friction factor</td>
<td></td>
</tr>
<tr>
<td>$\nu$</td>
<td>Kinematic viscosity</td>
<td>$m^2/s$</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density of fluid</td>
<td>$kg/m^3$</td>
</tr>
<tr>
<td>$\rho_L$</td>
<td>Density of load</td>
<td>$kg/m^3$</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Stefan-Boltzmann constant</td>
<td>$W/m^2K^3$</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Angular velocity of DC motor</td>
<td>$rad/s$</td>
</tr>
<tr>
<td>$\omega_{LA}$</td>
<td>Angular velocity of linear actuator</td>
<td>$rad/s$</td>
</tr>
<tr>
<td>$\omega_s$</td>
<td>Hour angle at sunrise/sunset</td>
<td>$degree$</td>
</tr>
</tbody>
</table>
Chapter 1

Solar Cooking: An Overview

1.1 Introduction

Cooking is an integral part of each and every human being as food is one of the basic necessities for living. Commonly used sources of energy for cooking are firewood, crop residue, cow dung, kerosene, electricity, liquefied petroleum gas (LPG), biogas etc. Half of the world’s population is exposed to indoor air pollution, mainly the result of burning solid fuels for cooking and heating. Wood cut for cooking purpose contributes to the 16 million hectares (above 4% of total area of India) of forest destroyed annually. The World Health Organization (WHO) reports that in 23 countries 10% of deaths are due to just two environmental risk factors: unsafe water, including poor sanitation and hygiene; and indoor air pollution due to solid fuel usage for cooking. In under-developed countries, women have to walk 2kms on average and spend significant amount of time for collecting the firewood for cooking. The cooking energy demand in rural areas of developing countries is largely met with bio-fuels such as fuel wood, charcoal, agricultural residues and dung cakes, whereas LPG or electricity is predominantly used in urban areas.

1.2 An Indian perspective

The energy for cooking accounts for 36% of the total primary energy consumption. According to Indian government survey-2001, 52.5% of people use firewood for cooking and LPG is used by 17.5% of the population as shown in table 1.1.
Chapter 1. Solar Cooking: An Overview

<table>
<thead>
<tr>
<th>Sl.No.</th>
<th>Cooking source</th>
<th>People</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Firewood</td>
<td>100,842,651</td>
<td>52.53</td>
</tr>
<tr>
<td>2</td>
<td>LPG</td>
<td>33,596,798</td>
<td>17.5</td>
</tr>
<tr>
<td>3</td>
<td>Crop residue</td>
<td>19,254,851</td>
<td>10.03</td>
</tr>
<tr>
<td>4</td>
<td>Cow dung cake</td>
<td>18,758,885</td>
<td>9.77</td>
</tr>
<tr>
<td>5</td>
<td>Kerosene</td>
<td>12,528,916</td>
<td>6.53</td>
</tr>
<tr>
<td>6</td>
<td>Coal, lignite, charcoal</td>
<td>3,932,730</td>
<td>2.05</td>
</tr>
<tr>
<td>7</td>
<td>Biogas</td>
<td>849,098</td>
<td>0.44</td>
</tr>
<tr>
<td>8</td>
<td>Electricity</td>
<td>338,054</td>
<td>0.18</td>
</tr>
<tr>
<td>9</td>
<td>Other sources</td>
<td>1,231,727</td>
<td>0.64</td>
</tr>
</tbody>
</table>

Table 1.1: Cooking sources used in India

In rural India, energy demand for cooking is met by firewood, agricultural residue, biogas, kerosene and cow dung cake. Using these sources of energy for cooking creates indoor air pollution [1]. Respiratory problems are common due to inhalation of hazardous gases from burning these fuels.

LPG and electricity are the most preferred source of energy for cooking in suburban and urban areas. 17.5% of Indian household or 33.6 million use LPG as cooking fuel. 76.64% of such households are from urban India making up 48% of urban Indian households as compared to a usage of 5.7% only in rural Indian households. In 2005, 27.2% of India’s population lived in urban areas. By 2030 this figure is estimated to grow to 45.8% [2]. Increase in crude oil price in the international market and increase in demand for LPG in India have caused the price of LPG to rise exponentially over the decade. This has forced the government to look for alternative source of energy.

India has abundant amount of solar insolation in most of the regions making it most ideal for harvesting solar energy. With almost 300 sunny days each year, one can confidently rely on this source of energy. India’s geographical location is in such a way that theoretically it receives $5 \times 10^{15}$ kWh/year of solar energy.

Since cooking is integral part of each and every household, cooking with solar energy will reduce the large difference between supply and demand of energy in future. With increasing population and economic growth, utilization of solar energy is a must for sustainable living [3–5].
1.3 Existing cooking sources

Source of energy used for cooking vary from place to place depending on its availability [6]. In rural areas, sources like firewood, biomass and crop residues are easily available [7]. LPG, kerosene are used in oil rich countries or countries which can afford fossil fuels [8, 9]. There are different energy sources used worldwide for cooking which are briefly discussed below.

Firewood:
This is a very commonly used cooking fuel in rural areas. Frequently used hardwood has energy content of 14.89MJ/kg. Out of this about 10.423MJ/kg is recoverable. Smoke from burning of firewood contains water vapor, carbon dioxide and other chemical and aerosol particulates. Wood and wood waste emit 195lbs/10^6Btu of carbon dioxide. Depending on population density, topography, climatic conditions and combustion equipment used, wood heating may substantially contribute to air pollution, particularly particulates. Wood combustion products can include toxic and carcinogenic substances. Particulate air pollution can contribute to human health problems like asthma and heart diseases [10].

Liquefied Petroleum Gas (LPG):
LPG is a mixture of hydrocarbon gases used as a fuel in heating appliances like cooking, water heating and in vehicles. It is also being used as aerosol propellant and a refrigerant to reduce damage to the ozone layer [11]. LPG is synthesized by refining petroleum or wet natural gas, and is usually derived from fossil fuel sources, being manufactured during the refining of crude oil, or extracted from oil or gas streams as they emerge from the ground. Most commonly available LPG is a mixture of propane (60%) and butane (40%) having calorific value of 46.1MJ/kg [11].

LPG is a low carbon emitting (139lbs/10^6Btu of CO_2) hydrocarbon fuel available in rural areas, emitting 19% less CO_2 per kWh than oil, 30% less than coal and greater than 50% less than coal-generated electricity distributed via the grid [10].

Crop Residue:
There are two types of agricultural crop residues, field residues and process residues. Field residues are materials left in agricultural field or orchard after the crop has been harvested, like stalks, stubble (stems), leaves etc. Process residues are those materials
left after processing of the crop into usable resource. Materials like husks, seeds, bagasse (fibrous residue obtained from sugarcane) and roots are used as cooking fuel in most part of the rural places.

**Cow dung:**
Cow dung is used as fuel as well as fertilizer in most of the developing countries like India. Caked and dried cow dung is most common cooking fuel in rural India. Dried cow dung is used as fuel in traditional cook stoves which are not properly ventilated. This has lead to many respiratory health issues. Additionally it is found to be causing arsenic poisoning in unsuspecting villagers. People are simply exposed to 1859.2 ng arsenic per day through direct inhalation, of which 464.8 ng could be absorbed in respiratory tract. Inhalation of arsenic leads to respiratory problems such as persistent coughs and reduced lung capacity [12].

**Kerosene:**
Kerosene is a combustible hydrocarbon liquid which is also known as paraffin. The heat of combustion is similar to that of diesel, having heating value around 43MJ/kg. Its use as a cooking fuel is limited to some portable stoves for backpackers and to less developed countries, where it is usually less refined and contains impurities. Usage of kerosene is not recommended for closed indoor areas without a chimney due to the danger of building up of carbon monoxide gas. Carbon dioxide emitted from kerosene is around 159 lbs/10^6 Btu.

**Biogas:**
Biogas is produced by anaerobic digestion or fermentation of biodegradable materials such as manure or sewage, biomass, municipal waste, green waste and crop residues. This is produced by the biological breakdown of organic matter in the absence of oxygen. Biogas primarily comprises of methane (50%-75%) and carbon dioxide (25%-50%). Small amount of nitrogen, hydrogen, hydrogen sulfide and oxygen is also present in biogas. The other major type of biogas is wood gas which is created by gasification of wood or other biomass. This type of biogas comprises primarily of nitrogen, hydrogen, and carbon monoxide, with trace amounts of methane [13]. Biogas is used as low cost heating application like cooking and also can be used as modern waste management facility where it is used to generate either mechanical or electrical power.

**Electricity:**
The first technology for cooking using electricity used resistive heating coils which heated
Chapter 1. Solar Cooking: An Overview

Iron hotplates or coil, on top of which the vessels were placed. Later in 1970s, glass-ceramic cook-tops started to appear. This has very low thermal conductivity, but lets infrared radiation pass very well, heating up the materials to be cooked. Halogen lamps are also used in some places as a heating element, which gives better efficiency of 60% compared to electric coil which has an efficiency of 55%. A Recent technology which is developed first for professional cooking and has entered domestic market is the induction cooking, which gives around 90% efficiency. These heat the cookware directly through electromagnetic induction. For using this type of cooking, pots and vessels with ferromagnetic bottoms are required. This form of flameless cooking has some advantages over conventional gas flame and electric cookers as it provides rapid heating, improved thermal efficiency, greater heat consistency, plus the same or greater degree of controllability as LPG [14].

1.4 Solar cooking

Due to exponential raise in the world’s population and resulting growth of industrial activities, the energy requirements are such that fossil fuel cannot be the only source on a sustainable basis. This implies that one has to look for alternate sources of energy which are more environmental friendly, cleaner and renewable. As cooking is an integral part of every human being, it consumes a major fraction of the household energy requirement. Existing cooking fuels are either derived from fossil fuels like LPG, kerosene etc. or air polluting like firewood, crop residue, cow dung etc. In this respect, solar cooking is a very simple, clean and environment friendly alternative.

Advantages of solar cooking:

- Prevents cause for global warming and global dimming, since it is clean source of energy. This also reduces indoor air pollution and prevents health problems due to this.
- Moderate cooking temperatures in solar cooking help preserve nutrients.
- Solar cookers can also be used to pasteurize water and food processor.
- Solar cooking reduces deforestation.
• Cooking fires are dangerous, especially for children and can cause damage to building also. In this regard, solar cooker is fire-free and safer compared to conventional cooking.

• Using solar cooking helps in reducing the burden on fossil fuels and hence sustainable.

• Many poverty-stricken families worldwide spend 25% or more of their income on cooking fuel. Sunlight is free and abundant. Money saved can be used for quality food, health care, etc.

• Vessels used for conventional cooking can be used for solar cooking and most of the existing solar cookers are portable.

1.4.1 Existing solar cooking methods and their disadvantages

Many scientists of 17th century knew about the principle of greenhouse effect where glass is used to trap heat from the sun. But Horace de Saussure, a French-Swiss scientist extended this principle to heat up food materials. As early as 250 years ago European scientists started exploring on solar energy, which is considered to be the father of today’s solar cooking movement. Lots of modifications and exploration has been done on solar cooking in order to improve the efficiency and ease of cooking [15–17]. The existing solar cooking methods are discussed in the following paragraphs.

1.4.1.1 Box type cooker

Box type cooker is one of the basic solar cooking models. It is a simple rectangular box covered with either glass or plastic [18–20]. Sun beam entering the cooker through this cover turns to heat energy when it is absorbed by the black colored absorber plate and cooking vessel. This heat input raises the temperature inside the box cooker until the heat loss from the box equals to the solar heat input. The box is thermally insulated to reduce the heat loss to ambient, so as to attain higher temperature and better efficiency [21, 22].

Heat is trapped inside the box due to greenhouse effect. Radiation from sun easily passes through the glass cover and is absorbed by dark materials like absorber plate and vessel. These hot objects emit longer wavelength heat energy which is blocked by the glass cover.
Chapter 1. *Solar Cooking: An Overview*

Hence this trapped energy raises the temperature of the space inside the box. In order to improve the box temperature, reflectors are used. Single or multiple reflectors are used, which are oriented such that the incident sunlight falling on the reflector, get directed to the box [23–25].

Disadvantages of box type cooker:

- Cooking has to be done outdoor and the cook has to stand in sunlight for a long time.
- Very slow cooking compared to conventional cooking. This takes 2-3 hours to cook and even more time during cloudy days with respect to less than half an hour taken by conventional methods.
- It is not possible to cook during rainy days.
- There is no control over the rate of cooking.
- This is not an interactive cooking.

1.4.1.2 Panel cooker

Panel cookers are very inexpensive and can be easily built using reflecting material like aluminium foil and cardboard. This uses shiny material to reflect sunlight to a cooking vessel which is enclosed in a clear plastic bag. Solar Cookers International developed a panel cooker model in 1994, commonly known as 'Cookit'. This can be manufactured locally by pasting reflecting material onto a folded cardboard. Food to be cooked is kept in a dark container covered with a tightly fitted lid. This is placed at the center of the panel. Variety of designs are available in this type of cooker like simple panel, high back, panel for tropics, bubble panel, Dars Diamond cooker etc.

Disadvantages of panel cooker are:

- Cooking has to be done outdoor and not possible to cook during rainy season.
- Very slow cooking compared to conventional cooking. At least one hour is required to cook during clear sunny day.
- Control over the rate of cooking is not there.
Chapter 1. *Solar Cooking: An Overview*

### 1.4.1.3 Parabolic cooker

Parabolic cooker is used for cooking at better cooking rates. It is comparable to conventional cooking and also for community level cooking of food [26–28]. Using a reflector of parabolic shape, sun beam is reflected onto a pot or vessel kept at the focus of the parabola. The axis of the parabola should be parallel to the sun beam so that maximum heat energy is obtained at the pot. Since highly focused light is obtained, temperature can go well above 150°C. Linear parabolic collector is also used for concentrating the sun light on to a pot kept at the focal line [29, 30]. Design and analysis of parabolic square dish solar cooker has been presented by others [31].

Disadvantages of parabolic cooker:

- Cooking has to be carried outdoor. Hence not possible to cook during rainy days.
- At the focal point, highly concentrated sun beam fall which is potentially hazardous.
- Parabolic collector requires either one-axis or two-axis tracking system to track the sun.
- Though cooking is as fast compared to conventional cooking, there is no control over the cooking rate.

### 1.4.1.4 Funnel cooker

In funnel cooker, sun light is concentrated into a cooking vessel or pot using a funnel shaped cooker. This makes use best of both parabolic and box type cooker. Sun beam is concentrated using a reflector like aluminium foil pasted on a folded cardboard. Black colored pot is kept inside a plastic cover which traps the heat by greenhouse effect. A simple system allows pressure-cooking to increase the cooking rate while releasing steam. This cooker can be used for cooking and pasteurizing water.

Disadvantages of funnel cooker:

- Cooking has to be done outdoor.
- There is no control over the cooking rate and very slow cooking compared to conventional cooking.
1.4.1.5 Scheffler cooker

Scheffler cooker is also known as paraboloid dish with fixed focus on ground. Cooking can be carried out in the kitchen without moving out using this type of cooker. The concentrating reflectors track the movement of the sun, reflecting the light of the sun and concentrating it at a fixed position. The reflected and concentrated sunlight enters a nearby kitchen directly to strike a cooking vessel or frying surface. High temperatures can be attained as it focuses the sunlight. Hence the time taken for cooking is comparable to conventional cooking.

In some configurations, the concentrated sunlight is used first to create steam which is transported by pipes to a nearby kitchen. Concentrating reflectors are placed on both the equatorial and polar sides of the receiver. These receivers are attached to the steam pipes that transport the steam to kitchen. Since this steam is generated directly for cooking, the whole system has to be maintained clean. This type of cooking is suitable for preparing food for large mass. The cooking temperature is limited to around 100°C.

Disadvantages of Scheffler cooker:

- The seasonal variation in the height of the sun requires changing not only the angle between the concentrating reflector and its axis of rotation, but also the shape of the reflector.
- Since focus point is fixed, continuous tracking is necessary.
- Focused sunlight falls only on outer part of the cooking vessel resulting in uneven cooking.
- Highly focused light is very hazardous as cooking is done at the focal point.
- There is no option for either decreasing or increasing the rate of cooking.

1.4.1.6 Other Types

In order to cook during cloudy days or night, either auxiliary source of energy is used or some kind of energy storage facility is provided. Box-type cooker with auxiliary heater inside the box is used to cook during cloudy days \[32\]. Solar cooker is also used as food processor where it is also utilized for solar water heating, solar still and solar drier \[33\]
and electric energy is used as auxiliary source in [34].

For collecting energy from sun, different methods like evacuated (vacuum) tube collectors are used which prevents the convection loss from the receiver improving the collector efficiency. Hot plate cooking system powered by solar thermal energy using concentrating evacuated tubular collector has been proposed in [35]. A solar cooking system using vacuum-tube collectors with heat pipes containing a refrigerant as working fluid has been proposed and analyzed [36]. A split-system solar cooker is described which has its flat-plate collector outdoors and the cooking chamber inside the kitchen, with heat pipes transferring the energy between the two [37]. Similarly vacuum-tube collector with integrated long heat pipes directly leading to the oven plate has been tried [38].

A solar cooking system with or without temporary heat storage has been developed and installed in different countries of the world [39, 40]. To reduce the complete dependency on solar insolation, energy storage system is being used. Phase change materials (PCM) are used for many solar thermal applications [41–44]. Solar cooker based on an evacuated tube collector with PCM storage unit has been investigated. Solar energy is stored in the PCM storage unit during sunshine hours and is utilized for cooking in late evening/night time [45–47].

1.5 Modeling techniques for solar systems

There are many different tools for modeling solar thermal systems [48]. The proper sizing of the components of a solar system is a very complex problem which involves both predictable parameters like physical dimensions and unpredictable parameters like weather. Advantage of modeling a system is that one can observe the behavior of different variables like temperature, power etc for different input parameters. This is also helpful in design and selection of physical dimensions in order to optimize the performance of the system.

'Transient simulation', also known as TRNSYS is a quasi-steady state simulation model developed by University of Wisconsin [49]. Mathematical model for the subsystem components like solar collectors, pumps, auxiliary heaters, heating and cooling loads, heat pumps etc. are modeled in terms of their ordinary differential or algebraic equations and validated experimentally [50]. These equations are solved using program written in FORTRAN. Subsystems can be interconnected according to the desired system requirement
Chapter 1. Solar Cooking: An Overview

Only subsystems which are used commonly in solar system are available.

Similarly Watsun Simulation Laboratory of the University of Waterloo, Canada has developed a simulation platform for modeling active solar systems known as WATSUN [53]. This provides information necessary for long-term performance calculation and design of the system. Weather data consisting of global radiation, wind speed, temperature and humidity are used in the simulation.

Another program, Polysun provides dynamical annual simulations of solar thermal systems and is used to optimize them [54]. In addition, this also performs economic viability analysis and ecological balance, which includes emissions from different major greenhouse gasses. Comparison between conventional system and the solar system can be done. To estimate the fraction of a total heating or cooling that will be supplied by solar energy for a given system, a method known as f-chart method [55, 56] is used. There are different parameters like collector area, storage volume, fluid flow rate, size of heat exchanger etc. This chart gives the result which is correlated with other simulated thermal systems. Only steady state results can be obtained, which can be used for draft design of any solar thermal system. It is possible to model a system to a high degree of accuracy, but most of the tools do not consider non-ideality and non-linearities of practical systems. Most of the models are very complex and needs expertise, care and skill to handle properly.

Based on the discussion till now, solar thermal energy is yet to be accepted as a major cooking source due to the following drawbacks.

1. Cooking should be carried outdoor.
2. Very slow cooking compared to conventional cooking.
3. It is not possible to cook during rainy days and no-sun days.
4. There is no control over the rate of cooking.
5. Interactive cooking is not possible.

1.6 Scope of the thesis

The discussion in the previous sections indicates that quite some effort has gone into the issues of solar thermal based cooking. However, solar based cooking has never been a
strong contender in the commercial market or even close to being a preferred method of cooking. They have been relegated to demonstration appliances to show case the solar based concepts. In this mode, cooking is no longer a time independent activity that can be performed at any time of day. One is forced to cook only at certain times when there is sufficient insolation. The geography of the cooking activity also shifts away from the kitchen. The kitchen is no longer the hearth of the home as the actual cooking activity shifts to the roof tops or high insolation platforms. This further adds to the inconvenience apart from being unable to cook at night or during cloudy conditions or during most of the winter days. Another issue of significant inconvenience is the general social structure in most families of the developing countries wherein the cooking activity is carried out by the senior ladies of the home. They are generally not athletic enough to be moving to and from the kitchen and the roof top to carry out the cooking exercise. As the solar cookers are enclosed spaces, interactive cooking is not possible let alone having any control on the rate of cooking. These are some of the more significant issues in the social psyche that has abundantly impeded the acceptance of solar thermal based cooking appliances. These issues and problems are in fact the motivating factors for this thesis. Based on these motivating factors, this thesis aims to propose solutions keeping the following points as the major constraints.

- cooking should be performed in the kitchen.
- one should be able to perform the cooking activity independent of the time of day or insolation.
- the cooking activity should be interactive
- the time taken for cooking should be comparable with the conventional methods in vogue.
- there should be a reduction in the use of conventional energy.

Using the constraints and the motivating factors discussed above as the central theme, this thesis proposes a method to transfer solar thermal energy to the kitchen and act as a supplement to the conventional source of energy like the LPG or other sources that are traditionally being used in the households. The method proposed is in fact a hybrid scenario wherein the solar thermal is used to supplement the traditional source. Solar photovoltaic cells are also used to power the electronics and apparatus proposed in this thesis. This thesis addresses in detail the issues in analysis, modeling, designing and
fabrication of the proposed hybrid solar cooking topology.

The block schematic of the proposed hybrid solar cooking system is shown in figure 1.1. The main goal of the proposed system is to transfer heat from sun to the cooking load that is located in the kitchen. The topology includes an additional feature for storing the energy in a buffer. The heat is first transferred from the solar thermal collector to a heat storage tank (that acts as the buffer) by circulating the heat transfer fluid at a specific flow rate that is controlled by a pump. The stored heat energy that is collected in the buffer is directed into the kitchen by circulating the heat transfer fluid into the heat exchanger, located in the kitchen. This is accomplished by controlling the flow rate using another pump.

Referring to figure 1.1, the solar thermal collector raises the temperature of the thermic fluid. The collector can be of a concentrating type in order to attain high temperatures for cooking. Concentrating collector like linear parabolic collector or parabolic dish collector is used to convert solar energy into heat energy. Absorption of energy from the incident solar insolation is optimized by varying the flow rate of circulating thermic fluid using a pump. This pump is energized from a set of photovoltaic panels (PV cell) which convert solar energy into electrical energy. The energy absorbed from the solar thermal collector is stored in a buffer tank which is thermally insulated. Whenever cooking has to be carried out, the high temperature fluid from the buffer tank is circulated through a heat exchanger that is located in the kitchen. The rate of cooking can be varied by controlling both the flow rate of fluid from the buffer tank to heat exchanger and also by controlling the amount of energy drawn from the auxiliary source. If the available stored energy is not sufficient, the auxiliary source of energy is used for cooking in order to ensure that cooking is independent of time and solar insolation. In the proposed hybrid solar cooking system, the thesis addresses the issues involved in optimization of energy extracted from sun to storage tank and its subsequent transfer from the storage tank to the load.

The flow rate at which maximum energy is extracted from sun depends on many parameters. Solar insolation is one of the predominant parameters that affect the optimum flow rate. Insolation at any location varies with time on a daily basis (diurnal variations) and also with day on a yearly basis (seasonal variation). This implies that the flow rate of the fluid has to be varied appropriately to maximize the energy absorbed from sun.
In the proposed system, flow rate control plays a very significant role in maximizing the energy transfer from the collector to the load. The flow rate of the thermic fluid in the proposed system is very small on the order of 0.02 kg/s. It is very difficult to sense such low flows without disrupting the operating point of the system. Though there are many techniques to measure very low flow rates, they invariably disrupt the system in which flow rate has to be measured. Further, the low flow sensors are far too expensive to be included in the system. A reliable, accurate and inexpensive flow measuring technique has been proposed in this thesis which is non-disruptive and uses a null-deflection technique. The proposed measuring method compensates the pressure drop across the flow meter using a compensating pump. The analysis, modeling, design and fabrication of this novel flow meter are addressed.

The design and implementation of different subsystems that involves the selection and design of solar concentrating collector and tracking are explained. Finally, it is essential to know the economic viability of the proposed system that is designed and implemented. To understand the economics, the life cycle cost analysis of the proposed system is presented in this thesis.
The major contributions of this thesis are:

- **Energy transport**: Major challenge in energy transport is to bring heat energy obtained from sun to the kitchen for cooking. Energy transferred from solar insolation to the cooking load has to be optimized to maximize the overall efficiency. This can be split into two parts, (a) optimizing efficiency of energy transferred from the collector to the energy buffer tank, (b) optimizing efficiency of energy transferred from the buffer tank to the load. The optimization is performed by means of a maximum power point tracking (MPPT) algorithm for a specific performance index.

- **Modeling of the cooking system**: There are several domains that exist in the solar cooking system such as electrical domain, thermal domain, and hydraulic domain. The analysis of power/energy flow across all these domains presents a challenging task in developing a model of the hybrid cooking system. A bond graph modeling approach is used for developing the mathematical model of the proposed hybrid cooking system. The power/energy flow across different domains can be seamlessly integrated using the bond graph modeling approach. In this approach, the various physical variables in the multi-domain environment are uniformly defined as generalized power variables such as effort and flow. The fundamental principle of conservation of power/energy is used in describing the flow of power/energy across different domains and thus constructing the dynamic model of the cooking system. This model is validated through experimentation and simulation.

- **Flow measurement**: A novel method of low fluid mass flow measurement by compensating the pressure drop across the ends of measuring unit using a compensating pump has been proposed. The pressure drop due to flow is balanced by feedback control loop. This is a null-deflection type of measurement. As insertion of such a measuring unit does not affect the functioning of the systems, this is also a non-disruptive flow measurement method. This allows the measurement of very low flow rate at very low resolution. Implementation and design of such a unit are discussed. The system is modeled using bond graph technique and then simulated. The flow meter is fabricated and the model is experimentally validated.

- **Design Toolbox**: Design of hybrid cooking system involves design of multi domain systems. The design becomes much more complex if the energy source to operate the system is hybrid solar based. The energy budget has to be evaluated considering
Chapter 1. *Solar Cooking: An Overview*

the worst case conditions for the availability of the solar energy. The design toolbox helps in assessing the user requirement and help designing the cooking system to fulfill the user requirement. A detailed toolbox is proposed to be developed that can be used in designing/selecting sub-systems like collector, concentrator, tracking system, buffer tank, heat exchanger, PV panel, batteries etc. The toolbox can also be used for performing life cycle costing.

Chapter 2 discusses the performance analysis of the proposed hybrid solar cooking system. The analysis focuses on the optimization issues and implementation through maximum power point tracking method which is used for deciding the optimal flow rate. Experimental and simulation results are presented in this chapter.

The bond graph model of the proposed system is discussed in-depth in chapter 3. The various parameters are estimated by system identification techniques as discussed in the chapter. The bond graph model of the proposed system is simulated and experimentally validated by fabricating a prototype.

Chapter 4 discusses the motivation and need for low flow measurement apparatus. A novel method to measure very low flow is proposed. The flow meter is analyzed, modeled and simulated. It is fabricated and tested to check that the proposed model is experimentally verified.

Chapter 5 addresses steps that are followed for designing a cooking system. This chapter presents the specification, selection and design of different subsystems like solar thermal collectors, tracking system, pump, photovoltaic cells, storage tank, heat exchanger and other devices. A section on measurement and monitoring electronics is also presented.

Chapter 6 addresses the life cycle cost analysis and other economic considerations. A tool box is created comprising of all the governing equations that are used in designing the proposed hybrid cooking system. This tool box can be used in sizing different sub-systems of the cooking system depending on the user requirement.

Thesis is concluded in chapter 7 with a section on future scope of the research work in this area.
Chapter 2

Maximum power point tracking

2.1 Introduction

Solar cooking has been developed and improved over last few decades. Initially in 1980s, the basic model of box type cooker was being used. This traps the heat from sun and is used for heating up the food. In order to decrease the time taken for cooking, reflectors are fixed to the box to trap more heat energy. These types of cookers are called as box type with multiple reflectors. They can be easily built using readily available materials, but takes 3-5 hours for cooking depending on the solar insolation. Panel cooker uses shiny material to reflect sunlight to a cooking vessel which is enclosed in a clear plastic bag. Various designs of panel cookers are available like high back, simple panel, panel for tropics, bubble panel, Dars diamond cooker etc. The reflector material is folded into a bowl shape and the cooking vessel is placed at the center where the sun’s beams focus. Another variation in solar cooking is the funnel cooker which combines the best of both parabolic cooker and box type cooker. Though cooking is faster compared to box type, it is not comparable with that using conventional source of energy. Parabolic cooker focuses larger amount of sun beam resulting in high temperature at the focal point. During clear sky day, cooking can be done as fast as compared to conventional cooking. But this is potentially hazardous as person who is cooking has to operate near focal point of the parabola. Control over the cooking rate is not there in any of the above mentioned cooking methods. Cooking has to be carried out either on rooftops or outdoors. Hence it is not possible to cook during rainy season. To overcome these problems, an alternate technique is being proposed where the energy from sun is transported to the kitchen. Cooking can be carried out without moving out of the kitchen.
Scheffler has proposed a parabolic cooker with fixed focus on ground. Parabolic reflector is used to focus sun beam directly into the kitchen. In this type, cooking can be carried out inside the kitchen without moving out. The disadvantage of this type is uneven cooking as heat is focused on only one side of the vessel. There is no control over the rate at which cooking is done. In the proposed cooking system, the rate of cooking can be controlled as per the user’s convenience.

In most of the existing solar cooking system, the time taken for cooking is dependent on availability of solar insolation at the location. Using conventional source of energy as an auxiliary source, time taken for cooking is made comparable with the conventional cooking methods. Energy from the sun is stored in a heat storage tank so that cooking can be carried out during cloudy days or during night time.

Consumption of conventional cooking fuel is minimized by maximizing the energy obtained from sun. This maximization is done by varying the flow rate of the fluid that is being circulated through the solar thermal collector. At lower flow rates, temperature of the collector and outlet fluid are higher resulting in higher heat loss to ambient. Increasing the flow rate increases the energy required for circulating the fluid, even though the collected heat energy improves. There exists an optimal flow rate for the given solar insolation and other external factors. Hence there is a need for variation in the flow rate dynamically in order to optimize the energy absorbed from the collector. This has lead to the concept of maximum power point tracking (MPPT). One more factor which decides the level of sophistication of solar cooker is control on the rate of cooking. This is achieved in the proposed system by varying the flow rate of heat transfer fluid through the heat exchanger.

This chapter proposes a hybrid solar cooking system, where solar energy is transported to the kitchen. Section 2.2 discusses the proposed system and its operation. Energy optimization using maximum power point tracking is explained in section 2.3. Simulation and experimental results are presented in chapter 2.4.
Chapter 2. Maximum power point tracking

2.2 Solar Cooking System

2.2.1 System Description

The block diagram of the proposed cooking system is as shown in figure 2.1. The solar thermal collector is in general placed at a high location preferably on the roof top. Since temperature above 100°C is necessary for this application, concentrating collector is used to collect solar energy and increase the temperature of the fluid. The heat exchanger is placed in the kitchen where the cooking is done. It transfers heat from the circulating fluid to the cooking load. All other components are placed at intermediate levels according to the building requirements. Pump-I is used to vary the flow rate of the fluid through the solar thermal collector. The energy extracted from the sun is stored in the buffer tank. The size of this tank is decided by the amount of energy that needs to be stored for late night or early morning cooking and amount of energy that needs to be saved from the other energy sources of the hybrid system.

Whenever food has to be cooked, the stored energy is transferred to the load through the heat exchanger. Using pump-II, flow rate of the fluid through the heat exchanger is varied. The auxiliary source of energy like LPG or electrical energy is used for supplementing the stored solar energy and it also reduces the time required for cooking as compared to previously proposed cooking systems like box-type cooker. Energy required from the auxiliary source is to be optimized for the given system, solar insolation at the location and the load profile.
2.2.2 System operation

The aim of the cooking system is to provide heat energy necessary for the cooking operation. In the proposed system, the main goal is to collect heat energy from sunlight and transfer it to the cooking load. This can be divided into two parts,

- Collecting solar energy using thermal collector and transferring it to the heat storage tank.
- Transferring heat energy from the buffer tank to the load.

2.3 Energy Optimization

2.3.1 Solar Heating

Solar heating is used for different applications like solar water heating, space heating, solar drying, solar desalination etc, to heat up water, air or any other fluid. If solar energy is absorbed and utilized without using any other external active sources like pump or blower, such a system is called as passive [57-59]. In some systems, active sources are used to move the fluid; such a system is called as active system. Flat plate collectors are used for applications where temperature less than 100°C is sufficient. Stagnation temperature of these type of collector is not more than 70°C-80°C. Advantage of flat plate collector is that it can absorb both direct radiation and diffused radiation. Hence it heats up liquid even during cloudy days.

For applications which require medium or high temperature, concentrating collectors are used. A concentrating collector comprises of a receiver, where the sunlight is absorbed and converted to required form of energy and a concentrator, which is an optical system to concentrate direct radiation onto the receiver. Concentration ratio CR is defined as the ratio of the area of aperture $A_a$ to the area of the receiver $A_r$.

$$CR = \frac{A_a}{A_r} \quad (2.1)$$

Temperature ranging from 100°C up to 2000°C can be obtained by increasing the concentration ratio. Method to concentrate sun beam can be either by refraction as in case of fresnel lens or by reflection. Reflection type of concentrating collectors are flat plate
Chapter 2. *Maximum power point tracking*

![Diagram of energy flow in solar collector](image)

**Figure 2.2: Energy flow in solar collector**

A concentrating collector with plane mirror reflectors, linear (cylindrical) parabolic collector, compound parabolic collector, paraboloid collector etc.

Figure 2.2 shows different energy flow paths in a concentrating collector. Total power available at a concentrating collector is given by equation (2.2), where \( G_b \) is the beam or direct irradiance in \( W/m^2 \).

\[
P_{in} = G_b A_a
\]  

(2.2)

Only part of the power which is falling on the reflector, is concentrated onto the receiver depending on the optical efficiency \( \eta_o \) given by (2.3). This efficiency depends on many factors like accuracy of tracking mechanism, material and shape of the reflector [62]. \( P_r \) represents the amount of power available at the receiver.

\[
P_r = G_b \eta_o A_a = \eta_o P_{in}
\]  

(2.3)

Under steady state conditions, useful power \( P_u \), obtained at the receiver can be expressed as in equation (2.4). \( P_L \) is the amount of power lost from the receiver to atmosphere through convection and radiation, where \( U_L \) is solar collector heat loss coefficient in \( W/m^2 \, °C \) and \( T_r \) is the temperature of the receiver in \( K \).

\[
P_u = P_r - P_L = G_b \eta_o A_a - U_L (T_r - T_{amb}) A_r
\]  

(2.4)

The useful energy gain per unit length of the receiver can be expressed as,

\[
P'_u = \frac{P_u}{L} = \frac{G_b \eta_o A_a}{L} - \frac{U_L (T_r - T_{amb}) A_r}{L}
\]  

(2.5)

This useful power per unit length can be obtained in terms of fluid temperature as given in equation (2.6).

\[
P'_u = \frac{F' A_a}{L} \left[ G_b \eta_o - \frac{U_L (T_f - T_{amb})}{CR} \right]
\]  

(2.6)
Where $F'$ is called as collector efficiency factor and $T_f$ is the local fluid temperature. For collector having circular cross section with outer and inner diameter as $D_o$ and $D_i$, $F'$ is given by (2.7) [61].

$$F' = \frac{1}{U_L \left[ \frac{1}{U_L} + \frac{D_o}{h_{fi}D_i} + \frac{D_o}{2k} \ln \left( \frac{D_o}{D_i} \right) \right]} \quad (2.7)$$

Where $k$ is absorber thermal conductivity in $W/m^\circ C$ and $h_{fi}$ is heat transfer coefficient inside absorbed tube in $W/m^2 ^\circ C$. Useful power gained at the collector can be expressed in terms of inlet fluid temperature $T_{fi}$ and ambient temperature as in equation (2.8), where $F_R$ is the heat removal factor. $F_R$ is a function of collector efficiency, specific heat of fluid, flow rate, dimension of collector, heat loss coefficient $U_L$ etc. Equation (2.8) is the equivalent of the Hottel-Whiller-Bliss equation for liquid flat-plate collector [60, 61].

$$P_u = F_R \left[ G_b \eta_o A_a - U_L (T_{fi} - T_{amb}) A_r \right] \quad (2.8)$$

Dividing the above equation by $P_{in}$, steady state collector efficiency can be obtained as,

$$\eta_c = \frac{P_u}{P_{in}} = \frac{P_r - P_L}{P_{in}} = F_R \left[ \eta_o - \frac{U_L (T_{fi} - T_{amb})}{G_b \times CR} \right] \quad (2.9)$$

Applying first principle of thermodynamics to the solar collector for open systems, useful power gain can also be written as rate of specific heat gained by the fluid.

$$P_u = \dot{m} C_p \Delta T_f \quad (2.10)$$

From equation (2.9), it is observed that the collector efficiency $\eta_c$ depends mainly on two factors $P_r$ and $P_L$. Power reflected from the concentrator can be optimized by improving the optical efficiency $\eta_o$. This can be achieved by using reflector having better reflectivity and shape. Once the reflector is fabricated, optical efficiency is fixed and can not be improved further.

The other part of the collector efficiency is $P_L$, which accounts for radiative and convective heat loss from the receiver to atmosphere. This is a function of the receiver temperature. Higher the temperature more is the heat loss. In order to increase the collector efficiency, the temperature of the receiver has to be kept as close as possible to the ambient temperature in order to reduce heat loss to ambient according to equation (2.5).

For a given constant inlet temperature of fluid and constant solar irradiance, as flow
rate increases, the collector efficiency increases as the collector outlet temperature comes down. Since heat loss from collector to atmosphere is directly proportional to receiver temperature, efficiency of collector is lesser at smaller fluid flow rates. At very large flow rates, outlet temperature tends to inlet temperature according to equation (2.10), as input power is considered to remain same.

Performance of linear parabolic collector is found according to the above equation as given in appendix A. This MATLAB program calculates steady state useful collected power and collector efficiency analytically. Instantaneous collector efficiency is calculated for an inlet fluid temperature of 150°C as in equation (2.11) where $I_b$ and $I_d$ are beam and diffused radiation, $W$ is the width of the collector, $L$ is the length of the collector, $\dot{m}$ is the mass flow rate, $C_p$ is the specific heat of the fluid, $T_{fi}$ and $T_{fo}$ are the inlet and outlet temperature of the fluid. Tilt factor for beam and diffused radiation are represented by $R_b$ and $R_d$ respectively. Collected useful heat is calculated for different mass flow rates and plotted in figure 2.3.

$$\eta_c = \frac{\dot{m}C_p(T_{fo} - T_{fi})}{(I_bR_b + I_dR_d)WL} \quad (2.11)$$

Inlet temperature to the collector $T_{fi}$ is varied by keeping the mass flow rate constant.
For lower inlet temperature, receiver temperature is less. Since heat loss $P_L$ is minimum, collector efficiency is maximum. On the other hand, as inlet temperature is increased, receiver temperature is more resulting in lower collector efficiency.

For linear parabolic collector, inlet temperature is set at three different values (50°C, 100°C and 150°C) and collector efficiency is calculated for varied range of mass flow rate as shown in figure 2.4. It can be observed from the plot that the collector efficiency increases as inlet temperature is decreased towards ambient.

### 2.3.2 Thermal energy storage

The storage system acts as a buffer between the collection system and the load exchanger. This stores energy when collected energy is more than the required amount and discharges when the collected amount is inadequate to supply the load requirement [64–66]. There are two major types of thermal storage technique, namely sensible heat storage and latent heat storage. In sensible heat storage, phase/state of the heat storage material will remain same either fluid or solid. Energy is stored by raising the temperature of the material. Where as, in case of latent heat storage, heating a material changes its phase at some particular temperature depending on its melting point or boiling point. The amount of heat stored depends upon the mass and the latent heat of fusion of the material. Material used for this type of application is referred as phase change materials (PCM). Though latent heat storage is an isothermal process, the system becomes complex as handling
two phases is difficult. For solar heating application, sensible heat storage in liquid form is preferred, since circulation of the fluid is easier. Energy stored for a given volume is higher in liquid as compared to gaseous form.

Consider a closed loop setup consisting of solar concentrating collector, a fluid pump and a storage tank with thermal insulation as shown in figure 2.5. Rate of fluid flow is varied by changing supply voltage given to the pump. At very low flow rate, efficiency of the collector is very less as the outlet temperature is high. On the other hand, when flow rate increases, collector efficiency increases and temperature stratification in storage tank is disturbed. However, more power has to be supplied to the pump to increase the flow rate. These two conflicting effects imply the existence of an optimal flow rate that gives optimal energy from sun.

**Stratification:**  [67–69] The term stratification is defined as a measure of the difference between the maximum and minimum storage tank temperatures at any given time. The basic phenomenon causing temperature stratification is change in density with change in temperature. For most of the fluids used in solar thermal storage, density decreases with increase in temperature. Density difference between hot and cold fluid in storage tank drives the fluid into a thermosyphon system (passive solar heating).

Characterization of stratification is analyzed by many researchers [70–73]. Different stratification indexes have been proposed which measure the amount of mixing and temperature distribution profile. Simulation and analysis of temperature profile is still a challenging
exercise. In a standard method, the storage tank is split into number of nodes having equal volume having constant temperature throughout that volume. Their temperature and heat transfer between adjacent nodes are calculated analytically.

Maintaining thermal stratification requires inhibition of mixing in the tank. Loss of stratification results from convective mixing between hot and cold layer and due to momentum force. Mixing depends on the design of the tank and the operating conditions like flow rate and temperature of incoming fluid and the temperature distribution in the tank. Forced convection mixing is due to the momentum of the fluid streams entering the storage tank and depends on the flow rate of the inlet fluid and the design of the inlet.

Stratification can be increased by a reduction in collector flow rate, which permits a larger temperature difference in storage tank. From figure 2.4, it is observed that inlet temperature has to be minimal to increase the collector efficiency. Maximum possible thermal performance is obtained when no mixing occurs in the storage tank. In this case, coldest stored fluid is circulated through the collector. Better stratification in storage tank helps in reducing the inlet temperature to the collector and hence the collector efficiency improves. Hence collector performance is better at lower flow rate considering stratification factor.

**Pumping power:** Power necessary for driving pump depends on type of the pump that is being used for circulating fluid. Useful power $P_o$ that is being added to the fluid flow is given by,

$$ P_o = \rho \ g \ H \dot{q} \quad (2.12) $$

Where, $\rho$ is the fluid density in $kg/m^3$, $H$ is the head added to the flow in $m$ and $\dot{q}$ is the flow rate in $m^3/s$. Input power necessary to drive the pump to deliver pressure of $\Delta P$ is much more than the output power given by equation (2.13), where the efficiency of the pump $\eta_{pump}$ is being considered.

$$ P_{pump} = \frac{\Delta P \times \dot{q}}{\eta_{pump}} \quad (2.13) $$

For a centrifugal pump, input power is proportional to flow rate given by equation (2.14). $k_{pump}$ is a pump constant depending on the type and design of the pump. As the mass flow rate increases, power required to circulate fluid through the collector increases drastically.

$$ P_{pump} = k_{pump} \dot{m}^3 \quad (2.14) $$
Figure 2.6: Pumping power

Figure 2.6 shows the pumping power required to circulate fluid through the collector. As the mass flow rate increases, power required to pump increases drastically. For different pump constant $k$, pumping power is plotted against variation in flow rate.

**Effective collector efficiency:** In an active mass transport solar system, additional energy is spent on the supply-side pump to drive fluid through the collector in order to extract energy from the sun. Hence efficiency of the system containing solar collector and pump can be expressed as given in equation (2.15). Figure 2.7 shows the variation in effective power collected by a concentrated solar collector for a constant inlet temperature. Initially, as the flow rate increases, power extracted from the collector increases. After reaching some flow rate, power required to pump the fluid increases more drastically as compared to increase in collector power. Figures 2.6 and 2.7 indicate the presence of an optimum flow rate at which the effective power collected is maximum. For different inlet temperatures, effective collector power is calculated and plotted.

Effective collector efficiency is calculated from this collector power as given in equation (2.15). $P_{pump}$ can be expressed in terms of $P_u$ and equation can be written, where $H_f$
Chapter 2. *Maximum power point tracking*

**Figure 2.7:** Effective Collector power

represents the friction head in the collector side loop.

\[
\text{Effective Collector Efficiency} = \frac{\text{Effective Collected Power}}{\text{Input Solar Power}} = \frac{P_u - P_{\text{pump}}}{P_{\text{in}}} = \frac{P_u}{P_{\text{in}}} \left[ 1 - \frac{g H_f}{C_p \Delta T_f \eta_{\text{pump}}} \right]
\]

For linear parabolic collector, the effective collector efficiency is analyzed for a constant inlet temperature. Efficiency is plotted against mass flow rate for different inlet temperatures as shown in figure 2.8.

At lower flow rates, increase in collector efficiency is more compared to increase in power required to pump the fluid. Effective collector efficiency increases at lower flow rates. Above some particular flow rate, pumping power increases and there is a flow rate at which power extracted from solar collector is maximum. In order to maximize the collector efficiency, one has to operate at this peak point. The effect of the pump constant \(k\) on the collector efficiency is plotted in figure 2.9. Flow rate corresponding to optimal effective collector efficiency varies as \(k\) changes.
2.3.3 Maximum Power Point Tracking (MPPT)

Performance analysis is carried out for different solar insolation level at constant ambient air temperature and inlet thermo-convector fluid temperature. From figure 2.10, it can be observed that for 100% solar insolation, optimal mass flow rate is around 0.04-0.05 kg/s. As solar insolation is reduced, the optimal flow rate shifts towards lower value. This shows that the optimal flow rate is not fixed for a given system. It is a function of location and solar insolation at that particular instant. Hence one has to vary the flow rate accordingly in order to extract optimal energy from the sun. This leads to the concept of maximum
Chapter 2. Maximum power point tracking

Figure 2.10: Effective Collector efficiency for different insolations

power point tracking also known as MPPT.

MPPT control block diagram is shown in figure 2.11. Inlet temperature $T_{\text{sin}}$ and outlet temperature $T_{\text{sout}}$ from the storage tank are sensed using temperature sensors. Power collected from the solar collector is calculated according to the equation given by $(2.16)$, where the mass flow rate $\dot{m}$ is sensed using a flow meter. Effective collector power which is calculated from $P_{\text{coll}}$ and $P_{\text{pump}}$, is fed to the MPPT optimizing algorithm. This gives duty ratio as output, which in turn controls the voltage level of the supply given to the pump through the power converter as shown in figure 2.11.

$$P_{\text{coll}} = \dot{m} C_p (T_{\text{sin}} - T_{\text{sout}})$$  \hspace{1cm} (2.16)

MPPT algorithm can be grouped into two major types, namely direct and indirect methods [74]. Indirect methods track the peak power from the measures of the system like temperature, irradiance or using empirical data or lookup table, by mathematical expressions through numerical approximations. For this, characteristic of the system has to be known before hand and it is specific only to that system. Hence, they do not track maximum power for any given external conditions exactly.

On the other hand, direct method calculates the actual power absorbed from the sun
This is an iterative method of obtaining MPP. This measure the solar effective collector and tries to maximize it. These types of algorithms are suitable for any irradiance and temperature. Most commonly used MPPT algorithm is perturbation and observation method [75]. This is an iterative method of obtaining MPP. This measures the solar collector characteristics and then perturbs the operative point to know the direction of change in power. Maximum point is reached when change in power collected is zero for a small change in the flow rate. Amount of the perturbation given is calculated according to the slope of the present operating point. This method has advantage over other methods as it can be applied to any system without prior knowledge about its actual characteristic.

Table 2.1 gives the pseudo code for implementing the MPPT algorithm. The power obtained from the collector and the power supplied to the pump are calculated. Rate of change in effective collector power is estimated. Accordingly the value of duty ratio is calculated. Duty ratio is either increased or decreased depending on the present operating point. If the operating point is on the left side of the inverted ‘U’ curve (figure 2.10), then change in duty ratio and change in power are of same sign. Hence duty ratio is increased such that operating point moves toward optimal value. On the other hand, change in duty ratio and power is in opposite direction when operating right side of optimal flow rate. The gain value $K$ is decided according to the system dynamics like time constant of the system and variation in solar insolation. Time interval at which temperatures and flow rate are sensed also decides selection of $K$.

Flow chart for implementing MPPT is as shown in figure 2.12. Effective collector power is calculated from feedback signals. Variation in this power with respect to variation in duty ratio decides the output of the program. Duty ratio obtained from this controls the voltage level that is being supplied to the pump.
\[ D = \text{MPPT}(T_{\text{in}}, T_{\text{sout}}, \dot{m}, V_p, I_p) \]

Local Variables: \( P_k, P_{k-1}, \Delta P_k, \Delta P_{k-1}, \Delta D_k, \Delta D_{k-1}, D_{k-1}, D_k, S \)

Constants: \( P_{\text{band}}, K \)

\[
P_k = \dot{m}C_p(T_{\text{in}} - T_{\text{sout}}) - V_pI_p
\]

\[
\Delta P_k = P_k - P_{k-1}
\]

if \( \Delta P \geq P_{\text{band}} \)

\[
\{ \text{if} (\Delta D_{k-1} = 0) \\
\quad \Delta D_k = K \times \Delta P_k \\
\text{else} \\
\quad \{ S = \text{sign}(\Delta P_k) \times \text{sign}(\Delta D_{k-1}) \\
\quad \Delta D_k = K |\Delta P_k| S \\
\} \\
\} \\
\}

else

\[
\{ \Delta P_k = \Delta P_{k-1} \\
\quad \Delta D_k = 0 \\
\} \\
\}

\[ D_k = D_{k-1} + \Delta D_k \]

\[ D = D_k \]

\text{Table 2.1: Algorithm for MPPT}
Chapter 2. Maximum power point tracking

Figure 2.12: Flowchart for MPPT algorithm
2.3.4 Load-side optimization

The other part of the system transfers heat energy from the buffer tank to the load which is placed in the kitchen. The buffer tank is used to store the heat energy in the form of sensible heat which is used for cooking. With this storage unit, cooking can be carried out even when energy from sun is not available. The block diagram of the energy transfer from the buffer tank to the load is shown in figure 2.13. The heat energy is transferred to the load by circulating the hot fluid stored in the buffer tank through a heat exchanger. The pump controls the flow rate in this loop thereby controlling the rate of heat transferred to the load.

The characteristic equation for a heat exchanger is given by equation (2.17). $U_m A$ gives the mean overall thermal conductivity between the heat exchanger and the load. This is decided by the dimension, shape of the heat exchanger and conductivity of the material used.

$$P_{fl} = U_m A \Delta T_m$$  \hspace{1cm} (2.17)

Where $P_{fl}$ is the heat flow from fluid to load, $U_m$ is the mean overall heat transfer coefficient, $A$ is the total surface area of the load heat exchanger and $\Delta T_m$ is the mean temperature difference between fluid and load. It is observed from equation 2.17 that $\Delta T_m$ should be maximized to increase the heat transfer rate. For a given temperature difference $\Delta T_m$, energy flow can be increased by maximizing the conductive barrier between the hot fluid and the cooking load. Design should be done such that the energy transferred to the load should be maximum with minimal pressure drop across it. $\Delta T_m$ can be maximized by taking the fluid from top of the storage tank which has the highest

![Figure 2.13: Block diagram for energy transfer from storage to load](image-url)
temperature due to density stratification. Once the heat energy is transferred to the load, the lower temperature fluid is fed back to the buffer tank.

The above equation can also be written as,

\[ P_{fl} = \dot{m} \cdot C_p \cdot (T_{hin} - T_{hout}) \]  

Where, \( \dot{m} \) is the mass flow rate through the heat exchanger, \( C_p \) is the specific heat of the fluid, \( T_{hin} \) and \( T_{hout} \) are the temperatures of fluid entering and leaving the heat exchanger respectively. Power transferred to the load can be controlled by varying the mass flow rate \( \dot{m} \). Pump-II is supplied by a variable power supply unit by which the flow rate can be controlled. When power supplied from the storage tank is not sufficient, the auxiliary source of energy like LPG or electrical energy is used.

LPG and solar energy are integrated in such a way that, both of them heat up only the load and are mutually exclusive. During the condition when both energy sources are supplying heat, the temperature of the heat-transfer fluid is higher than load. In such a case the entropy flow is to the load (sink). If the temperature of heat-transfer fluid is lower (during night or no-sun conditions), the piping structure is such that natural thermosyphon of the thermic fluid from the load side to buffer tank does not exist. Further isolation is provided by two insulating isolation valves that isolate the thermic fluid at the load exchanger and the buffer tank.

### 2.3.5 Selection of pipe diameter

Performance of the solar system is dependent on the pipe diameter used for circulation of the fluid. There are two contradictory effects which are dependent on diameter of the pipe. One of them is the heat loss to ambient, which varies directly with variation in diameter. Another effect is the pumping power required to circulate the fluid which increases as diameter is decreased. These two contradicting effects lead to the existence of an optimal pipe diameter for a given system.
2.3.5.1 Pressure drop

Consider a pipe of length $l$ having a constant diameter $D$. Pressure drop across this pipe is given by the Darcy-Weisbach formula \[77\] as in equation .

$$P_{\text{pipe}} = \lambda \frac{l}{D} \frac{\rho u^2}{2}$$  \hspace{1cm} (2.19)

Where, $u$ is the velocity of the fluid in $m/s$, $\rho$ is the density of the fluid and $\lambda$ is a dimensionless coefficient called the Darcy friction factor or coefficient of line hydraulic resistance, which can be found from a Moody diagram \[77\]. In case of laminar flow of fluids, the Darcy factor is calculated using Stoke’s formula \[77\] given by,

$$\lambda = \frac{64}{Re}$$  \hspace{1cm} (2.20)

Where $Re$ is the Reynold Number, given by

$$Re = \frac{D \times u}{\nu}$$  \hspace{1cm} (2.21)

Where, $\nu$ is the kinematic viscosity of the fluid. By substituting equations 2.20 and 2.21 in equation 2.19, the pressure drop along a pipe is obtained as,

$$P_{\text{pipe}} = 32 \nu \rho \frac{l}{D^2} u = \frac{64 \nu l m}{2 \pi \frac{D^4}{4}}$$  \hspace{1cm} (2.22)

In most of the solar applications, flow rate is very small and Reynolds number $Re$ is less than 2000. Hence the flow is considered as laminar. If flow is turbulent, the friction factor $\lambda$ is calculated according to a fit given by Colebrook \[78\],

$$\frac{1}{\sqrt{\lambda}} = -2 \log \left[ \frac{2.51}{Re} \sqrt{\lambda} + \frac{0.269 k_p}{D} \right]$$  \hspace{1cm} (2.23)

Where, $k_p$ is the absolute roughness of the pipe material used.

For both laminar and turbulent flow, as the diameter of the pipe decreases, the pressure drop increases. This increases the burden on the pump which is circulating the fluid. For a given flow rate, higher power is necessary to pump the fluid through a smaller diameter pipe.
Chapter 2. Maximum power point tracking

Figure 2.14: Effect of diameter on the performance

Figure 2.15: Variation in optimal diameter with fluid temperature
2.3.5.2 Heat loss

Heat energy is transferred from collector to storage tank and from storage tank to heat exchanger by circulating heat transfer fluid through the pipe. Since temperature of the fluid is much higher than ambient temperature, the fluid looses heat energy to ambient. Consider a pipe of length \( l \) carrying fluid having temperature \( T_f \). Heat is lost from fluid by conduction through the pipe wall given by equation (2.24), where inner and outer diameter of the pipe are represented as \( D_i \) and \( D_o \) and \( k_1 \) is the conductivity of the pipe material [79].

\[
P_{loss} = \frac{2\pi k_1 l}{\ln \left( \frac{D_o}{D_i} \right)} (T_f - T_{pipe}) \tag{2.24}
\]

This heat energy raises the temperature of the pipe. Heat from pipe is lost to ambient through the insulation by conduction and radiation given by [79],

\[
P_{loss} = \frac{2\pi k_2 l}{\ln \left( \frac{D_o+2t}{D_o} \right)} (T_{pipe} - T_{amb}) + A_o \sigma \epsilon_o (T_{pipe}^4 - T_{amb}^4) \tag{2.25}
\]

Where, \( A_o \) is the outer surface area of the pipe, \( k_2 \) is the conductivity of the insulation material, \( t \) is the thickness of the insulation, \( \sigma \) is the Stefan-Boltzmann constant and \( \epsilon_o \) is the emissivity of the outer surface.

Under steady state, all the temperatures reach an equilibrium value. Under this condition heat transferred from fluid to pipe becomes equal to heat transferred from pipe to ambient. For a given value of fluid and ambient temperatures and pipe temperature \( T_{pipe} \), the power loss \( P_{loss} \) can be calculated by solving the non-linear equations (2.24) and (2.25). It is observed from the above equations that the heat loss to ambient is directly proportional to the surface area of the pipe. Hence heat energy lost is higher if larger diameter pipes are used.

Discussions in 2.3.5.1 and 2.3.5.2 show that \( P_{pump} \) and \( P_{loss} \) change inversely as the pipe diameter is varied. Hence there exists an optimal pipe diameter for which \( P_{pump} + P_{loss} \) is minimal. For a given solar system, diameter of the pipe is selected in order to minimize the sum of these two power losses. Optimal diameter of the pipe is obtained by solving the equation (2.26).

\[
\frac{\partial P_{pump}}{\partial D} = -\frac{\partial P_{loss}}{\partial D} \tag{2.26}
\]
Chapter 2. Maximum power point tracking

\( P_{\text{pump}} \) and \( P_{\text{loss}} \) are calculated for a typical set of specifications. Variation in these powers are plotted with change in diameter of the pipe. Figure 2.14 shows that as the diameter increases, \( P_{\text{loss}} \) increases whereas \( P_{\text{pump}} \) decreases. From the plot of \( P_{\text{loss}} + P_{\text{pump}} \), it is observed that there exists an optimal diameter at 8.26 mm for which the total power loss is minimum.

Optimal pipe diameter is calculated for variation in parameters like viscosity, density and temperature of the fluid. It is observed that, even though there is a large variation in the minimum power loss, optimal diameter is not varying much. Figure 2.15 shows the effect of the fluid temperature on the optimal pipe diameter. Optimal pipe diameter is found to be in between 7 mm to 8.9 mm considering tolerance in the system parameters.

2.4 Experimental Results

Figure 2.16 shows the block diagram of the experimental setup for solar cooking system. Paraboloid dish concentrator is used to focus sun rays onto the receiver. Aluminium sheets are used as reflecting material. To improve optical efficiency, surface of the reflector is anodized. A linear actuator is fixed to the paraboloid with a lever system in such a way that when actuator moves to and fro, the paraboloid is rotated in east-west direction. Using an accelerometer sensor that is fixed on the paraboloid, the tilt angle is sensed. Early morning, the concentrator is fixed toward sun manually. The sensor considers this as the reference angle and tracks at a constant rate of 15° angle per hour.

A coil made of copper tube is placed at the focus of the parabola in order to receive the heat. Servo-therm oil is circulated through the collector to absorb heat energy. Stainless steel pipe with glass wool insulation over that is used for circulation of the oil from the collector to the tank. Thermocouples are placed to measure temperature of oil entering and leaving the receiver. A rotary pump is used to circulate oil through the receiver and put back into the heat storage tank. The pumps are driven by permanent magnet DC (PMD) motors, which are controlled by variable voltage power supplies. The heat storage tank is made of stainless steel material with good thermal insulation around that for better retention of heat. The flow rate of the fluid is measured using a flow meter (which will be discussed in detail in chapter 4).

On the load-side, hot oil from top of the heat storage tank is taken to the kitchen through
thermally insulated stainless steel tube. Heat is transferred from oil at higher temperature to cooking load using a heat exchanger. Helical shaped coil of copper is wound around the cooking vessel with thermal insulation to constrain the heat within the heat exchanger. Oil leaving the heat exchanger is pumped back from the kitchen to the buffer tank using another similar pump-motor drive. The mass flow rate on the load-side is measured using another flow meter. Thermocouples are placed at different places as shown in figure to measure the temperatures at various points of the system.

In order to show the maximum power point, an electrical heater is used in place of solar thermal collector to achieve control over input power. For the same external conditions like temperature of storage tank and input power, experiment is repeated for different flow rates. Collected power and power supplied to the pump are calculated. Effective collector power is plotted against flow rate as shown in figure 2.17. This clearly shows the optimal flow rate at which the collected power is maximum.
Chapter 2. Maximum power point tracking

2.5 Conclusion

In this chapter, the concept of transferring solar energy to the kitchen is proposed. The proposed method is based on the hybrid utilization of solar energy wherein the solar thermal is used to supplement the conventional or auxiliary source of energy and the solar photovoltaic cells that are used to power the pump drives, tracking and monitoring electronics. In order to maximize the energy transferred to the load, this chapter discusses the concept of maximum power point tracking as a means to optimize the flow rate of the thermic fluid. The concept of maximum power point tracking ensures continuous and dynamic control of the thermic fluid flow rate to maximize the energy collected from the sun. An algorithm for MPPT is proposed and implemented. Additionally, the choice of the pipe diameter is also an exercise in solving conflicting constraints viz. heat loss from the pipe surface area and reduction in pumping effort with increase in pipe diameter. This issue is also addressed in this chapter. The concepts of optimization for the proposed system have been verified by simulation and experiment.

The next chapter discusses the modeling of this system using bond graph technique.
Chapter 3

System model

3.1 Introduction

Optimizing the energy absorbed from sun is explained in the previous chapter. This chapter will mainly focus on the mathematical modeling of the proposed cooking system. Non-linear state space equations are obtained to represent the dynamics of the whole system. The method to estimate the system parameters of the physical system will be explained. Based on this, the proposed mathematical model will be experimentally validated. Maximum power point is also validated which was proposed in the previous chapter.

The model of the system is proposed in order to predict its dynamic behavior. This can be used to predict the behavior of the system for different input parameters. A system may comprise of several sub-systems and the energy/power flow in each subsystem may be in a different domains. To understand the behavior of the entire system, it becomes essential that the behavior of all sub-systems is seamlessly integrated. This is done by using the bond graph method that is based on power/energy flow in different domains such as electrical, hydraulic, thermal, mechanical, magnetic etc [80–83]. While modeling, only those non-idealities and non-linearities should be considered which will capture the important effects of the system response.

A solar hybrid cooking system was proposed in chapter 2, where energy from sun is directly transferred to kitchen and supplements conventional source of energy used for cooking. Energy is transferred to the heat storage tank and from this tank to the cooking
load. This chapter presents the bond graph method of modeling the entire system which includes various energy domains like thermal, hydraulic and electrical.

Modeling of the proposed cooking system is a very challenging task since this involves power/energy across different domains. Different techniques have been adapted to model solar cooking systems. Basic types of cookers like box type are modeled using mathematical model based on direct heat balance equations [84]. TRNSYS tool is used to obtain quasi-steady simulation model of most of the solar thermal systems, where ordinary differential and algebraic equations are solved [85, 86]. Simulation toolboxes like TRNSYS, WATSUN etc. have only limited number of subsystems like collector, pump, heat exchanger. It is possible only to use standard type of components (having standard dimension and material) which are available commercially. Modeling becomes more complicated as one includes more non-idealities which are observed in practical systems. There are preliminary bond graph models describing one dimensional fluid flow [87], and thermal energy transport and entropy flow [88]. However, these models do not address the multi-domain energy transport occurring in these systems. Multi-bond graph notation is another way to represent the behavior of energy, power, entropy and other physical properties of multi-domain system in a natural and concise way [89, 90].

A bond graph is a labeled and directed graphical representation of a physical system. The basis of bond graph modeling is power/energy flow in a system. As energy or power flow is the underlying principle for bond graph modeling, there is seamless integration across multiple domains. As a consequence, different domains such as electrical, mechanical, thermal, hydraulic, magnetic, etc. can be represented in a unified way. The power or the energy flow is represented by a half arrow, which is called the power bond or the energy bond as in [80, 81, 91]. The causality for each bond is a significant issue that is inherently addressed in bond graph modeling. As every bond involves two power variables, the decision on setting the cause variable and the effect variable is by natural laws. This has a significant bearing on the resulting state equations of the system. Proper assignment of energy direction resolves the sign-placing problem when connecting sub-model structures. The causality will dictate whether a specific power variable is a cause or the effect. Using causal bars on either end of the power bond, graphically indicate the causality for every bond. Once the causality is assigned, the bond graph displays the structure of the state space equations explicitly. Examples of engineering systems using bond graph modeling elements are discussed in [80, 88, 92-94].

3-2
This chapter proposes a seventh order dynamic model of the solar cooking system with heat storage tank using the bond graph technique. The effects of flow rate and pipe diameter on effective collector efficiency and overall system efficiency are proposed. The proposed model is simulated in MATLAB/Simulink environment. A method to estimate different parameters of the practical solar system is also explained. The simulation results are compared with the experimental results to validate the model.

Section 3.2 explains the experimental setup for the cooking system. Section 3.3 discusses the modeling process and gives the state equations of the system. Section 3.4 presents the method to estimate the parameters of the system. Both simulation and experimental results are presented in section 3.5 and finally concluded in section 3.6.

### 3.2 Solar Cooking System

The block diagram of the cooking setup is shown in figure 3.1. This consists of mainly a solar thermal collector, a storage tank and a heat exchanger. A cylindrical (linear) parabolic collector, a paraboloid or a concentrating collector is used to collect solar energy and increase the temperature of the fluid. The heat exchanger is placed in the kitchen where the cooking is done. Pump-I is used to vary the flow rate of the fluid through the collector. The energy extracted from the sun is stored in the buffer tank. Whenever food has to be cooked, the stored energy is transferred to the load through the heat exchanger using pump-II, which will vary the flow rate of the fluid through the heat exchanger. The auxiliary source of energy like LPG or electrical energy is used for supplementing the stored solar energy and it will as well reduce the time required for cooking.

Energy from sun is absorbed by the collector depending on the insolation. This is utilized to raise the temperature of the fluid that is being circulated from the buffer tank. The average temperature of the fluid in this tank increases gradually as fluid temperature is being risen by the input solar power. When pump-II is operated, hot fluid stored in the buffer tank is circulated through the heat exchanger. Heat gets transferred to the load by fluid loosing its heat. If this energy is not sufficient, the remaining energy for cooking is supplied directly from an auxiliary source.

Figure 3.2 shows the energy flow in the cooking system described above. The solar thermal collector absorbs solar energy, which is used to heat up the fluid. During this
process, part of the energy is lost to the ambient and the remaining energy is stored in the heat storage tank as buffer. Heat transferred from solar energy to the storage tank is dependent on the flow rate of the circulating fluid. This heat transfer rate is controlled using a hydraulic pump. Energy is transferred from the storage tank to the load through the heat exchanger. This flow of energy is controlled by the flow rate of fluid through the heat exchanger. Energy requirement for the load in addition to solar energy is provided by the auxiliary source of energy like LPG or electrical energy.

### 3.3 Modeling and Simulation

The bond graph method is applied in this chapter to model the dynamic system [80, 81, 95, 96]. The proposed cooking system is a complex multi energy domain system comprising power/energy flow across several domains such as thermal, hydraulic [97], electrical and mechanical. Components like energy sources, energy dissipating elements, kinetic and static energy storage elements are connected using bonds, which represents the energy exchange. In bond graph methodology, the various physical variables in the
multi-domain environment are uniformly defined as generalized power variables such as effort \((e)\) and flow \((f)\) as given in table 3.1. The product of these two variables is the power. In the hydraulic domain, pressure is the effort variable and flow rate is the flow variable. Similarly voltage-current and temperature-entropy rate are the effort-flow pairs in the electrical and the thermal domains respectively. At the 0-junction, the flow variable from all the bonds add up to zero and each bond shares the same effort decided by the flow causal bond. Similarly effort in all the bonds add up to zero sharing common flow at the 1-junction.

![Figure 3.3: Block diagram showing bond graph elements](image)

The entire system can be represented by the bond graph models that are shown in figures 3.4 and 3.5. Figure 3.4(a) represents the thermal model of the system. Solar energy incident on the collector is modeled as a flow source, giving entropy rate \(\dot{S}_m\) to the collector through bond 2. Entropy rate is calculated according to the power available at the collector and temperature of the collector.

For experimental purpose, an electric heater is used instead of solar thermal collector. By doing this, input power can be controlled in order to validate the proposed model. Figure 3.4(b) shows the thermal model corresponding to the experimental setup. The input source represents the equivalent solar energy available from the collector to a \(R_1\) field, which transfers energy to the thermal domain [98]. This electrical input acts like
a sun emulator, which gives energy equivalent to energy obtained from the sun. If an electric heater is used to represent the equivalent solar thermal energy collector, then bond 1 is in the electrical domain having voltage $V_{in}$ and current $I_{in}$, where as bond 2 is...
in the thermal domain with collector temperature $T_{coll}$ as effort and entropy rate $\dot{S}_2$ as flow variable. Resistance field, $R_1$ represents the energy transfer between the equivalent electrical source and the thermal domain without any entropy generation.

Energy gets stored in the mass of the collector which acts as a thermal capacitance represented by $C_{coll}$. Temperature of this collector is one of the state variables in the system. Part of the energy is lost to ambient through the field $R_2$, since temperature of the collector is higher than ambient temperature.

Fluid is circulated through the heat transfer tube (collector) at a rate as determined from the hydraulic model of figure 3.5. The modulated effort source $mS_c$ gives the hydraulic pressure difference $P_1$ between the inlet and outlet of fluid through the collector. Another pressure source, $P_2$ represents the pressure head developed by the pump-1. Kinetic storage element $L_{h1}$ decides the flow rate $f_1$ in the collector side. As density of the fluid varies with the change in temperature, pressure $P_1$ changes. Hydraulic resistance $R_{pipe1}$ represents resistance due to friction of pipe. Resistance $R_c$ accounts for the pressure drop in the storage tank, which is common to both collector side and load side hydraulic loops. As in the case of collector side, fluid is circulated through the heat exchanger using pump-2 on the load side. Flow rate $f_3$ is decided by the kinetic storage element $L_{h2}$. Pressure drop across the heat exchanger and the fluid outlet from that are combined together and represented as field $R_{ex}$ and $R_{c2}$. Pressure drop in the pipe on load side is represented by the resistance $R_{pipe2}$.

Referring to figure 3.3, $C_c$ represents the thermal capacitance of the heat storage tank including the fluid in the inlet pipe to the heater. Thermal capacitance of the fluid in the
outlet pipe from the collector is represented as $C_{c1}$. Entropy flow rate through the field $R_3$ accounts for the energy transferred from collector to $C_{c1}$. Outlet pipe fluid is modeled as thermal capacitance as it can store energy by a raise in the temperature. Energy is transferred from $C_{c1}$ to $C_c$ through the field $R_a$. Flow rate $f_1$ as determined from the hydraulic model of figure 3.5, decides the energy transferred from the collector to the buffer. The heat energy is stored in the fluid in the outlet pipe of the collector, $C_{c1}$ and also the storage tank $C$. Part of the entropy flows to ambient through the fields $R_4$ and $R_5$ respectively.

Circulation of fluid on the load side transfers energy to the load through the heat exchanger. $C_{c2}$ represents the thermal capacitance of the fluid in the outlet pipe of the heat exchanger. Entropy flow rate through the field, $R_b$ transfers energy from the heat storage tank to $C_{c2}$. From $C_{c2}$ part of the energy is lost to ambient through the field $R_b$ and remaining is transferred to the load through $R_7$. Whenever this energy is not sufficient for cooking, an auxiliary source of energy is used to supplement the solar energy. The energy from $V_{aux}$, which is fed through the field $R_9$, represents the auxiliary source.

### 3.3.1 Thermal Model

Consider a solar concentrating collector which is used for raising the temperature of a fluid. Input to this collector depends on the solar insolation at the location given by equation (3.1) as explained in section 2.3.1.

$$P_{in} = G_bA_a\eta_o$$

Where $G_b$ is the beam or direct irradiance in $\text{W/m}^2$, $A_a$ is the aperture area of the concentrator in $\text{m}^2$ and $\eta_o$ is the optical efficiency of the reflector. This input solar power is fed to the collector through bond 2 with effort being equal to $T_{coll}$. Entropy flow in the flow source is calculated as,

$$P(2) = P_{in} = G_bA_a\eta_o$$

$$\dot{S}(2) = \frac{P(2)}{e(2)} = \frac{G_bA_a\eta_o}{T_{coll}}$$

3-8
Chapter 3. System model

For the experimental setup, an electrical equivalent input is considered having same power as $P_{in}$ emulating solar collector as shown in figure 3.4(b). For equivalent electric power sources, solar input and auxiliary input, $R_1$ and $R_9$ are non-linear resistive fields, which interfaces electrical and thermal domain. Bond 1 is in electrical domain with voltage as effort and current as flow variable. $R_1$ is the electrical resistance of the heater coil. Effort and flow in bond 1 are given by equation (3.3). Power in bond 1, which is the product of effort and flow variable, is given by,

$$e(1) = V_{in}$$
$$f(1) = I_{in} = \frac{V_{in}}{R_1}$$
$$P(1) = e(1) \times f(1) = \frac{V_{in}^2}{R_1} \tag{3.3}$$

In the thermal domain, the effort is the absolute temperature and the flow is the entropy flow rate $\dot{S}$ in $J/Ks$. Bond 2 is in the thermal domain with effort being equal to $T_{coll}$. Since energy flow in bond 1 and bond 2 are equal, one can get the entropy flow in bond 2 as follows,

$$P(2) = P(1) = \frac{V_{in}^2}{R_1}$$
$$\dot{S}(2) = \frac{P(2)}{e(2)} = \frac{V_{in}^2}{R_1} \frac{T_{coll}}{R_2} \tag{3.4}$$

Field $R_2$ refers to the energy lost to the ambient from the collector. By maintaining that the power is conserved across the bonds of the field, the expressions for the flows are given by equation (3.5). $R_2$ is the thermal resistance to the entropy flow in $K/W$.

$$P(3) = P(4) = \frac{(T_{coll} - T_{amb})}{R_2}$$
$$\dot{S}(3) = \frac{(T_{coll} - T_{amb})}{R_2 \times T_{coll}}$$
$$\dot{S}(4) = \frac{(T_{coll} - T_{amb})}{R_2 \times T_{amb}} \tag{3.5}$$

Entropy flow rate from the collector to the $C_1$ has two parts. First part represents the entropy flow to raise the temperature of $C_1$ through the field $R_3$. Another part is due to the mass transfer, where fluid temperature is increased by circulating it through the collector, which is represented as the field $R_3^*$. The total power flow is given by $P(6) + P(6')$
or \( P(7) + P(7') \) as in equation (3.6). Since energy flow is same in bonds 6 and 7, entropy flow in these are obtained by applying conservation of energy principle.

\[
P(6) = P(7) = \frac{(T_{\text{coll}} - T_{c1})}{R_3}
\]

\[
P'(6) = P'(7') = (T_{c1} - T_e) \times \dot{q}_1 \rho C_p
\]

\[
\dot{S}(6) = \frac{P(6)}{T_{\text{coll}}} \quad \text{and} \quad \dot{S}'(6') = \frac{P'(6')}{T_{\text{coll}}}
\]

\[
\dot{S}(7) = \frac{P(7)}{T_{c1}} \quad \text{and} \quad \dot{S}'(7') = \frac{P'(7')}{T_{c1}}
\]

(3.6)

Where \( R_3 \) is the thermal resistance to the entropy flow from the collector to \( C_1 \) in \( W/K \). \( \rho \) and \( C_p \) are the density and the specific heat of the fluid in SI units \((kg/m^3 \text{ and } J/kgK \text{ respectively})\). Flow rate of fluid on the collector side is represented as \( \dot{q}_1 \) in \( m^3/s \).

Since temperature \( T_{c1} \) is higher than the ambient temperature \( T_{\text{amb}} \), there is an entropy flow to the ambient through the field \( R_4 \). Power and entropy rate in bonds 8 and 9 are given by equation (3.7).

\[
P(8) = P(9) = \frac{(T_{c1} - T_{\text{amb}})}{R_4}
\]

\[
\dot{S}(8) = \frac{(T_{c1} - T_{\text{amb}})}{R_4 \times T_{c1}}
\]

\[
\dot{S}(9) = \frac{(T_{c1} - T_{\text{amb}})}{R_4 \times T_{\text{amb}}}
\]

(3.7)

From the heat storage tank, fluid is taken out at temperature \( T_e \) and takes thermal energy from the thermal capacitance \( C_1 \) of the fluid mass in the inlet pipe. This raises fluid temperature to \( T_{c1} \). Since it is circulated at a rate \( \dot{q}_1 \), the energy put into the storage tank from the heat source is given by \((T_{c1} - T_e) \times \dot{q}_1 \rho C_p\). Equation (3.8) gives the entropy rate in bonds 11 and 12. This thermal resistance due to mass transfer is represented as \( R_a \).

\[
P(11) = P(12) = (T_{c1} - T_e) \times \dot{q}_1 \rho C_p
\]

\[
\dot{S}(11) = \frac{(T_{c1} - T_e) \times \dot{q}_1 \rho C_p}{T_{c1}}
\]

\[
\dot{S}(12) = \frac{(T_{c1} - T_e) \times \dot{q}_1 \rho C_p}{T_e}
\]

\[
R_a = \frac{1}{\dot{q}_1 \rho C_p}
\]

(3.8)
Chapter 3. System model

Similar to equations (3.5) and (3.7), one can write entropy flow rate equations from the storage tank to ambient through the field $R_5$. Entropy flow rate depends on the type of the insulation provided for the tank. Better the thermal insulation, lesser is the entropy flow rate to the ambient.

$$P(13) = P(14) = \frac{(T_c - T_{amb})}{R_5}$$

$$\dot{S}(13) = \frac{(T_c - T_{amb})}{R_5 \times T_c}$$

$$\dot{S}(14) = \frac{(T_c - T_{amb})}{R_5 \times T_{amb}}$$

On the load side, hot fluid is circulated from the storage tank through the heat exchanger at a flow rate of $\dot{q}_2$. Energy is extracted from the hot fluid and utilized to heat up the load. Entropy flows from the storage tank to the fluid in outlet pipe of the heat exchanger by mass transfer mechanism. $R_b$ represents the thermal resistance between the tank and $C_2$. Equation (3.10) gives the power and the entropy flow rates in bonds 16 and 17.

$$P(16) = P(17) = (T_c - T_{c2}) \times \dot{q}_2 \rho C_p$$

$$\dot{S}(16) = \frac{(T_c - T_{c2}) \times \dot{q}_2 \rho C_p}{T_c}$$

$$\dot{S}(17) = \frac{(T_c - T_{c2}) \times \dot{q}_2 \rho C_p}{T_{c2}}$$

$$R_b = \frac{1}{\dot{q}_2 \rho C_p}$$

From $C_2$, part of the energy is lost to ambient through the field $R_6$ and the remaining is transferred to the load through the field $R_7$. Equation (3.11) gives the entropy flow rate to the ambient from $C_2$, which is similar to the entropy flow rate to ambient from other thermal capacitances.

$$P(18) = P(19) = \frac{(T_{c2} - T_{amb})}{R_6}$$

$$\dot{S}(18) = \frac{(T_{c2} - T_{amb})}{R_6 \times T_{c2}}$$

$$\dot{S}(19) = \frac{(T_{c2} - T_{amb})}{R_6 \times T_{amb}}$$

Entropy flow rate from the hot fluid in $C_2$ to the load is through the field $R_7$. $R_7$ is the thermal resistance of the heat exchanger in $W/K$. The value of $R_7$ decides the entropy flow.
flow rate for a given temperature difference.

\[ P(21) = P(22) = \frac{(T_{c2} - T_{\text{Load}})}{R_7} \]

\[ \dot{S}(21) = \frac{(T_{c2} - T_{\text{Load}})}{R_7 \times T_{c2}} \]

\[ \dot{S}(22) = \frac{(T_{c2} - T_{\text{Load}})}{R_7 \times T_{\text{Load}}} \quad (3.12) \]

The load operates through three energy ports. Through one of the energy ports (bond 22), it gets entropy through the field \( R_7 \). It loses entropy by virtue of the load temperature being higher than ambient, wherein the entropy flows to the ambient through the field \( R_8 \) as given in equation (3.13).

\[ P(23) = P(24) = \frac{(T_{\text{Load}} - T_{\text{amb}})}{R_8} \]

\[ \dot{S}(23) = \frac{(T_{\text{Load}} - T_{\text{amb}})}{R_8 \times T_{\text{amb}}} \]

\[ \dot{S}(24) = \frac{(T_{\text{Load}} - T_{\text{amb}})}{R_8 \times T_{\text{amb}}} \quad (3.13) \]

The load can receive entropy through the third port from an auxiliary energy source. It gets added to the load entropy through the field \( R_9 \). Without loss of generality, one can consider the electrical energy equivalent of energy supplied by the auxiliary source, that can be either LPG or electrical. Electrical resistance of the auxiliary heater is represented as \( R_9 \). Bond 27 is in the electrical domain with \( V_{\text{aux}} \) as the effort variable and \( I_{\text{aux}} \) as the flow variable, where as bond 26 is in the thermal domain with \( T_{\text{Load}} \) as the effort variable and \( \dot{S}(26) \) as the flow variable. Maintaining conservation of energy across the field, one obtains the entropy flow rate in bonds 26 and 27.

\[ P(26) = P(27) = \frac{V_{\text{aux}}^2}{R_9} \]

\[ \dot{S}(26) = \frac{P(26)}{e(26)} = \frac{V_{\text{aux}}^2}{R_9} = \frac{V_{\text{aux}}^2}{T_{\text{Load}}} \quad (3.14) \]

### 3.3.2 Thermal Capacitances

There are five thermal capacitances in the thermal model discussed in the previous section, such as \( C_{\text{coll}}, C_{c1}, C_c, C_{c2} \) and \( C_{\text{Load}} \). Modeling and corresponding constitutive
relations of these capacitances are explained in this section. These capacitances decide the temperature of five different system components that determine the states of the system.

The constitutive relation for the thermal capacitance is given as,

\[ e f f o r t = \frac{1}{C} \int T h e r m a l \ P o w e r \cdot dt \]
\[ T = \frac{1}{C} \int \dot{S} \cdot T \cdot dt \]  

(3.15)

\[ C_{c o l l} \] represents the thermal capacitance of the collector, whose temperature raises due to the thermal power absorbed from the solar radiation. Figure 3.6 shows the capacitor model of the collector. Temperature of the collector is obtained from the equation (3.16), where \( \dot{S}(5) \) is the entropy flow rate into the thermal capacitance at the collector temperature.

\[ \frac{d}{dt} (T_{c o l l}) = \frac{\dot{S}(5) \cdot T_{c o l l}}{C_{c o l l}} \]
\[ T_{c o l l} = \frac{1}{C_{c o l l}} \int \dot{S}(5) \cdot T_{c o l l} \cdot dt \]  

(3.16)

Thermal capacitance of the fluid in the outlet pipe of the collector is represented as

\[ C_{c 1}, \] which indicates the amount of thermal power that can be stored for a given raise in temperature, \( T_{c 1} \). This is modeled similar to that of \( C_{c o l l} \) except for the capacitance value which is calculated differently. Capacitance value of the fluid depends on the density \( \rho \), volume \( Q \) and the specific heat \( C_p \) of the fluid.

\[ C_{c 1} = Q_{c 1} \rho C_p \]  

(3.17)

Where \( Q_{c 1} \) is the volume of the fluid outlet from the heat transfer pipe in \( m^3 \). \( \rho \) and \( C_p \) have the unit of \( kg/m^3 \) and \( J/kgK \) respectively. The model of the thermal capacitance is shown in figure 3.7.

For more accurate modeling, variation in the density and specific heat for temperature variation are considered. The model of the thermal capacitor which includes density
and specific heat variation is as shown in figure 3.8. Temperature of the fluid is used to calculate the density and the specific heat using a model or a look-up table having \( \rho \) Vs temperature and \( C_p \) Vs temperature characteristics. The expression for the thermal capacitance \( C_{c1} \) is given as,

\[
\frac{d}{dt} (T_{c1}) = \frac{\dot{S}(10) \cdot T_{c1}}{Q_{c1} \rho C_p}
\]

\( T_{c1} = \frac{1}{Q_{c1} \rho C_p} \int \dot{S}(10) \cdot T_{c1} \cdot dt \)  

(3.18)

\( C_c \) represents the thermal capacitance of the heat storage tank which stores energy by raising the fluid temperature, \( T_c \). The model for the thermal capacitance is given by the equation (3.19), which gives the expression for the storage tank temperature \( T_c \). \( Q_c \) represents the volume of the storage tank in \( m^3 \) and \( \dot{S}(15) \) represents the entropy flow rate in the bond 15 of figure 3.4.

\[
\frac{d}{dt} (T_c) = \frac{\dot{S}(15) \cdot T_c}{Q_c \rho C_p}
\]

\( T_c = \frac{1}{Q_c \rho C_p} \int \dot{S}(15) \cdot T_c \cdot dt \)  

(3.19)

The thermal capacitance of the fluid outlet from the heat exchanger is represented as \( C_{c2} \), which is modeled similar to \( C_{c1} \) and \( C_c \). Equation (3.20) gives the corresponding model,
where $Q_{c2}$ is the volume of the outlet fluid.

\[
\frac{d}{dt}(T_{c2}) = \frac{\dot{S}(20) \cdot T_{c2}}{Q_{c2} \rho C_p}
\]

\[
T_{c2} = \frac{1}{Q_{c2} \rho C_p} \int \dot{S}(20) \cdot T_{c2} \cdot dt
\] (3.20)

Equation (3.21) gives expression for the load temperature $T_{\text{Load}}$. The thermal capacitance associated with the load is considered as $Q_L \rho_L C_{pL}$, where density and specific heat of the load is calculated at the load temperature.

\[
\frac{d}{dt}(T_{\text{Load}}) = \frac{\dot{S}(25) \cdot T_{\text{Load}}}{Q_L \rho_L C_{pL}}
\]

\[
T_{\text{Load}} = \frac{1}{Q_L \rho_L C_{pL}} \int \dot{S}(25) \cdot T_{\text{Load}} \cdot dt
\] (3.21)

### 3.3.3 Hydraulic Domain Model

Figure 3.5 shows the bond graph modeling of the hydraulic domain part of the system. Pump-1 is used to circulate fluid in the collector. The pressure drop due to gravity, variation in density and pipe friction are modeled. Pressure across the heat storage tank when the fluid is flowing from top to bottom is given by the equation (3.22), which is represented as a field $R_c$.

\[
p(32) = -\rho \ g \ H_c
\] (3.22)

Where $\rho$ is the density of the fluid calculated at the temperature $T_c$, $g$ is the earth’s gravitational constant in $m/s^2$ and $H_c$ is the height difference between inlet and outlet of the tank.

Referring to figure 3.5, pressure $P_2$ gives the pressure head produced by pump-1 for a flow rate of $\dot{q}_1$. Hydraulic pressure across the collector and outlet from it are combined together and modeled as a modulating pressure source. In figure 3.5, the expression for $P_1$ is given in equation (3.23). Since the fluid is circulated against the gravitational force, the modulated effort source giving pressure $P_1$ is considered as a sink and is indicated with a negative sign.

\[
P_1 = -\rho \ g \ H_{\text{coll}} - \rho \ g \ H_{c1}
\]

\[
P_1 = -\rho \ g \ (\Delta H)
\] (3.23)
Chapter 3. System model

$H_{coll}$ is the height difference between the inlet and outlet of the collector and $H_{c1}$ is the height of $C_1$ from reference. Addition of these two height differences gives $\Delta H$, which is the total height difference between the inlet and the outlet of the storage tank. The density $\rho$ is calculated for the corresponding temperatures using a look up table.

Similarly, pressure drop across the heat exchanger and the outlet fluid, are combined together and modeled as a field $R_{ex} + R_{c2}$. Pressure $P_7$ gives the pressure across this field as in equation (3.24). The height difference between the inlet and the outlet of the heat exchanger and $C_2$ are considered as $H_{ex}$ and $H_{c2}$ respectively. The pump used for the circulation of fluid gives the pressure head of $P_6$ for a given flow rate of $\dot{q}_2$.

\[ P_7 = -\rho \ g \ H_{ex} - \rho \ g \ H_{c2} \] (3.24)

Since fluid circulation is laminar, pressure drop in the pipes are calculated and modeled according to equation (3.25) using Darcy-Weisbach formula [77].

\[ P_{pipe} = R_{pipe} \times \dot{q} \]
\[ R_{pipe} = \frac{64 \ \nu \ l \ \rho}{2 \ \pi \ \frac{d^4}{4}} \] (3.25)

Where $\nu$ is the kinematic viscosity of the fluid in $m^2/s$, $l$ is the length of the pipe in $m$, $\dot{q}$ is the volume flow rate in $m^3/s$ and $d$ is the diameter of the pipe in $m$.

The fluid mass circulating at some flow rate is modeled as a kinetic storage element. On the collector side, it is modeled as $L_{h1}$ and expression for the flow rate is obtained as given in equation (3.26). $P_{pipe1}$ and $P_{pipe2}$ represent the pressure drop in the pipe on the collector and the load side circulations respectively.

\[ f(29) = \dot{q}_1 = \frac{1}{L_{h1}} \int (P_1 + P_2 - p(31) - P_{pipe1}) \ dt \] (3.26)

Similarly for the load side circulation of fluid, equation (3.27) gives the expression for the flow rate $\dot{q}_2$.

\[ f(34) = \dot{q}_2 = \frac{1}{L_{h2}} \int (p(33) + P_6 - P_7 - P_{pipe2}) \ dt \] (3.27)
3.3.4 State Equations

From the bond graph model shown in the figures 3.4 and 3.5, the state equations can be obtained by applying the junction laws [80, 81, 95]. There are five thermal capacitances in the system, namely $C_{coll}$, $C_{c1}$, $C_{c2}$ and $C_{Load}$ in the thermal domain model and two hydraulic inertial components in the hydraulic domain model, $L_{h1}$ and $L_{h2}$. Therefore, overall the system is a 7th order system.

According to the 0-junction law, flow variables from all the bonds algebraically add up to zero having the same effort value. Applying this rule to junction A of figure 3.4, one arrives at,

$$\dot{S}(5) = \dot{S}(2) - \dot{S}(3) - \dot{S}(6) - \dot{S}(6')$$ \hfill (3.28)

Substituting for $\dot{S}(2)$, $\dot{S}(3)$, $\dot{S}(6)$ and $\dot{S}(6')$ from equations (3.4), (3.5) and (3.6) respectively, applying the governing equation for the thermal capacitance of the collector to obtain $\dot{S}(5)$ from equation (3.16), the following state equation is obtained.

$$\frac{d}{dt} (T_{coll}) = \frac{1}{C_{coll}} \left[ \frac{V_{in}^2}{R_1} - \frac{T_{coll} - T_{amb}}{R_2} - \frac{T_{coll} - T_{c1}}{R_3} - (T_{c1} - T_{c})q_1 \rho C_p \right]$$ \hfill (3.29)

Applying 0-junction law for the junction B of figure 3.4, one obtains,

$$\dot{S}(10) = \dot{S}(7) + \dot{S}(7') - \dot{S}(8) - \dot{S}(11)$$ \hfill (3.30)

Substituting for $\dot{S}(7')$, $\dot{S}(8)$ and $\dot{S}(11)$ into equation (3.18) from equations (3.6), (3.7) and (3.8), one obtains,

$$\frac{C_{c1}}{T_{c1}} \frac{d}{dt} (T_{c1}) = \frac{T_{coll} - T_{c1}}{R_3 T_{c1}} + \frac{(T_{c1} - T_{c}) q_1 \rho C_p}{T_{c1}} - \frac{T_{c1} - T_{amb}}{R_4 T_{c1}} - \frac{(T_{c1} - T_{c}) q_1 \rho C_p}{T_{c1}}$$ \hfill (3.31)

$$\frac{d}{dt} (T_{c1}) = \frac{1}{C_{c1}} \left[ \frac{T_{coll} - T_{c1}}{R_3} - \frac{T_{c1} - T_{amb}}{R_4} \right]$$ \hfill (3.32)

Applying the 0-junction law to junction C of figure 3.4, the following equation is obtained.

$$\dot{S}(15) = \dot{S}(12) - \dot{S}(13) - \dot{S}(16)$$ \hfill (3.33)
Chapter 3. System model

Substituting for $\dot{S}(12)$, $\dot{S}(13)$ and $\dot{S}(16)$ from equations (3.8), (3.9) and (3.10) respectively, and applying the governing equation for the thermal capacitance of the heat storage tank to obtain $\dot{S}(15)$ from equation (3.19), the following state equation is obtained.

$$\frac{d}{dt} (T_c) = \frac{1}{C_c} \left[ (T_{c1} - T_c) \dot{q}_1 \rho C_p - (T_c - T_{c2}) \dot{q}_2 \rho C_p - \frac{T_c - T_{amb}}{R_5} \right]$$  (3.34)

Considering the junction D, and applying the 0-junction law, the following equation is obtained.

$$\dot{S}(20) = \dot{S}(17) - \dot{S}(18) - \dot{S}(21)$$  (3.35)

Substituting for $\dot{S}(20)$, $\dot{S}(17)$, $\dot{S}(18)$ and $\dot{S}(21)$ respectively, the state equation for $T_{c2}$ is obtained from equations (3.20), (3.10), (3.11) and (3.12),

$$\frac{d}{dt} (T_{c2}) = \frac{1}{C_{c2}} \left[ (T_c - T_{c2}) \dot{q}_2 \rho C_p - \frac{T_{c2} - T_{amb}}{R_6} - \frac{T_{c2} - T_{Load}}{R_7} \right]$$  (3.36)

Consider the junction E. Similar to other 0-junctions, one obtains the expression as given in equation (3.37).

$$\dot{S}(25) = \dot{S}(22) + \dot{S}(26) - \dot{S}(23)$$  (3.37)

Substituting for $\dot{S}(25)$, $\dot{S}(22)$, $\dot{S}(23)$ and $\dot{S}(26)$ from the equations (3.21), (3.12), (3.13) and (3.14) respectively, one obtains the state equation as follows,

$$\frac{d}{dt} (T_{Load}) = \frac{1}{C_{Load}} \left[ \frac{T_{c2} - T_{Load}}{R_7} + \frac{V_{aux}^2}{R_9} - \frac{T_{Load} - T_{amb}}{R_8} \right]$$  (3.38)

In the hydraulic domain model shown in figure 3.5, there are two kinetic energy storing elements namely $L_{k1}$ and $L_{k2}$. At junction G, according to the 0-junction law, all the bonds should share the same effort. From equation (3.22), one obtains the following expression for $p(31)$ and $p(33)$.

$$p(31) = p(33) = p(32) = - \rho g H_c$$  (3.39)

At 1-junction, the pressures (effort variable) from all the bonds algebraically add up to zero and each bond shares the same flow rate (flow variable) decided by the effort causal bond.
Chapter 3. System model

Consider the junction F. Applying the 1-junction law to this, one obtains the expression as follows.

\[ p(29) = p(28) + p(30) - p(31) - p(37) \]
\[ P_3 = P_1 + P_2 - p(31) - P_{\text{pipe}1} \]  \hspace{1cm} (3.40)

Substituting for \( P_3 \), \( P_1 \) and \( p(31) \) from equations (3.26), (3.23) and (3.39) respectively, one obtains,

\[ P_3 = -\rho g H_{\text{coll}} - \rho g H_{c1} + P_{\text{pump}1} + \rho g H_{c} - P_{\text{pipe}1} \]
\[ \dot{q}_1 = \frac{1}{L_{h1}} \int P_3 \, dt \]  \hspace{1cm} (3.41)

Similarly consider the junction H. By applying the 1-junction law, one obtains

\[ p(34) = p(33) + p(35) - p(36) - p(38) \]
\[ P_5 = p(33) + P_6 - P_7 - P_{\text{pipe}2} \]  \hspace{1cm} (3.42)

Substituting for \( P_5 \), \( p(33) \) and \( P_7 \) from equations (3.27), (3.39) and (3.24) respectively, one obtains,

\[ P_5 = -\rho g H_{c} + P_{\text{pump}2} + \rho g H_{ex} + \rho g H_{c2} - P_{\text{pipe}2} \]
\[ \dot{q}_2 = \frac{1}{L_{h2}} \int P_5 \, dt \]  \hspace{1cm} (3.43)

Flow rates \( \dot{q}_1 \) and \( \dot{q}_2 \) are obtained from the hydraulic domain of the bond graph model, which are dependent on the temperatures \( T_{c1} \), \( T_c \), \( T_{c2} \) and the density of fluid at that temperature. Since density of the fluid is a non-linear function of temperature, it is not possible to represent in the \( \dot{x} = Ax + Bu \) form. Pressure drop in pipe, \( P_{\text{pipe}} \) is a function of flow rate, diameter and length of the pipe. Equations (3.29), (3.32), (3.34), (3.36), (3.38), (3.41) and (3.43) forms the state equation model of the system.

\[ \dot{x} = Ax + Bu \]  \hspace{1cm} (3.44)
The thermal system can be represented in the general state-space form as in equation (3.45). In equation (3.45), $\alpha_1$ and $\alpha_2$ represents the heat mass transfer given by $\dot{q}_1 \rho C_p$ and $\dot{q}_2 \rho C_p$, respectively. Equations (3.45), (3.41) and (3.43) together form the overall state space model of the system.

### 3.4 Parameter Estimation

This section explains the method to estimate the parameters from a practical solar cooking setup. The experimental setup is as shown in figure 3.9.

The electrical heater is used to emulate the solar collector, as one can control the energy input to the system without any dependency on solar insolation for experimental
Chapter 3. System model

purposes. Both the pumps are driven by variable voltage sources through pulse width
modulation based controller. The flow rates can be varied by controlling the applied
voltage to the pumps.

Flow rates $\dot{q}_1$ and $\dot{q}_2$ are measured during the experiment by using flow meters on the
collector and the load side circulation. Thermocouple based temperature sensors are in-
serted at different points in order to measure the temperatures $T_{c1}$, $T_c$, $T_{c2}$, $T_{Load}$ and
$T_{amb}$. The temperature of collector $T_{coll}$ is estimated from the state equations.

In the bond graph model shown in figure 3.4, there are two electrical resistances and seven
thermal resistances. Since dynamic response of the hydraulic system is faster compared
to dynamic response of the thermal system, one can neglect the dynamics of hydraulic
domain without loss of generality while estimating the thermal parameters.

Electrical resistance of the heater coil $R_1$ can be calculated directly by applying a known
voltage and measuring the current drawn by it. Similarly $R_9$, electrical resistance of the
auxiliary source of energy can be calculated experimentally. For experimental purpose,
electrical coil is used as the thermal source of energy. The electric coil is placed inside the
thermic fluid filled cylinder, which is surrounded by thermal insulation. As a consequence
$R_2$ is a very large value. Hence entropy flow rate from coil to atmosphere can be neglected
and $R_2$ is considered as infinity.

\[
\frac{V_{in}^2}{R_1} - \frac{T_{coll} - T_{c1}}{R_3} - \frac{(T_{c1} - T_c)\dot{q}_1\rho C_p}{R_3} = 0
\]

\[
\frac{T_{coll} - T_{c1}}{R_3} - \frac{T_{c1} - T_{amb}}{R_4} = 0
\]

\[
(T_c - T_{c1})\dot{q}_1\rho C_p - \frac{(T_c - T_{c2})\dot{q}_2\rho C_p - T_c - T_{amb}}{R_5} = 0
\]

\[
(T_c - T_{c2})\dot{q}_2\rho C_p - \frac{T_{c2} - T_{amb}}{R_6} - \frac{T_{c2} - T_{Load}}{R_7} = 0
\]

\[
\frac{T_{c2} - T_{Load}}{R_7} + \frac{V_{aux}^2}{R_9} - \frac{T_{Load} - T_{amb}}{R_8} = 0
\]

A specific voltage $V_{in}$ is applied to the heater coil and flow rates on both the sides are
set by varying the pressure head of the pumps. The pump pressure heads are varied by
controlling the applied voltage through pulse width modulation (PWM). The system is
allowed to reach steady state where temperatures at different points settle to the steady
state values. These values are noted along with the flow rates $\dot{q}_1$ and $\dot{q}_2$. Under steady
state, all the derivatives of the states become zero and one obtains five expressions as
given in equations (3.46) to (3.50).

It is now required to estimate $T_{coll}$, $R_3-R_8$ from the above five non-linear equations. For each experimental set of values, one gets five sets of non-linear equations. Figure 3.10 shows the block diagram of the parameter estimator used for estimating the unknowns from the five non-linear equations. $f(x)$ represents the non-linear equations given by (3.46) to (3.50), where $x$ is the vector containing the unknowns. The values of unknowns, $x$ are varied to minimize $f(x)$. Error value $e$ is multiplied by a matrix $M$, which tries to tune the parameter value $x$ according to $e$. Unknowns $T_{coll}$, $R_3-R_8$ are found using this technique.

![Block diagram of parameter estimator](image)

**Figure 3.10: Block diagram of parameter estimator**

The unknowns can also be found using non-linear least square technique that minimizes the error between estimated values and the experimental values. Newton’s trust region reflective [99] method is used for solving these equations in Matlab [100].

Thermal capacitance values are calculated from the physical dimensions and properties of the experimental setup. For the heater coil, the thermal capacitance is given by the equation (3.51), where $C_v$ is the volumetric heat capacity of copper which is $3.45 \text{ MJ/m}^3\text{K}$ and $Q_{coll}$ is the volume of the collector in m$^3$.

$$C_{coll} = Q_{coll} \times C_v \quad (3.51)$$

The thermal capacitance of the fluid is calculated using equation (3.17). The volume of different thermal capacitor elements like $Q_c$, $Q_{c1}$, $Q_{c2}$ and $Q_L$ are measured. Using the density and specific heat properties of the fluid, the thermal capacitance values, $C_c$, $C_{c1}$,
Chapter 3. System model

$C_{c2}$ and $C_{Load}$, can be estimated as given in equation (3.52).

\[ C_c = Q_c \rho \, C_p \]
\[ C_{c1} = Q_{c1} \rho \, C_p \]
\[ C_{c2} = Q_{c2} \rho \, C_p \]
\[ C_{Load} = Q_L \rho_L \, C_{pL} \]

(3.52)

3.5 Results

For the setup shown in figure 3.9, the experiment is carried out for different input voltages $V_{in}$ representing different atmospheric conditions. The system is allowed to reach steady state and the temperatures at different points are measured and tabulated as in table 3.2. Flow rates are tabulated in liters per minute (LPM).

Parameters are estimated as explained in section 3.4. Using non-linear least square technique, the system parameters are obtained as, $R_3=3.5386$, $R_4=0.4747$, $R_5=0.5016$, $R_6=0.2925$, $R_7=0.23$ and $R_8=0.1805$.

These estimated parameters are hugged-into the bond graph simulated model. The bond graph model for the system is shown in figures 3.4 and 3.5. The system has been modeled and simulated in Matlab-Simulink [101] as shown in figure 3.11. With the similar input values like $V_{in}$, $q_1$, $q_2$ and $T_a$, the model is simulated until steady state is reached. Temperature reached by each state variable is compared with the experimental values. Percentage error between simulated values and experimentally obtained values are computed and tabulated in table 3.4. On an average the estimate errors are lesser than 5%.

The thermic fluid is heated up to a specific temperature using the electric heater.

<table>
<thead>
<tr>
<th>Sl.</th>
<th>$V_{in}$ V</th>
<th>$P_{in}$ W</th>
<th>$\dot{q}_1$ LPM</th>
<th>$\dot{q}_2$ LPM</th>
<th>$T_a$ C</th>
<th>$T_{c1}$ C</th>
<th>$T_c$ C</th>
<th>$T_{c2}$ C</th>
<th>$T_{Load}$ C</th>
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</thead>
<tbody>
<tr>
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<td>100</td>
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<td>0.7965</td>
<td>28.5</td>
<td>40.58</td>
<td>39.78</td>
<td>38.28</td>
<td>32.58</td>
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<td>156</td>
<td>1.4859</td>
<td>0.6383</td>
<td>28.2</td>
<td>44.58</td>
<td>43.43</td>
<td>40.78</td>
<td>31.78</td>
</tr>
<tr>
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<td>0.6293</td>
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<td>42.15</td>
<td>41.1</td>
<td>34.3</td>
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<tr>
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<td>0.5289</td>
<td>26.3</td>
<td>44.7</td>
<td>43.65</td>
<td>41.7</td>
<td>33.4</td>
</tr>
</tbody>
</table>

Table 3.2: Experimental results
Chapter 3. System model

<table>
<thead>
<tr>
<th>Sl.</th>
<th>Experimental Result</th>
<th>Simulation Result</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$T_{c1}$</td>
<td>$T_c$</td>
</tr>
<tr>
<td>1</td>
<td>40.6</td>
<td>39.8</td>
</tr>
<tr>
<td>2</td>
<td>44.6</td>
<td>43.4</td>
</tr>
<tr>
<td>3</td>
<td>43.3</td>
<td>42.1</td>
</tr>
<tr>
<td>4</td>
<td>44.7</td>
<td>43.7</td>
</tr>
</tbody>
</table>

Table 3.3: Simulation and Experimental results

<table>
<thead>
<tr>
<th>Sl.</th>
<th>$T_{c1}$</th>
<th>$T_c$</th>
<th>$T_{c2}$</th>
<th>$T_{Load}$</th>
</tr>
</thead>
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<td>1.5779</td>
<td>1.8685</td>
<td>2.7719</td>
<td>2.5350</td>
</tr>
</tbody>
</table>

Table 3.4: Error percentage between simulation and experimental results

Energy input to the system is stopped at $t=0$. Temperatures at different points are measured at regular intervals. Similar conditions are applied to the bond graph simulation also. Figure 3.12 shows both the simulation and the experimental results.

Simulation is carried out for different flow rates on the collector side keeping the input power at a constant value. Effective collected power is the power collected from the solar collector excluding the power spent for circulation of fluid. Ratio of this energy to the total input energy gives the effective collector efficiency as in equation (3.53) as explained in section 2.3.1. $P_{\text{pump1}}$ represents the power drawn from the pump-1 which is the product of voltage applied and the current drawn from it. Product of $V_{in}$ and $I_{in}$ gives the input power to the system. Overall effective efficiency of the system is calculated as in equation (3.54).

\[
\eta_{coll} = \frac{\left( T_{c1} - T_c \right) \rho C_p \dot{q}_1 - P_{\text{pump1}}}{V_{in}I_{in}}
\]

\[
\eta_{\text{Total}} = \frac{P_{\text{Load}} - P_{\text{pump1}} - P_{\text{pump2}}}{V_{in}I_{in}}
\]

Effective collector efficiency $\eta_{coll}$ is calculated for different flow rates by varying the input voltage to the pump-1 and plotted as shown in figure 3.13. The voltage applied to the centrifugal pump is initially zero, which implies that the pressure in the pipes correspond to the thermosyphon pressure. As the flow rate increases, heat loss to the ambient is reduced improving efficiency of the collector. But above some flow rate, one needs to spend more energy for the circulation against pipe friction pressure drops. Simulation is carried out for different diameters of the pipe. At optimal flow rate, there is about 6%
Figure 3.11: Simulink model of the system
increase in efficiency as compared to thermosyphon flow rate. As diameter of the pipe is decreased, the efficiency curve moves up. This is due to the decrease in the pipe exposure area which presents a higher thermal resistance for the entropy flow to the ambient. Figure 3.15 shows the simulation result of variation in temperature at different points.

Figure 3.12: Comparison of simulation result with experimental result

Figure 3.13: Variation in effective collector efficiency with flow rate for different pipe diameters
Chapter 3. System model

3.6 Conclusion

This chapter presents the detailed analysis and modeling of hybrid solar cooking system. The proposed system is modeled by primarily using the method of bond graphs which is a multi-energy domain modeling approach. The system comprises of energy interactions.
in the hydraulic domain which is interlinked with the energy interactions in the thermal domain. The thermal domain further complicates the issue as it includes non-linearities. The bond graph model inherently handles all these energy transactions. The energy storage and energy loss based on the entropy flow rate are modeled as field elements. Without loss of generality, all elements including the field elements always maintain the fundamental energy conservation principle. The thermal domain is modeled primarily by using R-fields and C-fields. The dynamic model is a non-linear model and it includes the dynamics of both the hydraulic and thermal domains.

This developed model of the proposed system is simulated and validated through experimentation. The effects of the flow rate and pipe diameter on the solar power collected and the utilized power are discussed. These results are shown by simulating the bond graph model. The method to estimate the various parameters of the practical system is explained. The simulation using these estimated parameters closely match with the experimental results thereby validating the proposed bond graph model. Thus the proposed mathematical model can be used for both analysis and synthesis of a hybrid solar cooking system with different specifications.
Chapter 4

Flow measurement

4.1 Introduction

Optimization of energy obtained from the sun and modeling of the cooking system have been explained in previous two chapters. In order to optimize the energy absorbed by the solar thermal collector, the mass flow rate of the fluid that is being circulated need to be measured as explained in section 2.3.3. The mass flow rate at which energy transported is maximum, can be very low of the order of thermosyphon flow rate. In order to track the maximum power at any instant, the power absorbed from the collector has to be estimated accurately. Hence, it is very essential to have a flow measuring technique, which can accurately measure very low mass flow rate of the fluid.

This chapter presents a novel method of fluid flow measurement where even very low flow rate can be measured accurately without disturbing the system. This is a null-deflection type of measurement. As insertion of such a measuring unit does not affect the functioning of the systems, this can also be classified as a non-disruptive flow measurement method.

Mass flow rate is defined as the quantity of fluid mass that flows through a given cross sectional area per second. Accurate mass flow rate measurement is also essential in many industrial processes such as recipe formulations, material balance determinations and custody transfer operations. The fluid volume is a function of ambient temperature and pressure, while the mass is unaffected by changes in temperature or pressure. Therefore,
mass flow measurement is a more reliable measure of flow in high accuracy systems compared to volume flow measurement.

Several types of flow meters have been developed, tested and used over the years [102, 103]. These can be classified into two major types namely, disruptive and non-disruptive. The disruptive type of flow meter causes change in the operating point, whereas the operating point is not disrupted by the inclusion of the non-disruptive type of flow meter. The disruptive types generally use the differential pressure as a measure of mass flow rate. Differential pressure flow meters are based on the Bernoulli principle, which describes the relation between pressure and velocity of flow. Venturi-type and wedge-type flow meters have proved to be effective in applications where a significant pressure drop can be detected. However, such flow meters require a minimum fluid velocity (Reynolds number more than 75,000) to be able to function properly and further they have a non-linear scale. Other common flow sensing techniques are based on differential pressure data [104], micro-mechanical flow sensor for low flow rate [105], momentum forces or Coriolis forces [106, 107], Pitot tubes [108], vortex-shedding meter [109, 110], turbine flow meters and strain gauges [111]. Others have tested and used piezo-resistive sensors [112], a combination of differential thermal (convection cooling) [113] and differential pressure approaches like elbow flow meters [114], and a vortex flow sensing technique [115]. Though variable area flow meters like rotameter, movable vane meter can measure very low flow rate at a constant pressure drop, it can not be used in case of high viscous fluid. These meters impede the flow of fluid, causing a significant pressure drop across the measuring equipment. These techniques cannot be used for very low flow measurement. Even traditional macro scale flow sensing techniques and equipment are not suitable at low flow rates (Reynolds Number less than 100) because of the low signal-to-noise ratio (SNR) values that they generate.

Non-disruptive type of flow meters does not disturb the operating point in the system in which the flow rate is to be measured. There are a number of non-disruptive methods existing like the ultrasonic flow meter, ultrasonic pulse-Doppler flow meter [116], magnetic flow meter, non-invasive calorimetric method etc. An ultrasonic flow meter has many advantages, such as low pressure loss and wide measuring range [117], but requires high initial cost for setting up and cannot be used for clean liquids. Ultrasonic flow meter also requires additional multi path meter for higher accuracy. Though magnetic flow meter has the minimum obstruction to the flow path and high linearity, it can only measure flow rate of conductive fluids. Other direct flow measuring techniques that rely on electrical
or electromagnetic signals can only be used in certain applications [118] depending on the physical and chemical properties of the fluid.

Therefore, there is a need to develop a reliable and accurate flow measuring technique and instrumentation that can be used in a variety of micro scale flow applications such that it is non-disruptive and inexpensive. This chapter proposes a null-deflection flow meter, where in the pressure drop across the flow resistance is compensated by a compensating pump.

The basic working principle is explained in section 4.2. Modeling and simulation using bond graph technique is discussed in section 4.3. A discussion on the design aspects of such a unit is explained in section 4.4. In section 4.5, the practical implementation of the proposed system is discussed. Effect of pressure transducer is described in section 4.6 and section 4.7 explains about the linearity issues of the proposed flow meter. Section 4.8 shows some experimental results. The chapter is concluded in section 4.10.

### 4.2 The Principle of flow meter

The proposed flow measurement system is as shown in figure 4.1. An artificial resistance to the flow of fluid is being introduced. This can be done by increasing the friction in the pipe, smoothly varying the constrictions like venturi, orifice constriction or any other existing standard methods. There will be a pressure drop due to this flow resistance. Differential pressure sensor is used to measure the pressure drop across this flow resistance. Pressure at these two points are represented as $P_1$ and $P_2$.

Pressure drop along a pipe with constant diameter is given by the Darcy-Weisbach formula [77],

$$P_{\text{pipe}} = \lambda \frac{L \rho u^2}{D^2}$$  \hspace{1cm} (4.1)

where $P_{\text{pipe}}$ is the pressure drop along a pipe, $L$ is the length of the measured section, $\rho$ is the density of the fluid, $u$ is the velocity of the liquid, $D$ is the inner diameter of the pipe and $\lambda$ is a dimensionless coefficient called the Darcy friction factor or coefficient of line hydraulic resistance, which can be found from a Moody diagram [77]. In case of laminar
flow of fluids, the Darcy factor is calculated using Stoke’s formula \cite{77} given by,

\[ \lambda = \frac{64}{Re} \]  

(4.2)

where \( Re \) is the Reynolds Number, given by

\[ Re = \frac{D \times u}{\nu} \]  

(4.3)

\( \nu \) is the kinematic viscosity of the fluid. By substituting equations (4.2) and (4.3) in equation (4.1), pressure drop along a pipe is obtained as,

\[ P_{\text{pipe}} = 32\nu\rho \frac{L}{D^2} u \]  

(4.4)

For laminar flow of fluids, the characteristic equation which relates volume flow rate \( Q_v \) and the pressure drop \( P_{\text{pipe}} \) can be written as

\[ P_{\text{pipe}} = K \times Q_v = 32\nu\rho \frac{L}{D^2} \times \frac{\pi D^2}{4} \times u \]

\[ = \left( \frac{128\rho\nu L}{\pi D^4} \right) \times (A_p \times u) \]  

(4.5)

Where,

\( Q_v = A_p \times u \) and \( K = \frac{128\rho\nu L}{\pi D^4} \)  

(4.6)

In the equation (4.6), \( K \) is obtained by substituting for \( u \) in equation (4.4), considering cross sectional area \( A_p \) for circular pipe. Where \( \nu \) is the kinematic viscosity of the fluid and cross sectional area \( A_p \) is considered as circular. For laminar flow, pressure drop along a pipe is directly proportional to the flow rate of the fluid as in equation (4.5).

In case of turbulent flow, the characteristic equation is given by,

\[ P_{\text{pipe}} = K \times Q_v^2 \]  

(4.7)

Where,

\[ \frac{1}{K} = A_p \times \sqrt{\frac{2D}{\lambda \rho L}} \]  

(4.8)

Most of the existing differential pressure flow meters like elbow, flow nozzle, orifice, pitot
Chapter 4. Flow measurement

![Block diagram of proposed flow meter](image)

Figure 4.1: Block diagram of proposed flow meter

tube, venturi, wedge etc have a medium or high pressure drop though they are simple and easy in design. Inclusion of this type of meter causes significant pressure drop in a system and changes the operating point especially when the operating flow rate is very small. In order to balance the pressure drop due to this obstruction to flow of fluid, a pump is being introduced as shown in figure 4.1. This pump has a smoothly variable pressure head. The input voltage to this pump is a function of the pressure drop across the flow resistance. For a given flow rate, at a specific input voltage to the pump, the pressure head due to the pump cancels the pressure drop due to the flow of fluid and the pressure difference \( P_1 - P_2 \) becomes zero.

As the flow rate increases, the pressure drop across the flow resistance also increases as per equations (4.5) or (4.7) depending on the type of flow. Therefore, increased voltage has to be applied to the pump in order to compensate (balance) the pipe pressure drop. The voltage applied to this pump will give the measure of mass flow rate. The dynamic response of the flow meter can be improved by automatically controlling the pressure drop to zero using a controller like the Proportional-Integral (PI) controller. Whenever there is a small change in the flow rate, the controller will vary the output signal according to the pressure drop and finally adjust \( P_2 \) to \( P_1 \).

In this proposed method, the pump exerts an influence on the measured system so as to oppose the effect of the pressure drop due to the flow rate which is being measured. This influence and the measurand are balanced until they are equal in magnitude but opposite in direction, yielding a null measurement. A null instrument offers certain intrinsic advantages over other modes of measurement technique. By balancing the unknown input against a known standard measurable input (voltage input to the pump), the null method minimizes interaction between the measuring system and the measurand. As
each input comes from a separate source, the significance of any measuring influence on
the measurand by the measurement process is reduced. In effect, the measured system
sees a very high input impedance, thereby minimizing loading errors. This is particularly
effective when the measurand is very small value. Hence, the null operation can achieve a
high accuracy for small input values and low loading errors. Null-deflection also implies
that the indication is independent of the calibration of the indicating device (pressure
sensor) or any characteristics of it.

For any flow rate, the controller adjusts its output such that the pressure drop \( P_1 - P_2 \)
is made zero under steady state. Therefore inclusion of this type of meter does not af-
fect the original operating point and hence the functioning of the system. Hence it is a
non-disruptive type of meter. One of the major advantages of the null-deflection type of
measurement is that one can measure very low flow rates (close to zero).

4.3 Modeling

The bond graph method is applied in this section to model the flow dynamics [80, 81, 119].
The proposed flow meter system is a complex multi energy domain system comprising
power/energy flow across several domains such as hydraulic, electrical and mechanical.

Fluid is pumped from the sump using a pressure source like a centrifugal pump and cir-
culated through a load and the proposed flow meter which is shown in figure 4.2. Figure
4.3 shows the bond graph model of the system. Pressure head generated by the main
pump is represented as $P_h$. $R_1$ and $P_{R_1}$ are lumped flow resistance and pressure drop due to friction in the pipe respectively. $C_1$ and $C_2$ represent hydraulic capacitance at air columns where differential pressure is measured. The intentional flow resistance introduced between these two columns for the measurement purpose is represented as $R_2$ and corresponding pressure drop as $P_{R_2}$.

According to the pressure difference at hydraulic capacitances $C_1$ and $C_2$, the controller generates its output. Proportional-integral (PI) controller is used for controlling the pressure drop to zero. Output of this controller is fed as input voltage to a centrifugal pump whose model is shown in a box with dashed lines in figure 4.3. The electrical domain is represented by the armature resistance $R_a$ and inductance $L_a$ of the DC motor. Equation for the state $i_a$ is given by equation (4.9). A modulated gyrator interfaces between the electrical and the mechanical domain. Equation (4.10) gives the angular speed of the motor shaft which is decided by the inertia of the system $J$ and friction $B$. Pressure head $P_e$ is obtained from the mechanical domain through another modulated gyrator whose modulation depends on the shaft speed. $P_c$ represents the compensating pressure which is obtained after deducting friction head $P_{fric}$ from $P_e$ as in equation (4.11). State equations for the centrifugal pump are obtained from the bond graph model by applying the junction rules [80, 81, 119].

![Figure 4.3: Bond graph model of the system](image-url)
\[
\frac{d}{dt}(i_a) = \frac{1}{L_a} [V - e_b - v_R] \\
= \frac{1}{L_a} [V - K_e \times \omega - i_a \times R_a]
\] (4.9)

\[
\frac{d}{dt}(\omega) = \frac{1}{J} [T_l - T_e - B \times \omega] \\
= \frac{1}{J} [K_e \times i_a - K \times \omega \dot{q} - B \times \omega]
\] (4.10)

\[
P_c = P_e - P_{fric} \\
= K \times \omega^2 - \dot{q} \times R_{friction}
\] (4.11)

Similarly applying the 0-junction law, the state equations for the flow meter are obtained as,

\[
\frac{dP_1}{dt} = \frac{1}{C_1} (\dot{q}_1 - \dot{q}_3) = \frac{1}{C_1} \left( \frac{P_h - P_1}{R_1} - \dot{q}_3 \right)
\]

\[
\frac{dP_2}{dt} = \frac{1}{C_2} (\dot{q}_3 - \dot{q}_5) = \frac{1}{C_2} \left( \dot{q}_3 - \frac{P_h}{R_{Load}} \right)
\] (4.12)

Applying the junction rules to the first 1-junction, one gets equation (4.13).

\[
P_1 = P_h - P_{R_1}
\] (4.13)

Similarly equation (4.14) is obtained by applying 1-junction rule to the other junction.

\[
P_1 + P_c - P_{R_2} - P_2 = 0
\] (4.14)

The controller takes signals $P_1$ and $P_2$ and varies $P_c$ to make $P_1 = P_2$. Substituting this in equations (4.14) and (4.13),

\[
P_c = P_{R_2} \text{ and } P_L = P_h - P_{R_1}
\] (4.15)

Equation (4.15) shows that the pressure drop across $R_2$ is compensated by the pump. Load pressure $P_L$ (see figure 4.3) is invariant with or without the introduction of the flow meter. Inclusion of flow meter does not affect the system as the load pressure and source pressure are unchanged. Under steady state, flow rate into capacitance $C_1$, $\dot{q}_2$ and flow rate into $C_2$, $\dot{q}_4$ vanish. Hence flow rate $\dot{q}_1$ becomes equal to flow rate $\dot{q}_5$. 

4-8
Figure 4.4 shows the block diagram of the control system. The function of the controller is to keep the pressure difference at zero. Output $y$ is the pressure difference $P_2 - P_1$ between the two ends of flow meter. This is sensed using a differential pressure transducer and fed back as feedback signal. Since the output needs to be controlled to zero, the reference signal is set at zero. Error signal $e$ obtained is given to a PI controller. Output of this controller which is called as control voltage $V_c$ is fed to the dc motor driving the pump. Corresponding to the control voltage, the pump generates pressure $P_c$ in a direction to oppose the pressure drop in the flow meter system.

The system of figure 4.3 is modeled and simulated in Matlab-Simulink as shown in figure 4.5. $P_1$ and $P_2$ are the feedback signals which are fed to the PI-controller. $K_p$ and $K_i$ are the proportional and integral gains respectively. The output of the controller is given to the dc motor-pump model and the compensating pressure $P_c$ is obtained.

For the initial 5 seconds, the model is run without inserting the flow meter in the system. At $t=5$ sec, a flow resistance $R_2$ is introduced in the path of flow. Figure 4.6 shows that as the flow resistance is introduced, pressure drop across the flow resistance increases and this affects the flow rate in the system. The flow rate decreases due to insertion of $R_2$. The compensating pump is activated in order to nullify this pressure drop at time $t=10$ sec as shown. This makes the pressure drop to zero and the flow rate regains its original value. This result clearly shows the concept of compensating the pressure drop. In very low flow rates, even small pressure drop due to the measuring unit affects the actual flow and hence the functioning of the measurand system. By compensating this pressure drop, the measurand is not being disturbed.

The controller is activated at $t=10$ sec. In accordance to the pressure difference $P_1 - P_2$, the controller generates its output voltage. Figure 4.8 shows this output voltage, which is fed to the dc motor-pump system. Steady state value of this voltage is a measure of the flow rate. Figure 4.7 shows the variation in the compensating pressure from the pump.
due to the output voltage from the controller. Figure 4.6 shows the compensation of the
pressure drop by the compensating pump and flow rate in the system regains its original value.
Chapter 4. *Flow measurement*

### 4.4 Design

The design of the proposed flow meter involves various considerations like selection of pressure transducer, pump and other accessories according to user specifications. Three major user specifications are range, accuracy and minimum flow rate that needs to be measured. The range and minimum flow rate requirements are contradictory specifications and hence one needs to compromise between selections of these two parameters. For very low flow rates, it is very expensive to have very large range of measurement. After finalizing the minimum flow rate to be measured, select flow resistance such that the pressure drop is significant enough for sensing at that flow. The pump should be able to generate the minimum pressure head to compensate the pressure drop due to even the smallest flow rate. Power rating of the pump is decided by the pressure drop corresponding to maximum flow rate to be measured. Proof pressure (i.e. absolute pressure the sensor can handle) of the pressure transducer should be more than the absolute maximum pressure at the insertion position of the meter. Valves and other accessories also should be capable of withstanding this pressure. Proportional-integral (PI) controller is being used in order to make steady state error zero. Selection of controller parameters like proportional and integral gains are selected based on methods like Ziegler-Nichols rule [120].

### 4.5 Implementation

Figure 4.9 shows the block diagram of the experimental setup of the flow meter. This consists of two air columns at the ends of the meter and a centrifugal pump and flow resistance in between them. The flow resistance is implemented by constricting the cross sectional area of the pipe. The control valve is fixed at the top to adjust the initial fluid level in the air column. Another outlet of the air column is connected to the terminal of the differential pressure sensor. Micro structure pressure sensor ASCX01DN has been used for this proposed experimental setup. This can measure up to 1PSI (6,895Pa) differential pressure with ±1% accuracy. Typical accuracy of the pressure sensor used in lab prototype is ±0.2% and they are calibrated for full span within ±1% accuracy. Once water reaches the desired level, both the control valves (valve-1 and valve-2) are closed completely (air tight). The output of this sensor is fed to a PI-controller (Proportional Integral). The output of the controller drives the dc motor, connected to the pump shaft through a variable power supply as indicated in figure 4.2. As flow rate is increased,
pressure $P_1$ increases. The controller senses the differential pressure and increases the voltage input to the pump to increase pressure $P_2$. Finally under steady state, the controller output settles at a constant voltage which makes this pressure difference zero. The output of the controller or input voltage to the pump is measured. This voltage gives a measure of the flow rate.

![Differential Pressure Output](image)

**Figure 4.9:** Engineering drawing of flow measurement system

The circuit diagram used for controlling the compensating pump is as shown in figure 4.10. The controller takes the feedback signal from the pressure sensor. Potentiometer TR1 sets the reference voltage corresponding to zero differential pressure. Feedback voltage from the sensor is compared with this reference voltage to give error signal. Output from error amplifier (U2B) is fed to a Proportional-Integral (PI) controller. Resistors R11 and R7 and capacitor C5 decides the proportional and integral constants of the controller given by equations (4.16). Output of the PI controller is given as reference to a linear voltage regulator LM723. Compensating pump is driven by the output of this voltage regulator available at J3.

\[
\frac{output}{error} = K_p + \frac{K_i}{s} = K_p + \frac{R_{11}}{R_7}; \quad K_i = \frac{1}{R_7 \times C_5}
\]  \hspace{1cm} (4.16)

For testing and calibration of this measuring unit, the experimental setup is arranged as shown in figure 4.11. Water is pumped from a sump using centrifugal pump. This flows through the flow measurement system and load (or head). Voltage input to the
main pump is kept constant so that flow rate remains constant. Finite amount of water is collected from the outlet of pipe and time taken for this is noted. Actual flow rate is calculated by knowing these two quantities. Output voltage of the controller is noted. This is repeated for different flow rates by varying the power supply to the main pump. Relation between the flow rate and the output voltage are found by regression.
4.6 Effect of pressure transducer performance limits

For any given flow meter application, there are three specifications namely the range, accuracy and minimum flow rate that needs to be measured. The lowest measurable flow rate for a given flow meter is crucial in selecting the flow meter for applications where flow is very small. In the proposed measuring system, lowest pressure drop that can be sensed by the pressure transducer decides the lowest flow rate that can be measured for a given flow resistance.

The cost of the pressure transducer depends on the range, accuracy and especially on the lowest pressure that can be measured. As pressure transducer absolute range decreases, cost increases. In the proposed system, one can measure even very low flow rate using the pressure transducer of similar range compared to that used for medium or high flow rate measurement. Let $P_{\text{min}}$ be the minimum pressure that can be sensed using a transducer. For a given hydraulic flow resistance $R_h$, the lowest flow rate $\dot{q}_{\text{min}}$ that can be measured is given by equation (4.17).

$$P_{\text{min}} = \dot{q}_{\text{min}} \times R_h$$

(4.17)

In order to measure flow rate less than $\dot{q}_{\text{min}}$, $R_h$ has to be increased in same proportion. This can be done by increasing the constriction of the flow resistance. Therefore, by keeping the minimum flow rate and flow resistance product constant, the same pressure transducer can be used to measure even very small flow rate. This kind of flexibility is not found in the existing differential pressure flow meters.

The flow resistance is increased at low flow rates in order to amplify the pressure drop such that it is within the range of the pressure transducer. This implies that the flow measurement can be made at the accuracy as that of the pressure transducer independent of the flow rate. This is one of the significant advantages of the proposed null deflection flow measurement system.

4.7 Linearity

The relation between the flow rate and output voltage is characterized by the type of pump used for compensation and the type of flow resistance. Let function $f_1$ denote the
characteristic of the pump relating input voltage and the pressure head $P_c$ for a given flow rate. Let pressure drop across the flow resistance $P_{R_2}$ be given by a function $f_2$.

$$P_c = f_1(V, \dot{q}) \text{ and } P_{R_2} = f_2(\dot{q})$$

(4.18)

Under steady state operation, since these two pressures are balanced, one can apply the following,

$$P_c = f_1(V, \dot{q}) = P_{R_2} = f_2(\dot{q})$$

(4.19)

Let us relate flow rate and output voltage by a function $g$, such that $\dot{q} = g(V)$. In order to obtain a linear relationship between flow rate and output voltage, $g$ has to be a linear function. Replacing $\dot{q}$ by $g(V)$, equation (4.19) becomes,

$$f_1(V, g(V)) = f_2(g(V))$$

(4.20)

Since $f_2$ is single-valued and monotonically increasing function for the input range of non-negative real numbers, inverse of $f_2$ exists and is defined as $f_2^{-1}$. Equation (4.20) can be re-written as,

$$g(V) = f_2^{-1} [f_1(V, g(V))]$$

(4.21)

Since $g$ is a linear function, it should satisfy scalar multiplication and superposition.

$$g(\alpha V) = \alpha g(V) \quad \forall \alpha \in \mathbb{R}$$

(4.22)

Substituting this condition in equation (4.21), one obtains,

$$g(\alpha V) = \alpha g(V) = f_2^{-1} [f_1(\alpha V, \alpha g(V))]$$

(4.23)

From equations (4.21) and (4.23),

$$f_2^{-1} [f_1(\alpha V, \alpha \dot{q})] = \alpha \times f_2^{-1} [f_1(V, \dot{q})]$$

(4.24)

Similarly applying superposition principle for function $g$, one obtains equation,

$$f_2^{-1} [f_1 (V_1 + V_2, \dot{q}_1 + \dot{q}_2)] = f_2^{-1} [f_1 (V_1, \dot{q}_1)] + f_2^{-1} [f_1 (V_2, \dot{q}_2)]$$

(4.25)
From equations (4.24) and (4.25), one observes that function $f_2^{-1} \circ f_1$ has to be linear homogeneous function of order one with two variables in order to get linear relationship between flow rate and output voltage. For any pump selected for compensation, function $f_1$ can be obtained from its state equations. One need to select the flow resistance characteristic function $f_2$ such that $f_2^{-1} \circ f_1$ is linear homogeneous function of order one, which is of the form $aV + b\dot{q}$. Selection of this type of flow resistance will ensure that output voltage is linear with respect to flow rate. Relationship between flow resistance and the corresponding pressure drop for different types of shapes are given in [102]. Alternatively, if the flow resistance cannot be modified, the motor-pump and drive system characteristic $f_1$ has to be designed or selected such that $f_2^{-1} \circ f_1$ is a linear homogeneous function of order one.

### 4.8 Experimental Results

The experiment is conducted for the setup as shown in figure 4.11. Voltage input to the main pump which is used to circulate fluid is varied. For each input voltage, finite volume of water is collected and time taken for that is noted. From this, the flow rate is calculated. Voltage applied to the compensating pump to nullify the pressure is noted. For different flow rates the controller voltages are noted and plotted. This experiment is repeated for different external conditions. Reproducibility and repeatability are verified experimentally. Linear curve fitting is obtained as shown in figure 4.12. For the pump used, the regression fit is obtained as,

\[
\text{Flow rate in } \text{kg/s} = 0.0113 \times \text{Voltage} - 0.0071 : R^2 = 99.4\% 
\]

From the above equation, one observes that there is an offset (intercept). This is due to the fact that for zero flow rate, one needs to apply some minimum voltage to the pump in order to overcome friction losses in it.

An analog controller (using op-amp) has been used to compensate the pressure drop across the flow resistance. Output from the pressure sensor is sampled continuously and corrective control action is taken. Hence the controlling voltage also gets updated continuously according to the error voltage. Proportional constant $K_p=0.1868$ and integral constant $K_i=20.024$ have been used in the laboratory prototype. The system takes around 1 second to settle since hydraulic time constant of the system is larger compared to the
controller time constant. Figure 4.13 shows the transient response of the controller. Initially feedback signal is disconnected. Once the flow reaches steady state, the controller is activated. Output voltage of the controller is measured and plotted. Corresponding simulation result is shown in figure 4.8.

The laboratory prototype of the proposed system has been tested for measuring mass flow rate from 0.005 kg/s to 0.035 kg/s (Reynolds number 700) having turndown ratio (ratio of maximum to minimum flow rate that can be measured with the meter) of 7:1. It should be noted that the above mentioned range is not due to the limitation of the proposed flow measurement concept, but due to the limitation on the pressure head delivered by the external pump used in the experimental setup of figure 4.11. However, this principle can be extended to measure flow rate smaller than this. Root mean square of the percentage error with reference to full scale is obtained as ±1.8%. Accuracy of the measurement system can be improved by using pressure sensor of better accuracy.
4.8.1 Comparison

A comparative study of the existing flow meters with their possible minimum measurable flow rate and relative cost has been carried out. For each flow meter, the minimum measurable flow rate is calculated from the minimum diameter and the lowest Reynolds number necessary for the measurement [121]. The calculated values are tabulated in table 4.1. The first six (1-6) flow meters are based on differential pressure type, which are having non-linear relationship between flow rate and the output. The remaining flow meters have a linear scale.

From table 4.1, it is observed that most of the disruptive type of flow meters are applicable for flow rates corresponding to Reynolds number greater than 10,000 though their cost is lower. Only few non-disruptive flow meters like magnetic, ultrasonic type can be used for low flow measurements (Reynolds number of the order of 5,000) which are having very high initial cost. Though the proposed flow meter has been used to measure only up to 5g/s experimentally, one can measure still lower flow rates compared to this based on the discussion of section 4.6. As the proposed system is non-disruptive and null-deflection type, one can measure flow rates very close to zero.
Chapter 4. Flow measurement

<table>
<thead>
<tr>
<th>Sl.</th>
<th>Type</th>
<th>Min. diameter in mm</th>
<th>Minimum Re number</th>
<th>Mass flow rate in kg/s</th>
<th>Mass flow rate in g/s</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Orifice V-cone</td>
<td>12</td>
<td>8,000</td>
<td>0.0754</td>
<td>75.398</td>
<td>Low</td>
</tr>
<tr>
<td>2</td>
<td>Venturi</td>
<td>50</td>
<td>75,000</td>
<td>2.945</td>
<td>2,945.25</td>
<td>Med</td>
</tr>
<tr>
<td>3</td>
<td>Flow nozzle</td>
<td>50</td>
<td>50,000</td>
<td>1.963</td>
<td>1,963.5</td>
<td>Med</td>
</tr>
<tr>
<td>4</td>
<td>Low loss venturi</td>
<td>75</td>
<td>12,800</td>
<td>0.754</td>
<td>753.98</td>
<td>High</td>
</tr>
<tr>
<td>5</td>
<td>Pitot</td>
<td>75</td>
<td>100,000</td>
<td>5.890</td>
<td>5,890.49</td>
<td>Low</td>
</tr>
<tr>
<td>6</td>
<td>Elbow</td>
<td>50</td>
<td>10,000</td>
<td>0.393</td>
<td>392.7</td>
<td>Low</td>
</tr>
<tr>
<td>7</td>
<td>Magnetic</td>
<td>2.5</td>
<td>4,500</td>
<td>0.009</td>
<td>8.836</td>
<td>High</td>
</tr>
<tr>
<td>8</td>
<td>Turbine</td>
<td>6</td>
<td>5,000</td>
<td>0.0236</td>
<td>23.562</td>
<td>High</td>
</tr>
<tr>
<td>9</td>
<td>Ultrasonic-Time of flight</td>
<td>12</td>
<td>10,000</td>
<td>0.0942</td>
<td>94.25</td>
<td>High</td>
</tr>
<tr>
<td>10</td>
<td>Ultrasonic-Doppler</td>
<td>12</td>
<td>4,000</td>
<td>0.0377</td>
<td>37.699</td>
<td>High</td>
</tr>
<tr>
<td>11</td>
<td>Vortex Shedding</td>
<td>40</td>
<td>10,000</td>
<td>0.3142</td>
<td>314.16</td>
<td>High</td>
</tr>
<tr>
<td>12</td>
<td>Proposed</td>
<td>diameter used 9</td>
<td>700</td>
<td>0.005</td>
<td>5</td>
<td>Low</td>
</tr>
<tr>
<td>13</td>
<td>Proposed- theoretical</td>
<td>No limit</td>
<td>No limit</td>
<td>No limit</td>
<td>No limit</td>
<td>Low</td>
</tr>
</tbody>
</table>

**Table 4.1: Comparison of existing flow meters**

### 4.9 Integration with cooking system

The flow meter discussed in this chapter is used in the proposed cooking system to measure flow rate of the fluid on the collector side and on load side as shown in figure 4.14. Two identical flow meters are designed as per figure 4.2 and calibrated individually as explained in section 4.5. The measured mass flow rates from these flow meters are used to calculate the energy transported from the solar thermal collector to the buffer tank and from the buffer tank to the cooking load. The overall performance is optimized based on these energy calculations as explained in the maximum power point tracking (MPPT) algorithm in section 2.3.3.
Chapter 4. **Flow measurement**

The preceding chapters elucidated the need for flow rate control. This implies that the flow rate needs to be sensed with accuracy and precision. As the flow rates involved in this proposed system are very low, existing flow measurement methods and apparatus are either too expensive or disruptive. Even if they are non-disruptive, they need additional constraints to be applied on the properties of the thermic fluid and the pipe material. This chapter proposes a novel technique to measure mass flow rate of any type of liquid by compensating the pressure drop across a flow resistance using a compensating pump. Since the pipe pressure drop is canceled by the pump pressure, the pressure drop across the unit is zero. This is a null deflection and non-disruptive method of measurement. This allows the measurement of very low flow rate with low resolution. The flow meter system is modeled using bond graph and its dynamic equations are obtained. Simulation using the bond graph technique has been discussed. This method is experimentally verified and validated. This type of flow meter is used to measure flow rate of the fluid on both collector side and the load side to perform the energy optimization.

It may be noted that the pump used for compensating the pressure drop is generally very small as the hydraulic power is very low at low flow rate. This is used only to produce a pressure head equal to the maximum pressure drop across the flow resistance. As flow rate is small, the pump wear may not be a serious issue in this case. The accuracy is dependent on the accuracy of the differential pressure sensor. For lower flows, the flow resistance can be increased to bring the pressure drop near to the full scale reading of the

---

**Figure 4.14:** Experimental setup with flow meter

4.10 Conclusion
same pressure sensor. It is important to note this point that, the resolution of the differ-
ential pressure transducer need not be stringent. Even a coarse resolution can be used for
high precision low flow measurement by increasing the adjustable flow resistance. This
in fact is one of the significant advantages of this concept which leads to an inexpensive
and accurate low flow measurement apparatus.

The measurement technique proposed in this chapter can also be used for different low
flow measurement applications like

- Fuel flow measurement in vehicles, which can be used to get instantaneous mileage
  in order to optimize speed at which fuel consumption is minimum
- Custody transfer of oil and gas
- Accurate chemical dosing in chemical treatment, pharmaceuticals
- Biomedical applications, etc

The next chapter discusses the design process of the complete hybrid cooking system that
is proposed in this work.
Chapter 5

Design of heat transport system

5.1 Introduction

The flow measurement technique was discussed in previous chapter which is essential to optimize the energy flow in the proposed cooking system. In a practical cooking system, there are many more components whose design issues are to be addressed. This chapter mainly focuses on systematic design procedure for the proposed cooking system.

Design is a complex and challenging task since the working system involves multi domain subsystems like thermal, mechanical, electrical and hydraulic. The complexity of design further increases as this system is a hybrid combination consisting of both solar thermal and conventional cooking source of energy. Design of each subsystem is performed considering the worst case conditions for a given location and cooking load.

An overall schematic of the hybrid solar cooking system is shown in figure 5.1. This system can be split into the following major components,

- Solar thermal concentrator and collector, with tracking system.
- Heat storage tank with thermal insulation.
- Heat transfer fluid, piping for circulation with thermal insulation.
- Heat exchanger system.
- Pumps for circulation of the fluid with flow rate adjustment mechanism.
Specifications that are involved in design of proposed solar cooking system can be divided into two major parts. They are,

- **User specifications**: User gives the specifications related to the cooking requirements like number of hours of cooking in a day, amount of night time cooking and type of conventional cooking source used. Distance between kitchen and rooftop is also very important during design, which has to be specified by the user according to the building structure.

- **Design parameters**: For a given user specifications, design parameters are obtained such as size of solar thermal collector, size of heat storage tank, design of heat exchanger, circulating pump, size of PV panel with battery and design of sensing instrumentation.

Solar energy is trapped using a concentrating collector, which raises the temperature of the circulating fluid. Depending on the user requirement and available solar insolation...
at the location, a thermal collector of appropriate size can be selected. Maximum energy that can be extracted from sun is a dynamic quantity dependent on the solar insolation available at that instant. On an average, 5 kWh of energy is available per square meter on a clear sky day. As per user specification, the average cooking load requirement per day can be estimated in kWh. The size of the collector can be calculated in order to meet this requirement considering overall efficiency of the collector.

As energy obtained from the solar collector is not uniform, energy storage facility is necessary. For a given location, number of cloudy days or no sunny days can be estimated. This has to be considered while designing the size of the storage tank. Amount of energy required for night and early morning cooking also has to be considered while sizing.

Dimension and design of the heat exchanger is decided according to the cooking load and type of application. Two pumps are used to circulate the fluid which are driven by PV panels. Pressure head of these pumps are decided according to the height difference between kitchen and rooftop where solar collectors are placed.

In order to obtain sizes of PV panel and battery storage, it is necessary to obtain all electrical requirements like pump, linear actuator for tracking, sensing and control circuits. Details of these issues are discussed in the following sections.

5.3 Estimation of Solar collector size

Design of size of solar thermal collector and PV panel mainly depends on two factors.

- Availability of solar energy at an user defined location
- Amount of energy required for cooking.

5.3.1 Estimation of solar insolation

Performance of the proposed solar cooking system is directly affected by the amount of solar insolation available to the system. Amount of energy received at a particular location is a function of different parameters like, time of day, day, latitude of the location and
atmosphere. The monthly averages of daily extra terrestrial global solar radiation for a horizontal surface \( \overline{H}_o \) at any location can be given by equation (5.1).

\[
\overline{H}_o = \frac{24}{\pi} I_o \left[ \sin(\phi) \sin(\delta) \omega_s + \cos(\phi) \cos(\delta) \sin(\omega_s) \right] \text{ kWh/m}^2/\text{day} \quad (5.1)
\]

Where,

- \( I_o \) = Extra-terrestrial beam normal irradiance on a day in kW/m\(^2\)
- \( I_{sc} = \) Solar constant = 1367 W/m\(^2\)
- \( N = \) Day of the year (\( N=1 \) on January 1st and \( N=365 \) on December 31st)
- \( \phi = \) Latitude of the location in degrees
- \( \delta = \) Declination angle in degrees = 23.45 \( \sin \left( \frac{360(N-80)}{365} \right) \)
- \( \omega_s = \) Hour angle at sunrise/sunset = \( \cos^{-1} (-\tan(\phi) \tan(\delta)) \)

It can be seen from the above expressions that the extra-terrestrial horizontal insolation is a function of latitude and the day of year only. Hence, it can be calculated for any location for any given day. Instead of calculating for each day, one can calculate for a typical day of a month which is representative of the average of that month. To obtain solar insolation available on a horizontal surface, atmospheric effects like water vapor should also be considered.

A stochastic measure of these atmospheric effects is called as clearness index \( K_T \), which is a periodic function of time of year. In order to include the atmospheric effects on the insolation at a given place, a model for the clearness index is used. \( K_T \) is calculated according to the model proposed by Ravinder [122].

The global solar insolation with atmospheric effects is obtained as,

\[
H_{tc} = K_T \overline{H}_o \quad (5.2)
\]

Where \( H_{tc} \) represents the solar insolation at the earth’s surface on a horizontal plate at any location on any given day. The monthly average daily insolation is calculated for all the months over a year considering a typical day of each month. From these values, the lowest value of the solar insolation is considered for the design of solar thermal and photo voltaic collectors.
5.3.2 Sizing of solar collector

Using the equations as discussed in section 5.3.1, the minimum solar insolation available can be calculated for any given location. Using this information, sizes of solar thermal collector and PV panel can be obtained. Solar thermal collector is used to raise the temperature of the heat transfer fluid that is used for transferring heat to the heat exchanger for cooking food. The PV panels convert energy from the sun into electrical form which is being used to energize the electrical equipments and sensing instruments used in the proposed system.

5.3.2.1 Solar thermal collector

The primary function of the solar thermal collector in this system is to heat up the heat transfer fluid. In order to calculate the size of the collector required, it is necessary to know the energy requirement for a specified cooking pattern. According to the type and the amount of cooking fuel used per day, the average energy requirement can be estimated according to equation (5.3). The average fuel consumption for cooking per day, \( V_f \) is multiplied by the calorific value \( C_{pf} \) of that fuel to obtain the amount of energy required in \( kWh/day \).

\[
E_{req} = \sum_i (V_f C_{pf_i}) \quad \forall i (5.3)
\]

Where \( V_f \) is fuel consumed in \( m^3/day \) and \( C_{pf} \) is the calorific value of fuel in \( kWh/m^3 \). The incident solar energy available at the location is calculated as explained in previous section. Considering \( \eta_1 \) as the efficiency of the collector that is being used, area of the collector in \( m^2 \) can be calculated as in equation (5.4).

\[
A_{col} = \frac{E_{req}}{\eta_1 H_{tc}} (5.4)
\]

Where \( H_{tc} \) is as given in equation (5.2). For the proposed system, the temperature of the fluid should reach above 100°C. Liquid flat plate collectors are used for water heating application wherein about 80°C can be obtained. This type of collectors absorbs both direct and diffused sunlight. Therefore it can be used even during cloudy days.
Concentrating collector has to be used for medium or high temperature application, where sunlight is focused onto either a point or a line. Concentrating collectors can be classified on different basis. Based on the operating principle, it can be divided as reflecting type and refracting type. Depending on the type of focusing, it can be either line focusing or point focusing. Similarly it can also be classified according to the concentration ratio, type of tracking etc. The design and principle behind the functioning of few concentrating collectors are discussed in the following paragraphs.

**Linear parabolic collector:** This is also known as cylindrical parabolic collector. The reflecting material has a cylindrical parabola shape that focuses sunlight falling on it onto a line located along the focal axis. The concentration ratio \( CR \) is defined as the ratio of the area of aperture \( A_a \) to the area of the receiver \( A_r \). Rim angle is the angle made by extreme point on parabola at the focal point with respect to normal of aperture. \( CR \) varies from 10 to 80 having rim angles of 70° to 120°. The reflecting surface can be silvered glass, electro-polished anodized aluminum sheet or silver-coated acrylic film. Copper or steel tubes are placed at the focal line of the parabola with heat resistant black paint or a selective surface like black chrome. To increase the heat retention, a transparent glass cover surrounds the absorber tube. The supporting structure of the concentrator should be able to withstand self-weight and wind-loading.

Figure 5.2: Linea parabolic collector

Figure 5.2 shows a typical linear parabolic collector. Fluid is passed from one end along the focal line of the parabola. The concentration ratio for such a concentrating collector
of width $W$, length $L$ and absorber tube having diameter $D_o$ is given by equation (5.5).

$$ CR = \frac{A_a}{A_r} = \frac{(W - D_o) L}{\pi D_o L} = \frac{W - D_o}{\pi D_o} $$

(5.5)

In the proposed cooking system, if linear parabolic collector is to be used, then heat transfer fluid is passed through the absorber tube. The fluid that is allowed to flow through the pipe absorbs energy from the sun and consequently, the temperature of the fluid at the outlet is raised as compared to that at inlet. The analysis of a typical cylindrical parabolic collector is given in appendix A.

There are different modes for tracking the sun by the linear parabolic collector depending on the orientation. If its focal axis is placed in east-west direction, tracking is only for declination that can be done once a day. However if the focal axis is in north-south orientation, then both diurnal tracking for time of day and declination tracking for time of year are required. Acceptance angle is twice the maximum angle the sun beam makes with the normal to the aperture while still being able to reach the absorber. This decides the frequency at which the orientation has to be adjusted.

**Paraboloid collector:** Higher concentration can be achieved in two dimensions by using a paraboloid dish collector. A paraboloid is produced when a parabola rotates about its axis which looks similar in appearance to a satellite dish. The shape of a parabola is defined such that incoming rays which are parallel to the dish’s axis will be reflected toward a single point called as focal point. The collector is placed at the focal point in order to absorb heat energy from the sun as shown in figure 5.3.

![Paraboloid collector](image)

Figure 5.3: Paraboloid collector

Let the diameter of the paraboloid aperture area be $D_c$, and the diameter of the absorber area be $d_c$. The concentration ratio is given by equation (5.6). Since $D_c$ is very large
compared to $d_c$, $d_c^2$ on the numerator can be neglected. Compared to the concentration ratio of cylindrical parabolic concentrator, the concentration ratio in paraboloid concentrator is much higher due to the squaring effect.

\[ CR = \frac{A_a}{A_r} = \frac{\pi \left(D_c^2 - d_c^2\right)}{\pi \left(d_c^2\right)} \]

\[ CR \approx \frac{D_c^2}{d_c^2} \quad \text{if} \quad D_c >> d_c \quad (5.6) \]

The thermal losses from a collector are primarily radiative and can be reduced by decreasing the absorber aperture area. High collection efficiency and high quality thermal energy are the features of a paraboloid collector as the absorber area is very small. However, it should be noted that a paraboloid is most effective under clear sky conditions. The paraboloid is not very effective for diffused incident insolation. In the proposed cooking system, the fluid is circulated through the receiver to absorb energy. The amount of energy that is received at the absorber depends on the aperture area of the paraboloid which is according to the energy requirement for a specific cooking load pattern.

**Fresnel lens concentrator:** This is based on the refraction principle. Fresnel lens is a modification of the conventional plano-convex lens. The design enables the construction of lenses having large aperture and short focal length with lesser weight and volume of material compared to conventional lens. This is achieved by breaking the lens into a set of concentric annular sections known as Fresnel zones. For each of these zones, the overall thickness of the lens is decreased, effectively chopping the continuous surface of a standard lens into a set of surfaces of the same curvature, with discontinuities between them. This allows a substantial reduction in thickness and volume of material of the lens, at the expense of reducing the focusing quality of the lens. Glass and plastic can be used as refracting materials for fabricating these types of lenses. However, plastic lenses are used more often as they are economical and easy to mould unlike glass.

### 5.3.2.2 Sizing of PV panels

Photo-voltaic (PV) cell converts solar energy into electrical form. In the proposed system, two electricity driven pumps are used to circulate the heat transfer fluid through the solar collector and heat exchanger. Control over the heat transfer is by varying the input supply to these pumps. Energy necessary to circulate can be obtained from PV panels.
making the cooking system stand alone (isolated from grid). The PV panels are also used to operate other electrical equipments like linear actuator and sensing instruments like temperature sensor and flow meter. In order to calculate the size of the panel required, the energy required for operation of all these electrical loads should be estimated.

**Linear Actuator:** The solar collector tracks the sun by using a linear actuator. This is mounted on to the thermal collector through a lever mechanism such that the collector rotates in an east-west direction when the linear actuator moves up and down. The power rating of the linear actuator depends on the size and weight of the collector and the torque required to tilt it. Maximum torque is required when the solar collector is in the extreme end. In this case, the linear actuator has to tilt the collector against its mass. Equation (5.7) gives the maximum load torque \( T_{LA} \), where \( F \) is the maximum force, \( r_{LA} \) is the radius of the collector, \( m_{LA} \) is the mass and \( \omega_{LA} \) is the angular speed. Let the power rating be \( P_{LA} \). The angular speed has to be more than 15°/hour. Let \( h_1 \) be the time required to move the collector from one end to another end with supplying rated voltage to it continuously.

\[
T_{LA} = F \times r_{LA} = (m_{LA} g) \times r_{LA}
\]

\[
P_{LA} = T_{LA} \times \omega_{LA}
\]

(5.7)

**Pump:** There are two pumps used in the proposed system as shown in figure 5.1. One of them is used to circulate the fluid on the solar collector side to absorb heat from sun and the other one is used on the load side to transfer heat from the buffer tank to the heat exchanger. Let \( P_{pump1} \) be the power rating of pump-I that is placed on the collector side. \( P_{pump1} \) is decided by the required maximum rate of flow of fluid and the pressure head for the specified system. Let \( h_2 \) be the number of hours pump-I is being operated in a day. This may be any value between 6-8 hours depending on the day of the year. Similarly \( P_{pump2} \) and \( h_3 \) are the power rating and number of hours of operation of pump-II. Pump-II is operated only when cooking has to be carried out.

**Monitoring and controlling:** Sensing equipments like temperature sensors (and display system), flow meters requires some amount of power to operate. Thermocouple sensors are placed at different places to sense the temperature of the circulating fluid. Energy has to be supplied also to the electronics that controls the linear actuator. Another electronic board, that is used to vary the flow rate for energy optimization, also needs to be supplied by the PV panels. Let \( P_{mc} \) and \( h_4 \) be the power required by all the
above mentioned instruments and average number of hours in operation.

Designing the size of the PV panel depends on the following three factors,

- Energy required for the daytime load
- Energy required for the nighttime load
- Energy required during no-sunny days

Energy required during daytime can be met directly from the PV panel. But energy required from the load when sun is not available has to be obtained from a storage system like battery. Energy is stored into the battery during daytime and retrieved from it to supply the electric loads during nighttime or during no-sunny days. For the energy that is being drawn from the battery, efficiency of charging into battery and discharging from battery has to be considered.

Let $W_{h1}$ be the total energy required during daytime per day represented in kWh. This includes energy required by linear actuator, pump-I, pump-II and other monitoring and controlling instruments. The energy is obtained by adding the individual energy in kWh according to their power rating and duration for which it is being operated.

Let $W_{h2}$ be the total energy required during nighttime. This involves supply of power to mainly load-side pump (pump-II), monitoring and control on load-side. Then $W_{h1} + W_{h2}$ is the total energy required to meet no-sunny days demand which includes both daytime and nighttime load for specified number of cloudy days.

Total energy required to be supplied from the PV panel is given by,

$$W_{h_{total}} = W_{h1} + \frac{W_{h2}}{\eta_B} + \frac{(W_{h1} + W_{h2}) \times D_{no-sun}}{\eta_B}$$

$$= W_{h1} \left(1 + \frac{D_{no-sun}}{\eta_B}\right) + \frac{W_{h2}}{\eta_B} \left(1 + D_{no-sun}\right)$$

(5.8)

Where, $\eta_B$ is the overall battery efficiency and $D_{no-sun}$ is the number of no-sunny days.

Knowing insolation $H_{tc}$ as calculated in section 5.3.1, the area of PV needed to fulfill the electrical loads is obtained as given in equation (5.9), where $\eta_{PV}$ is the efficiency of...
### Table 5.1: Different types of PV panels

<table>
<thead>
<tr>
<th>Type</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Organic Cells</td>
<td>5%</td>
</tr>
<tr>
<td>Amorphous Silicon</td>
<td>5-6%</td>
</tr>
<tr>
<td>Polycrystalline Silicon</td>
<td>12-14%</td>
</tr>
<tr>
<td>Monocrystalline Silicon</td>
<td>15-18%</td>
</tr>
<tr>
<td>Thin film</td>
<td>18%</td>
</tr>
<tr>
<td>Group III-V Technologies (like GaAs)</td>
<td>25%</td>
</tr>
<tr>
<td>Multi-junction Solar cell</td>
<td>40%</td>
</tr>
</tbody>
</table>

The PV panel is calculated as:

$$ A_{PV} = \frac{W h_{total}}{H_{tc} \times \eta_{PV}} $$

(5.9)

Efficiency of different types of PV panels are given in Table 5.1 [123].

### 5.4 Sizing of the storage tank

Heat storage tank stores solar energy in the form of heat by raising the temperature of the fluid. When sunlight is not available, this stored energy is utilized for cooking. The size of the tank depends on the maximum amount of energy that needs to be stored and the physical properties of the fluid used for circulation. The amount of energy to be stored depends strongly on the amount of night cooking load.

$$ E_{night} = K_1 \times E_{req1} $$

(5.10)

Where $E_{night}$ is the amount of energy required for night cooking and $K_1$ is the fraction of daily load required for night cooking. Then number of no-sunny days is another factor that affects the size of the storage tank. Energy to be stored in the tank can be calculated as given in equation (5.11), where $K_2$ is the fraction of cooking load that need to be considered for sizing of tank during no-sunny days and $\eta_{storage}$ is the efficiency of storing and retrieving the energy from the tank.

$$ E_{storage} = \frac{[K_1 E_{req1} + K_2 E_{req1} \times D_{no-sun}]}{\eta_{storage}} $$

(5.11)
Chapter 5. Design of heat transport system

Thermal energy can be stored mainly in two ways, namely sensible heat storage and latent heat storage as explained in section 2.3.2. The proposed system uses the sensible heat storage method. The energy is stored by raising the temperature of the material. Specific heat of a fluid is the measure of heat energy storage capacity. For a temperature raise of $\Delta T$ and mass of $m$, energy stored in the fluid is given by $mC_p\Delta T$. The energy stored in the tank will leak to the ambient with time. Therefore all energy put into the storage tank will not be available for usage. Therefore, efficiency of the storage system has to be taken care while estimating the size.

From equation (5.11), the size of the storage tank can be calculated as,

$$ E_{storage} = m_{storage} C_p (T_{max} - T_{min}) $$

$$ m_{storage} = \frac{E_{storage}}{C_p (T_{max} - T_{min})} $$

$$ V_{storage} = \frac{m_{storage}}{\delta} \quad (5.12) $$

Where $m_{storage}$ is the mass of the storage tank, $V_{storage}$ is the volume of the tank and $\delta$ is the density of the fluid. $T_{max}$ and $T_{min}$ are the maximum and minimum temperature of the fluid in the tank.

In order to retain heat, thermal insulation has to be provided around the tank. Two temperature sensors are placed, one at the top of the tank and another one at the bottom to sense the fluid temperature. A drain valve has to be placed at the bottom to remove the fluid either during maintenance or replacement. An oil level indicator is fixed to monitor the fluid level in the tank as shown in figure 5.1.

A cylindrical shaped tank is used in order to get better temperature stratification [66]. Consider a tank with diameter $D$ and height $L$, having volume $V$. The volume of the tank is expressed as given in equation (5.13).

$$ V = \frac{\pi D^2 L}{4} \quad (5.13) $$

The total surface area $S_a$ of the tank with these dimensions is given by,

$$ S_a = \frac{\pi D^2}{4} \times 2 + \pi D L \quad (5.14) $$
The ratio of surface area $S_a$ to the volume $V$ is given as $SV$ in equation (5.15).

$$SV = \frac{S_a}{V} = \frac{2}{L} + \frac{4}{D}$$  \hspace{1cm} (5.15)

Substituting for $L$ from equation (5.13), $SV$ is obtained as,

$$SV = \frac{\pi D^2}{2V} + \frac{4}{D}$$  \hspace{1cm} (5.16)

The amount of material required for fabrication and thermal insulation of the tank depends on the total surface area. Hence cost of the heat storage tank is directly proportional to the surface area. In order to minimize the cost for a given volume of tank, the derivative of $SV$ with respect to $D$ has to be zero.

$$\frac{dSV}{dD} = \frac{\pi D}{V} - \frac{4}{D^2} = 0 \Rightarrow D = \frac{V}{L} = 1$$  \hspace{1cm} (5.17)

Therefore, from equation (5.17), optimal dimension of the tank for a given volume is obtained having diameter equal to height of the tank.

### 5.5 Sizing of battery

Battery is an electrochemical cell, which holds electrical energy in the form of chemical energy. The battery size depends on the type of the battery and the maximum amount of energy to be stored. Energy requirement from the electrical components which are used during the night time need to be supplied from the energy stored in the battery. Collector tracking system and pump-1 are operated only when sun light is present. Resetting the collector tracking system back to east direction needs equivalent of energy as was used for moving from east to west. Collector has to be brought back to east side during night time. Flow meter-1 is not used during night time. Hence these components need not be supplied from the battery. Battery has to supply the remaining electrical subsystems like pump-2, temperature sensors and flow meter-2 during the night time. Total energy that needs to be stored in the battery is calculated by adding individual Wh required during
night time.

\[
W_{h_{\text{battery}}} = \sum W_{h} \\
Q_{\text{battery}} = \frac{W_{h_{\text{battery}}}}{V} \text{ in } Ah
\]  

(5.18)

Where \(Q_{\text{battery}}\) is the amount of energy that needs to be stored in the battery in terms of \(Ah\) and \(V\) is the voltage of the battery terminal. Since voltage is almost constant in a battery, \(Ah\) is used as a capacity unit in specifying a battery. There are two terms that are related to a battery. They are, depth of discharge (DOD) and state of charge (SOC).

DOD is the maximum amount of output energy that may be removed from the battery for a given application. SOC is the amount of charge energy left in the battery.

Every battery has different DOD rating depending on the type of the chemicals used. Energy should not be discharged below the level specified by DOD of the battery. Otherwise the life of battery in terms of number of charge-discharge cycles will be adversely affected. If the battery is fully charged, it is said to have 100% energy. DOD rating of the battery depends on the type of battery. For solar applications, deep discharge batteries are preferred which have DOD of up to 80%.

Life of a rechargeable battery is specified by number of charge-discharge cycles. During its operation, it can be charged and discharged these number of times. This is valid under the condition that the charging and the discharging currents are well below the rated value.

**Capacity:** Capacity indicates the energy storing capacity of a battery. The actual capacity is measured in watt-hours (Whr). Since the voltage across the battery is almost constant dc, the capacity is also expressed in amp-hours (Ahr). In general, batteries are specified as \(C_x\), where \(C\) indicates the capacity in Ahr and subscript \(x\) denotes the charge/discharge rate. The maximum current drawn from the battery is decided by the ratio of \(C\) to \(x\). If current drawn is higher than this value, then the capacity of the battery comes down exponentially from the rated value.

From above discussed factors, capacity of the battery can be calculated as given by equation (5.19). \(Q_{\text{battery}}\) is calculated considering nominal voltage of the battery. \(\eta_{\text{battery}}\)
Chapter 5. Design of heat transport system

denotes the efficiency of the battery.

\[ C_{battery} = \frac{Q_{battery}}{\eta_{battery} DOD} \]  

(5.19)

Maximum current \( I_{max} \) drawn from the battery is estimated depending on the maximum load that are connected at any time to the battery. Then \( x \) value can be calculated as,

\[ x = \frac{C_{battery}}{I_{max}} \]  

(5.20)

5.6 Selection and sizing of pumps

There are two pumps used in the proposed system, one to circulate the fluid through the solar thermal collector and another through the heat exchanger. The purpose of these two pumps is to give necessary pressure head to control the flow rate and hence the energy collection.

5.6.1 Sizing of Pumps

Rating of the pump depends on the static head to which the fluid has to be pumped and the flow rate. Pump-I has to circulate the fluid from the storage tank to the solar collector and back to the tank. During steady flow condition in a closed circuit, pump has to supply only friction head. In order to start the flow, pump has to circulate the fluid against corresponding delivery head. Hence rating of the pump depends on the static head to which the fluid has to be pumped and the flow rate. Let \( H_{d1} \) be the delivery head in m and \( Q_1 \) be the maximum volume flow rate in \( m^3/s \) required on the collector side. Then pressure head \( P_{pump1} \) and power \( Power_{pump1} \) required for pump-I is given by equation (5.21), where \( \eta_{pump} \) is the typical efficiency of the pump, \( \rho \) is the density of the fluid and \( g \) is the acceleration due to earth’s gravitation.

\[ P_{pump1} = \rho \ g \ H_{d1} \ in \ Pa \]

\[ Power_{pump1} = \frac{P_{pump1} \ Q_1}{\eta_{pump}} \ in \ W \]  

(5.21)
Chapter 5. *Design of heat transport system*

Similarly, rating of the another pump can be decided according to the delivery head and the flow rate requirement on the load side circulation.

\[ P_{\text{pump}2} = \rho \ g \ H_{d2} \text{ in } Pa \]
\[ Power_{\text{pump}2} = \frac{P_{\text{pump}2} Q_2}{\eta_{\text{pump}}} \text{ in } W \]  \hspace{1cm} (5.22)

Where \( P_{\text{pump}2} \) is the pressure head required by pump-II, \( Power_{\text{pump}2} \) is the power required, \( H_{d2} \) is the delivery head in \( m \) and \( Q_2 \) is the maximum volume flow rate required on the load side.

Based on the operating principle, the pumps are broadly classified as positive displacement pump and rotodynamic pump.

### 5.6.2 Positive displacement pump

A positive displacement pump causes fluid to move by trapping a fixed volume of fluid, then forcing (displacing) that trapped volume into the discharge pipe. Depending on the mechanism used for moving the fluid, classification can be done as rotary pump and reciprocating pump.

#### 5.6.2.1 Rotary pump

These types of pumps can be classified as multiple rotor or single rotor according to the number of rotors. This type of pump moves the fluid by rotary motion. Vacuum is created by the rotation of the pump which draws the water from the inlet pipe.

Gear pumps are the simplest type of rotary pumps, consisting of two gears laid out side-by-side with their teeth enmeshed. The gears turn away from each other, creating a current that traps fluid between the teeth on the gears and the outer casing, eventually releasing the fluid on the discharge side of the pump as the teeth mesh and go around again. Many small teeth maintain a constant flow of fluid, while fewer, larger teeth create a tendency for the pump to discharge fluids in short, pulsing gushes.

Screw pump consists of two screws with opposing thread which rotate in opposite directions. The rotation of the screws, and consequently the shafts to which they are
mounted, draws the fluid through the pump. Other than these two, there are different types of positive displacement pumps like lobe, shuttle block, flexible vane or sliding vane, helical twisted roots pump etc, which have different structures but the principle of pumping being the same.

5.6.2.2 Reciprocating pump

Reciprocating pumps are those which cause the fluid to move using one or more oscillating pistons, plungers or membranes (diaphragms). In a reciprocating pump, a volume of liquid is drawn into the cylinder through the suction valve on the intake stroke and is discharged under positive pressure through the outlet valves on the discharge stroke. The discharge from a reciprocating pump is pulsating and changes only when the speed of the pump is changed. Since the intake is always a constant volume, volume flow rate per stroke length is fixed for a pump of given dimension. Often an air chamber is connected on the discharge side of the pump to provide a more even flow by evening out the pressure surges.

There are two types of reciprocating pumps, namely plunger pump and diaphragm pump. Plunger pumps comprises of a cylinder with a reciprocating plunger in it. The suction and discharge valves are mounted in the head of the cylinder. In the suction stroke the plunger retracts and the suction valve opens causing suction of fluid into the cylinder. In the forward stroke the plunger pushes the liquid out of the discharge valve.

In diaphragm pumps, the plunger pressurizes the hydraulic oil which is used to flex a diaphragm in the pumping cylinder. Diaphragm valves are used to pump hazardous and toxic fluids.

5.6.3 Dynamic pump

There are different types of dynamic pumps. Two important classifications are centrifugal and axial. These types of pumps operate by developing a high liquid velocity and converting kinetic energy into pressure. Dynamic pumps have lower efficiency compared to displacement pumps. Dynamic pumps are also able to operate at fairly high speeds and high fluid flow rates.
5.6.3.1 Centrifugal pump

This is also called as Radial flow pump. A centrifugal pump is a rotodynamic pump that uses a rotating impeller to increase the pressure and flow rate of a fluid. The fluid enters the pump impeller along or near to the rotating axis and is accelerated by the impeller, flowing radially outward or axially into a diffuser or volute chamber, from where it exits into the downstream piping system. As liquid moves radially outward, velocity increases due to centrifugal force. This velocity is converted to pressure which is needed to discharge the liquid.

Centrifugal pumps are typically used for large discharge through smaller heads. Advantages of centrifugal pump are smooth flow, uniform pressure in the discharge pipe, low cost and high operating speed.

5.6.3.2 Axial flow pump:

This is also called as propeller pump. These pumps develop pressure by the propelling or lifting action of the vanes on the liquid. Axial flow pumps differ from radial flow in that the fluid enters and exits along the same direction parallel to the rotating shaft. The fluid is not accelerated but instead "lifted" by the action of the impeller. Axial flow pumps operate at much lower pressures and higher flow rates than radial flow pumps.

5.6.4 Internal gear rotary pump

Since solar cooking system requires pumps to circulate servotherm oil having high viscosity, internal gear rotary pump is more suitable. Figure 5.4 shows a typical internal gear rotary pump.

Internal gear pumps are used on liquids, fuel oils and very useful for circulating oils of viscosity from 1cP to over 1,000,000 cP. In addition to their wide viscosity range, the pump has wide temperature range as well, handling liquids up to 400°C. The distance between the ends of the rotor gear teeth and the head of the pump is called as the clearance. As this clearance is adjustable to accommodate high temperature, maximize efficiency for handling high viscosity liquids and to accommodate for wear. Advantages of internal gear pump are non-pulsating discharge; self-priming and can run dry for short periods. They
are bi-rotational and hence can be used in both the directions. Working principle of gear pumps is as follows,

- Liquid enters the suction port between the rotor (large exterior gear) and idler (small interior gear) teeth.

- Liquid travels through the pump between the teeth of the "gear-within-a-gear" principle. The crescent shape divides the liquid and acts as a seal between the suction and discharge ports.

- The pump head is now nearly flooded, just prior to forcing the liquid out of the discharge port. Inter-meshing gears of the idler and rotor form locked pockets for the liquid which assures volume control.

- Rotor and idler teeth mesh completely to form a seal equidistant from the discharge and suction ports. This seal forces the liquid out of the discharge port.

Advantages of internal gear pumps are,

- Excellent for high-viscosity liquids
- Constant and even discharge regardless of pressure conditions and non-pulsating discharge
- Only two moving parts, they are reliable, simple to operate, and easy to maintain
- Operates well in either direction
Chapter 5. Design of heat transport system

- Single adjustable end clearance
- Low NPSH (Net positive suction head) required
- Self priming

5.7 Heat exchanger

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. Typical applications involve transferring of heat from one fluid at higher temperature to another fluid at lower temperature. In many heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix or leak. Such exchangers are referred to as direct transfer type, or simply recuperators. In contrast, exchangers in which there is intermittent heat exchange between the hot and cold fluids via thermal energy storage and release through the exchanger surface are referred to as indirect transfer type, or simply regenerators [124, 125].

The heat transfer surface is a surface of the exchanger core that is in direct contact with fluids and through which heat is transferred by conduction. That portion of the surface that is in direct contact with both the hot and cold fluids and transfers heat between them is referred to as the primary or direct surface. To increase the heat transfer area, appendages may be intimately connected to the primary surface to provide an extended, secondary, or indirect surface. These extended surface elements are referred to as fins. Thus, heat is conducted through the fin and convected (and/or radiated) from the fin (through the surface area) to the surrounding fluid, or vice versa, depending on whether the fin is being cooled or heated. As a result, the addition of fins to the primary surface reduces the thermal resistance on that side and thereby increases the total heat transfer from the surface for the same temperature difference.

Classification of heat exchangers can be done in various ways. According to the heat transfer process, it can be mainly classified as indirect contact type and direct contact type. Depending on the number of fluids, two, three or more than three fluid type of heat exchangers exist. There are different types like tubular, plate-type, extended surface and regenerative type that are based on the physical construction. Fluid can be passed
through the exchanger once or multiple times. Different flow arrangements are possible like counter-flow, parallel-flow, cross-flow, split-flow and divided flow.

In the proposed system, the heat exchanger is used to transfer heat from the fluid at higher temperature to the food which is being cooked. The ideal characteristics for the heat exchanger for this application are,

- Temperature difference between the circulating fluid and the food has to be maximum to have higher heat transfer
- For a given temperature difference, design of heat exchanger has to be such that the energy flow has to be maximum
- Area of conductive barrier has to be maximized
- Pressure drop across the heat exchanger has to be minimized to reduce burden on pump-II
- Heat exchanger size and dimension have to be flexible with different types of vessels
- Thermal insulation has to be very good to prevent heat loss to ambient

5.8 Monitoring and control system

Temperatures at different points are monitored using sensors. Flow meters are used to measure the fluid flow rate on the collector and the load side. Control circuits are used to track the sun, maximum power point etc. This section describes selection and operation of different monitoring and control systems used in the proposed cooking system.

5.8.1 Temperature measurement

Solar cooking application requires temperature sensors to sense the temperature at different points like fluid temperatures at inlet and outlet of the collector, heat storage tank, heat exchanger etc. Temperature sensor which measures temperature up to 200°C is required with accuracy of 1°C. Sensor has to be fixed in the system such that temperature of the fluid is measured when it comes in contact with the fluid.
Chapter 5. Design of heat transport system

5.8.1.1 Different types of sensors

Thermistor: This is also known as resistance thermometers or resistance temperature detectors (RTD) like PRT (Platinum resistance thermometer). This makes use of the property of materials by which change in the electrical resistance of the material with changing temperature can be predicted. This can be used to measure temperature up to 600°C with high accuracy. This has wide operating range but very less sensitive to small temperature change.

Silicon bandgap temperature sensor: This is commonly used temperature sensor in electronic equipments. This is available in a silicon integrated circuit (IC) at a very low cost. They operate as a 2-terminal Zener and the breakdown voltage is directly proportional to the absolute temperature giving 10mV/K. Some of the precision temperature sensor ICs are LM135, LM235 and LM335. Temperature can be measured accurately using these sensors up to 100°C.

Thermocouple: A thermocouple is a junction between two different metals that produces a voltage related to a temperature difference between the two metals. Wide range of temperatures can be measured.

Standard thermocouples used for temperature sensing are made of specific alloys, which in combination have a predictable temperature-voltage relationship. For different temperature ranges specific alloys are used. Thermocouples are standardized against a reference temperature of 0°C. Cold junction compensation is used to adjust the output voltage according to the ambient temperature.

K-type (Chromel-Alumel junction) thermocouple is used in our application. Another important requirement is that the thermocouple used should be ungrounded type which should not pickup noise when it comes in contact with metallic parts. For signal conditioning and cold junction compensation, AD595A IC is used as shown in figure 5.5. AD595 is a complete instrumentation amplifier and thermocouple cold junction compensator on a monolithic chip. It combines an ice point reference with a pre-calibrated amplifier to produce a 10 mV/°C output directly from a thermocouple signal.
5.8.2 Flow measurement

Flow rate of the fluid through the solar thermal collector is measured using the measuring technique proposed in chapter 4. The flow resistance and range of the pressure that are used in the flow measurement system are selected according to the maximum flow rate requirement in the system. Since maximum flow rate on the collector side and load side are approximately same, two identical flow meters are being used. As per the flow rate, flow meters are fabricated and calibrated.

5.8.3 Control system

5.8.3.1 Tracking of Solar Collector

The main purpose of the tracking system is to track the sun continuously during daytime. The tracking system consists of linear actuator, accelerometer sensor and a controller. Linear actuator is a device that applies force in a linear manner. Generally this consists of a rotating motor, from which angular rotation is converted to linear motion using some kind of gear system like lead screw or ball screw. One end of the linear actuator is connected to the solar collector using lever mechanism such that the actuator movement results in the collector being tilted in east-west plane.

Accelerometer sensor is placed at an angle same as that of the collector. From this sensor, information regarding the tilt angle of the collector is obtained, which is used as
a feedback signal for controlling the linear actuator. Figure 5.6 shows the control block diagram of the tracking system. Initially the collector is placed such that it points in the direction of the sun. This is daily reference reset position. The reference angle is calculated from this initial reset value and with the knowledge of constant rate of movement of the sun at 15° per hour. The actual angle of the collector is measured using the accelerometer sensor and compared with the reference value. This error signal is fed to a controller, which gives the controlling pulses to the power converter. According to the error signal, the control voltage $V_c$ is applied to the linear actuator in order to reduce the error angle.

![Control block of the tracking system](image)

**Figure 5.6:** Control block of the tracking system

Necessary controlling voltage $V_c$ is obtained from the pulses using power converter. The circuit diagram of the power converter used for this application is as shown in figure 5.8. Four MOSFETs are used to obtain variable voltage anywhere between -12V to +12V. Switching pulses are obtained from the digital controller, which is implemented on a micro-controller.

Signals obtained from the micro-controller are either high (3.3V or state 1) or low (0V or state 0). Comparator op-amps convert these signals from 0-3.3V level to 0-12V level. Output of the op-amps drive four MOSFETs as shown in figure 5.8. Since MOSFETs $Q_2$ and $Q_4$ are connected across 12V supply, they should not be switched ON simultaneously. Similarly MOSFETs $Q_1$ and $Q_3$ are also operated in complementary fashion. Reference threshold voltage is given to inverting terminal of the op-amps by adjusting R2. This should be a value between 0 and 3.3V. Dead-band logic is provided at the input of the op-amp using resistor, capacitor and diode combination to prevent short circuit of the supply voltage as shown in figure 5.8.

There are four possible switching combinations which are listed in table 5.2. When $Q_2$ and $Q_3$ are ON, a positive voltage is obtained at the output and when $Q_1$ and $Q_4$ are ON, a negative voltage (-12V) is obtained. Other two states given in the table give 0V and...
are used for free wheeling of current. Figures 5.7(a) and 5.7(b) show the pulses during forward and reverse operation. $T_1$ and $T_2$ represent PWM pulse given at the input. $Q_1$ to $Q_4$ shows the state of the switches. Effective average voltage in one switching cycle can be varied by varying the time duration for which output is kept in active state of either 12V or -12V. For example, if 6V is required, then for 50% of the time +12V is applied and for the remaining 50% of the time 0V as shown in the figure 5.7(a). As the time duration of active state is increased, higher voltage can be obtained. Similarly, the output voltage can be varied anywhere from 0 to -12V by keeping S1 ON and switching S2. This is called pulse width modulation (PWM).

When the error signal is a positive value, then the controller has to give pulses such that the collector has to rotate in forward direction (east-west rotation is considered as forward and vice versa) such that it will result in reduction of error. Duty ratio is defined as the ratio of the time duration of active state to total time period of the switching cycle. Duty ratio, for which the active state is applied, is calculated depending on the magnitude of the error. Higher the magnitude of error, higher is the duty ratio. The collector is rotated
### Table 5.2: Different possible switching states

<table>
<thead>
<tr>
<th>$S_1$</th>
<th>$S_2$</th>
<th>$Q_1$</th>
<th>$Q_2$</th>
<th>$Q_3$</th>
<th>$Q_4$</th>
<th>Output Voltage</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>ON</td>
<td>ON</td>
<td>OFF</td>
<td>OFF</td>
<td>0</td>
</tr>
<tr>
<td>0</td>
<td>1</td>
<td>OFF</td>
<td>ON</td>
<td>ON</td>
<td>OFF</td>
<td>+12V</td>
</tr>
<tr>
<td>1</td>
<td>0</td>
<td>ON</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
<td>-12V</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
<td>ON</td>
<td>0</td>
</tr>
</tbody>
</table>

in the reverse direction when the error signal is negative in order to bring the error to zero.

![Circuit Diagram of the Power Converter](image)

**Figure 5.8:** Circuit diagram of the power converter

#### 5.8.3.2 Power Supply for pumps

As discussed in section 2.3.3, circulating pumps are controlled in a manner to vary the flow rate in the system to achieve optimization of energy. Figure 5.9 shows the control block diagram for MPPT. Energy obtained from the sun is calculated using temperature data and flow rate. Voltage supplied to the pump and current drawn gives the additional power consumed to circulate the fluid. Energy optimization algorithm calculates
the voltage that has to be applied to the pump as shown in figure 2.12. A chopper drive is designed to generate voltage corresponding to the signal obtained from the controller.

The chopper drive schematic is shown in figure 5.10. Variable DC voltage is obtained from a constant DC bus voltage of $V_{dc}$ by switching the voltage between $V_{dc}$ and zero. The motor is coupled to drive the pump. By varying the motor voltage, flow rate of the pump can be controlled.

Figure 5.11 shows the chopper circuit to drive the pumps. A MOSFET is used to switch the output voltage between zero and full DC voltage obtained from rectifying 230V AC voltage. When the MOSFET is turned ON, the voltage across the motor terminals is full DC voltage and when the MOSFET is turned OFF, voltage across the motor terminals is zero as motor current freewheels through the diode D1. By varying the ON-time, the motor voltage can be varied. TL494 is used to generate pulses to control the MOSFET. Voltage can be set either by external signal $V_c$ at J2 or through potentiometer R2. Load current is sensed using R7 and limited by setting R10. LM7815 voltage regulator supplies
15V required for the controller IC.

\[ 
\text{Figure 5.11: Circuit diagram of the power supply for pump} 
\]

### 5.9 Experimental Results

#### 5.9.1 Experimental setup

Figure 5.1 shows the block diagram of the experimental setup for solar cooking system. Paraboloid dish concentrator is used to focus sun rays onto the receiver. Aluminium sheets are used as reflecting material. To improve optical efficiency, surface of the reflector is anodized. A linear actuator is fixed to the paraboloid with a lever system in such a
Chapter 5. Design of heat transport system

way that when actuator moves to and fro, the paraboloid is rotated in east-west direction. Using an accelerometer sensor that is fixed on the paraboloid, the tilt angle is sensed. Early morning, the concentrator is fixed toward sun manually. The sensor considers this as the reference angle and tracks at a constant rate of 15° angle per hour.

A coil made of copper tube is placed at the focus of the parabola in order to receive the heat. Servo-therm oil is circulated through the collector to absorb heat energy. Stainless steel pipe with glass wool insulation over that is used for circulation of the oil from the collector to the tank. Thermocouples are placed to measure temperature of oil entering and leaving the receiver. A rotary pump is used to circulate oil through the receiver and put back into the heat storage tank. The pumps are driven by permanent magnet DC (PMDC) motors, which are controlled by variable voltage power supplies. The heat storage tank is made of stainless steel material with good thermal insulation around that for better retention of heat. The flow rate of the fluid is measured using a flow meter proposed in chapter 4.

On the load-side, hot oil from top of the heat storage tank is taken to the kitchen through thermally insulated stainless steel tube. Heat is transferred from oil at higher temperature to cooking load using a heat exchanger. Helical shaped coil of copper is wound around the cooking vessel with thermal insulation to constrain the heat within the heat exchanger. Oil leaving the heat exchanger is pumped back from the kitchen to the buffer tank using another similar pump-motor drive. The mass flow rate on the load-side is measured using another flow meter. Thermocouples are placed at different places as shown in figure to measure the temperatures at various points of the system.

5.9.2 Characteristic of Storage tank

Oil is heated up by circulating it through the collector. Once oil temperature in the storage tank reaches above 100°C, circulation on both the collector side and the load side are stopped. Variation in the storage temperature is measured and plotted along time as shown in figure 5.12. Exponential curve fit is done for \( T_s - T_a \) verses time. The time constant of decay for storage temperature is obtained as 19.34 hours with regression coefficient \( R^2 = 97.51\% \).
Chapter 5. Design of heat transport system

5.9.3 Characteristic of pumps

Rotary pumps are used for circulation of the oil through the solar heat collector and the heat exchanger. Rate of flow of the oil is measured using flow meter. Power required to pump the oil is calculated by measuring the input voltage and the current drawn by the pump. Pumping power is calculated for different flow rates by varying the supply voltage given to the pump. Pumping power is plotted against the flow rate as shown in figure 5.13. For rotary pump, the pumping power is a function of square of the flow rate. A second order polynomial function is fitted to the experimental result as shown in figure 5.13.

Similarly, the pumping power is characterized for pump-2 as shown in figure 5.14.

Figure 5.12: Decay of storage temperature

\[ T_s - T_a = 85.912 \ e^{-0.0517t} \]
\[ R^2 = 97.51\% \]
5.9.4 Performance with constant input power

Experiments are conducted by emulating the solar thermal collector by an electric heater. Input power given to the system can be controlled independent of solar insolation by

\[
P_{p1} = 64248q^2 + 322.43q
\]

\[R^2 = 99.67\%
\]

\[
P_{p2} = 54598q^2 + 27.981q
\]

\[R^2 = 99.61\%
\]
controlling the input voltage supplied to the electric heater. This section presents experimental results for a constant input power.

The oil is circulated through the electric heater while keeping the input power at 1225 W. Figure 5.15 shows the variation in the collected power and the pumping power. Raise in oil temperature of the storage tank is shown in figure 5.16. It is observed from the plot, as the oil temperature increases, the collector efficiency comes down due to increase in the heat loss to the ambient. Increase in the oil temperature decreases the viscosity easing the circulation of the same. Decrease in the pumping power can also be observed from the figure 5.15.

Temperature of the oil in the storage tank is heated up to around 120°C and this heated oil is circulated through the heat exchanger using another rotary pump. Figure 5.17 shows the power at different parts the system. \( P_{so} \) is the power that is being taken out from the storage tank for heating the load and \( P_{he} \) is the power input to the heat exchanger. Power given to the pump to circulated the oil through the heat exchanger is represented as \( P_{p2} \). Initially high power is taken from the storage tank to heat up the pipe running
from terrace to the kitchen and also to raise the temperature of the heat exchanger. As temperature reaches steady value, power also settles in constant value. Figure 5.18 shows the decay of storage tank temperature due to loading.

Figures 5.19 and 5.20 show the performance of the system during both the collector side and the load side circulation. Constant input power of 1600 W is given at the input of the heater. When the storage tank temperature is at 130 °C, the load side circulation is started. The power taken out from the storage tank for heating up of the load is shown as $P_{s_o}$, which increases initially and settles at around 1000 W as the temperature of the pipe and the heat exchanger reaches steady state value. It is observed from the plot that the storage tank temperature decreases during loading. This is due to the reason that the power taken out from the storage tank is higher than the collected power. As the temperature of the oil decreases, the collected power increases. $P_{s_o}$ and $P_{coll}$ converges as time elapses and hence the storage temperature reaches a steady state value gradually. Pumping power $P_{p1}$ and $P_{p2}$ are also shown in the figure.
Figure 5.17: Load power and pumping power

Figure 5.18: Variation in storage tank temperature during loading
Figure 5.19: Load power and pumping power

Figure 5.20: Variation in storage tank temperature during loading
5.9.5 Performance with daily insolation variation

This section presents experimental results with variable input power. The input power $P_{in}$ given to the electric heater is equivalent to the solar insolation at the location of experiment on 19th March, 2010. Solar insolation is measured using a pyranometer and voltage supplied to the electric heater is varied accordingly. Solar heater is emulating paraboloid collector of aperture area of 3.75 $m^2$ and efficiency of 40%. Experiments are started at 8.00 in the morning.

Figure 5.21 shows the variation in the collected power as the input power is changed. The collected power decreases due to increase in the oil temperature even though the input power increases. Flow rate of the oil is maintained at around 0.0166 $kg/s$. Power required for the circulation is also shown in the figure. Increase in the buffer temperature is shown in figure 5.22.

Similar experiment is conducted on 22nd March, 2010. Flow rate through the collector is maintained at 0.011 $kg/s$. Variation in $P_{in}$, $P_{coll}$ and $P_{pump}$ are shown in figure

![Figure 5.21: Input power, collector power and pumping power](image-url)
Figure 5.22: Variation in storage tank temperature during heating

5.23. Figure 5.24 shows the temperature of the storage tank. Collector efficiency is calculated as the ratio of the power received at the collector to the input power and plotted in figure 5.25. When the oil temperature is closer to ambient temperature, the collector efficiency is high. As the buffer temperature increases, the collector efficiency decreases due to increase in the heat loss to ambient.
Chapter 5. Design of heat transport system

Figure 5.23: Input power, collector power and pumping power
Figure 5.24: Variation in storage tank temperature during heating
Figure 5.25: Collector efficiency during heating
Figures 5.26 and 5.27 show the experimental results for full day profile. Experiment is carried out from 8:00AM to 5:00PM on 23rd March, 2010. Initially, the oil is heated up till 12:30PM by circulating it only through the collector. When the temperature of the storage tank reached till 160 °C, the load side circulation is enabled. Power taken out from the storage tank and the power input to the heat exchanger are shown in the figure.

![Figure 5.26: Variation in power](image-url)
5.10 Conclusion

The entire cooking system that is proposed is presented in an integrated manner wherein the various sub-systems and components of a solar cooking system are discussed. Solar insolation available at the location of installation is estimated. The governing equations for insolation estimation are presented. Sizing of solar thermal concentrating collector like linear parabolic collector and paraboloid dish collector are presented along with design equations. Sizing of Photovoltaic (PV) panels to supply electrical loads are described along with the governing equation to select the panel rating. Design of heat storage tank dimensions is addressed. Battery is used to store electrical energy obtained from the PV panels. Selection and sizing of the circulating pumps that are used to control the flow rate of the thermic fluid are described in detail. A discussion on the heat exchanger is also included. The electronics for monitoring of different parameters like temperature and flow rate is presented. Control of linear actuator and pumps are explained in detail. The results for such a system is presented and analyzed.

Next chapter discusses the life cycle costing and design toolbox for the proposed solar hybrid cooking system.
Chapter 6

Life Cycle Costing

6.1 Introduction

The detailed design of the cooking system is discussed in the last chapter where only selection and sizing of different subsystems are discussed. While designing a practical system, cost plays a major role which has to be used for the design iteration. The initial portion of the chapter explains the life cycle cost of the proposed cooking system based on the analysis done in the previous chapter. The design procedure of such a system is described in the later portion of the chapter. This is structured and presented as modules that forms the framework for a design toolbox in designing the cooking system for a given set of specifications.

Life cycle costing (LCC) refers to the accumulated worth of all the costs that are both recurring and non-recurring related to a project, during its life span [126, 127]. Life cycle begins with identification of the economic need and ends with retirement and disposal activities. Life cycle can be projected on a functional or an economical basis. The LCC economic model provides better assessment of long-term cost effectiveness of projects than can be obtained with only initial cost decisions.

The life cycle begins at the moment when an idea of a new system is born and finishes when the system is safely disposed. In other words, the life cycle begins with the initial identification of the needs and requirements and extends through planning, research, design, production, evaluation, operation, maintenance, support and its ultimate phase out. LCC are summations of cost estimates from inception to disposal for both
Chapter 6. Life Cycle Costing

equipment and projects as determined by an analytical study. It is the estimate of total
costs experienced in annual time increments during the project life with consideration for
the time value of money. The main objective of LCC analysis is to choose the most cost
effective solution from a group of alternatives to achieve the most inexpensive long-term
cost over the life span.

Payback period of a system is the period of time required for the return on investment to
repay the sum of the original investment. This shows how long it takes to pay for itself.
This can also be used for feasibility between different options. For the same life span,
lower the payback time, more is the profit.

Hence, the purpose of life cycle costing is to estimate the effects of costs over the en-
tire life span of the project. While setting up of the system, iterations have to be carried
out between design and life cycle costing in order to minimize the life cycle cost while
meeting necessary functional requirements. This chapter discusses different parameters of
the LCC and their selection procedure for a solar cooking system with user specifications.

6.2 LCC Parameters

The life cycle costing is an important factor for comparing the alternatives and deciding
on a particular process for completing a project. LCC is calculated considering different
components as given by,

\[ LCC = \text{Capital cost} + \text{Replacement cost} + \text{Maintenance cost} + \text{Energy cost} \]
\[ \quad - \text{Salvage} - \text{Carbon Credit} \]

Capital cost is the initial investment that is spent on purchase and installation of different
components of the system at the beginning of the system life. This includes cost of
equipments required for setting up of the system. Some of the components in the system
need to be replaced at regular time duration, which is included as replacement cost.
Maintenance cost is the cost incurred for annual maintenance of the system. Energy cost
is the money spent on annual conventional energy consumption. Salvage is the money
obtained while disposing the system at the end of the life span. Carbon credit is the
amount earned by this system depending on the amount of mitigation of \( CO_2 \) emission.
Each of these costs is converted to its present worth in order to calculate LCC. Present
worth is calculated by considering interest rate and inflation rate. The various costs involved in the proposed solar cooking system are discussed.

6.2.1 Capital costs

Capital cost would be the money spent on purchasing/fabricating different sub-systems of the cooking system. Following are the sub-systems with their costs:

- **Cost of the concentrator:** In order to estimate the cost of the collector, its size is required. This depends on the solar insolation available at the location and the energy required for cooking. The area of the concentrating collector is calculated as explained in section 5.3.2.1 of the previous chapter. Let $SA$ be the surface area of the collector. The cost varies depending on the type of material used for the reflector and the quantity used. Let $RT_2$ be the cost of material for a unit area. Then total cost of the material is estimated to be equal to $SA \times RT_2$. Let this cost be $CP_1$.

- **Cost of the solar collector, frame and mount:** This includes the cost of the solar thermal receiver kept at the focus of the reflector. Cost for frame and mounting structure is also included in this. Let this cost be $CP_2$, which is calculated proportionate to cost of the concentrator, $CP_1$.

- **Cost of Photovoltaic (PV) panel:** It is essential to calculate the electrical energy requirement of the system to estimate the cost of the PV panel. From the energy requirement, the size of the panel can be calculated as discussed in the design section 5.3.2.2. In general, the cost of PV panel is specified in terms of cost per peak watt. Let $RT_3$ be the cost per peak watt and $P_{pv}$ be the peak power requirement. Hence the total cost can be calculated as $RT_3 \times P_{pv}$. Let this cost be $CP_3$.

- **Cost of PV panel structure, mounting and wiring:** Let this cost be $CP_4$, which includes cost of structure, wiring and mounting for PV panels. This is calculated as a fraction of PV panel cost, $CP_3$.

- **Cost of heat storage tank:** Storage tank is used to store the energy in the form of hot fluid. The cost of this buffer tank includes not only cost of the tank but also the cost of the thermal insulation. The cost depends on the size of the tank. This is obtained according to the amount of energy to be stored. The total material required is calculated from the total surface area of the tank. Let $ST_{SA}$ be the
surface area of the storage tank and \( ST \) be the cost of the material used per unit area. The cost of the tank material is estimated as \( ST-SA \times ST \). For total cost, cost of insulation material also has to be considered. Let \( ST\_insu \) be the cost of insulation per unit surface area of the tank. The cost of the insulation material is estimated as \( ST-SA \times ST\_insu \). Let \( CP5 \) be the total cost for fabrication of the tank.

- **Cost of heat transfer fluid:** In order to calculate the cost of the fluid, total amount of fluid required has to be estimated. The amount of fluid required is almost equal to volume of storage tank as the volume of fluid in pipes is very small compared to this. Let \( Roil \) be the cost of the fluid per unit volume and \( Voil \) be the volume of the fluid required. Total cost is calculated as \( Roil \times Voil \) and let this cost be \( CP6 \).

- **Cost of linear actuator:** Let the cost of linear actuator be \( CP7 \), which is being used for tracking the sun.

- **Cost of pumps:** Let \( CP8 \) and \( CP9 \) be the cost of circulating pump-I and pump-II respectively.

- **Cost of temperature sensors:** Let \( CP10 \) be the cost of temperature sensors and conditioning circuit for them. This depends on the number of temperature sensors used in the system.

- **Cost of heat exchanger:** Cost of the heat exchanger mainly depends on the size and dimension of the vessel used for cooking. This also includes the cost of thermal insulation that is being used around that. Let \( CP11 \) be the total cost of the heat exchanger.

- **Cost of battery:** In order to calculate the cost of battery, size of the battery has to be calculated as given in the design procedure of section 5.5 in previous chapter. Let \( C_{bat} \) be the cost of battery per Ah capacity. According to the capacity requirement, the cost of battery is calculated. Let \( CP12 \) be the cost of the battery and its charging circuit.

- **Cost of flow meter:** Let \( CP13 \) be the cost of the flow meters. This includes cost of fabrication of the meter and pressure sensor.

- **Cost of piping:** It is essential to estimate the length of the pipe from the collector to the storage tank and from the storage tank to the heat exchanger. Let \( CP14 \) be the total cost of the piping which includes cost of pipe and insulation over that.
• Cost of other electronics: Let $CP15$ be the cost of electronics required for monitoring and control purpose.

The total capital cost is calculated by adding all the above costs. Equation (6.2) gives the present worth of all the capital cost $PW1$.

$$PW1 = CP1 + CP2 + CP3 + CP4 + CP5 + CP6 + CP7 + CP8 + CP9$$
$$+ CP10 + CP11 + CP12 + CP13 + CP14 + CP15$$

(6.2)

6.2.2 Replacement costs

Replacement cost is the cost incurred in replacing any of the components or equipments during the system life span. Most of the equipments like solar collector, storage tank, PV panel, linear actuator, pumps, heat exchanger and monitoring instruments are designed to have a life of 25 years or more. Hence these components need not be replaced. But sub-systems like battery which undergo charge-discharge cycle, have shorter life span compared to the system life and hence have to be replaced at regular intervals. Manufacturer of the battery specifies the number of charge-discharge cycle, according to which it has to be replaced. The life of this may be taken as 4 years. If the life costing of the cooking system is performed for 25 years, then the battery has to be replaced once in every 4 years. The old batteries are replaced with new batteries with the rebate of 7% on total cost of battery bank in most of the countries like India [128]. Similarly, the fluid which is being used for heat transfer has limited number of heating-cooling cycles. This also has to be replaced on a regular basis with the rebate of 5% on total cost. Let $R$ be the total replacement cost that has to be spent once in every 4 years. This is calculated as $0.95 \times CP6 + 0.93 \times CP12$. This replacement costs have to be converted to the present worth for calculating LCC. Let their present worth be $PW2, PW3,...$ and $PW6$ respectively.

6.2.3 Maintenance and running costs

Maintenance cost is the money spent on maintaining the system so as to keep it in good operational condition. The proposed solar cooking system is almost maintenance free except the conventional cooking fuel. Amount of conventional energy source required depends on the user specification for which it is designed. Let annual energy cost of conventional source be represented as $EC$. Other than this, a small annual maintenance
cost such as cleaning of solar reflector, PV panel, cleaning of storage tank, heat exchanger and pipe, maintenance of monitoring instruments may be included for LCC analysis. Let this be $AM$. Let $ME$ be the total of annual maintenance cost, $AM$ and energy cost, $EC$. Present worth of this cost be considered as $PW7$.

6.2.4 Salvage

This is also called as residual value, which is the remaining value after it has been fully depreciated. In other words, the residual value could be defined as an estimated amount that an entity can obtain when disposing of an asset after its useful life has ended. This is the sum that is obtained by selling off the different sub-systems after their life time. Let the sum that can be obtained be $S$ and its present worth be $PW8$.

6.2.5 Carbon credit

The carbon credit potential of the proposed system is determined based on the mitigation of $CO_2$ emissions in its life span. Energy conversion from clean energy sources like solar energy is more reliable and environmental friendly compared to conventional energy sources. Utilization of this kind of system has potential to mitigate $CO_2$ emissions. The average intensity of $CO_2$ emission from coal thermal power plant in India is 1.57 kg/kWh \[128\]. Let the total mitigation of $CO_2$ emissions from the cooking system for the life span be $CO2$ in kg. In case of thermal power plants, only 30%-40% of the total thermal input is converted to electricity. Remaining energy is lost to ambient in the form of heat energy causing global climate change. In addition to heat dissipation, large amount of $CO_2$ is emitted. Since solar thermal collector and PV panels are used in the proposed system, large amount of $CO_2$ emission can be prevented.

The amount of carbon credit earned by this system depends on the amount of mitigation of $CO_2$ emission per year. This is represented as $CC$. Carbon credit is considered for the time interval beginning from pay back period to the life span. Present worth of these carbon credits be $PW9$
6.2.6 Payback period

This is the time period required for the return to repay the investment on the system. In the proposed system, the payback period is the time when the total investment on the system equals the amount of money saved on the conventional energy source. Amount of saving is calculated by deducting LCC from the present worth of energy cost, if conventional energy is used in place of the solar cooking system. This is calculated, considering annual cost of conventional energy for the year varying from 1 to life span. Figure 6.1 shows the plot of savings versus year. It can be observed from the figure that after 12 years, amount of saving crosses zero and becomes positive. This shows that the payback period of the system is 12 years. Once, all the investment is returned, whatever energy is obtained from the system accounts for the saving.

6.3 Life Cycle Cost Analysis

Life cycle costing is discussed using cash flow diagram which is shown in figure 6.2. In figure, $PW_1$ is the present worth of the capital cost which is same as capital cost itself at 0th year. $ME$ indicates the total annual maintenance cost and energy cost which occurs every year. $R1, R2...R5$ represent the replacement cost of battery and heat transfer fluid...
at the end of 4th, 8th ..., 20th year. Salvage value is represented as \( S \) at the end of the life span that is at the end of 25th year.

Present worth of each of these cost has to be calculated as explained below.

- **Capital cost:** Since capital cost incurs at 0th year, the present worth is same as the capital cost.

- **Replacement cost:** It is seen from the cash flow diagram that replacement takes place in 4th, 8th.. and 20th year. Present worth is calculated for each of them.

\[
P_{W2} = \frac{R_1 (1 + f)^{N1}}{(1 + i)^{N1}} = \frac{R_1 (1 + f)^{4}}{(1 + i)^{4}} \quad (6.3)
\]

Where \( R_1 \) is the replacement cost, \( i \) is the rate of interest, \( f \) is the rate of inflation and \( N1 \) is the number of years from replacement year to 0th year. Similarly, present worth of replacement cost for different years is calculated by substituting corresponding value for \( N1 \). Present worth \( PW3, PW4.. \) and \( PW6 \) are calculated by replacing \( N1 \) by 8,12..and 20 respectively.

- **Annual maintenance and energy cost:** Let \( ME \) be the total maintenance and energy cost per year. Present worth of each year forms a geometric progression. Sum of present worth of all these costs is calculated at 0th year as,

\[
P_{W7} = ME \times \frac{1 + f}{i - f} \left[ 1 - \frac{(1 + f)^{N2}}{(1 + i)^{N2}} \right] = ME \times \frac{1 + f}{i - f} \left[ 1 - \frac{(1 + f)^{25}}{(1 + i)^{25}} \right] \quad (6.4)
\]

Where \( N2 \) is the life of the system in years.
• Salvage: \( S \) represents the salvage value that is obtained at the end of the life cycle. The present worth of which is calculated at 0th year as given in equation (6.5).

\[
P W_8 = \frac{S (1 + f)^{N_2}}{(1 + i)^{N_2}} = \frac{S (1 + f)^{25}}{(1 + i)^{25}} \tag{6.5}
\]

• Carbon credit: Annual carbon credit value is calculated as in equation (6.6).

\[
CC = 1.57 \times \text{Energy}_\text{solar} \times 365 \times \frac{1650}{1000} \tag{6.6}
\]

Where 1.57 \( \text{kg/kWh} \) represents the average intensity of \( CO_2 \) emission from coal thermal power plant in India. The factor 1650\(Rs/\text{ton}\) (\(33\$/\text{ton}\)) represents the monetary value of one carbon credit for mitigation of 1\(\text{ton}\) of \(CO_2\) emission. Carbon credit is considered for the period from pay back time, \(PB\) to life span, \(N_2\). Present worth of carbon credit for those years is calculated as,

\[
P W_9 = CC \times \left( \frac{1 + f}{1 + i} \right)^{PB} \times \left[ \frac{1 - \left( \frac{1 + f}{1 + i} \right)^{N_2 - PB + 1}}{1 - \left( \frac{1 + f}{1 + i} \right)} \right] \tag{6.7}
\]

The total present worth gives the life cycle costing, which is calculated as given in equation (6.8).

\[
LCC = PW_1 + PW_2 + PW_3 + PW_4 + PW_5 + PW_6 + PW_7 - PW_8 - PW_9 \tag{6.8}
\]

From \(LCC\), annual life cycle costing \(ALCC\) is calculated from the following equation.

\[
ALCC = \frac{LCC}{\left( \frac{1 + f}{1 + i} \right) \left[ 1 - \left( \frac{1 + f}{1 + i} \right)^{N_2} \right]} \tag{6.9}
\]

Annual life cycle costing gives a better understanding on the merits of the system chosen for the task. If this system is supplying \(\text{Energy}_\text{solar}\) amount of energy per day, then per unit cost of the energy can be calculated as in equation (6.10). This cost gives the clear picture of amount of saving as compared to conventional source and feasibility of the system.

\[
cost = \frac{ALCC}{\text{Energy}_\text{solar} \times 365} \tag{6.10}
\]
Figure 6.3 shows the cost of unit energy for varying percentage of solar fraction. Solar fraction is the ratio of amount of energy obtained from solar to the total energy requirement. Using above equations, cost is calculated for different values solar fraction. For lower solar fraction, per unit cost is very high as there are components, on which fixed amount has to be spent. Above 70%, cost per unit comes below the cost of conventional energy, indicating that the system is more profitable compared to conventional.

6.4 Bill of materials

Bill of materials consists of all the materials used in setting up a solar cooking system. This is a list of raw materials, sub-assemblies, sub-components, components, parts etc and quantities of each needed to get the final product or system. Size and quantity of some of the materials depends on the production and user requirement and some of them are independent of it.

Materials which depend on the energy transfer requirement are,

- Solar Concentrating collector
• PV panel
• Heat storage tank
• Heat transfer fluid
• Pumps
• Heat exchanger
• Battery

Following are the materials that are independent of the energy transfer requirement

• Linear actuator and control circuit
• Temperature sensors and its electronics
• Flow meter
• Pipes and fittings
• Control circuit for the pumps

### 6.5 Design Toolbox

The design toolbox is split into different modules according to physical subsystems in the proposed cooking system. This helps the designer to design all the subsystems based on the requirement and it is easier to scale this to any required size.

Initial modules calculate the energy available at the user defined location and the energy required for cooking. From these data, size and dimension of different components are calculated which are part of the cooking system. This includes solar thermal and PV collector, heat storage tank, pumps, heat exchanger, battery and measuring instruments. Once sizes of different subsystems are calculated, life cycle costing and annual life cycle costing is analyzed.

Each module is discussed in detail in the following sections.
6.5.1 Module 1

Finds annual profile of solar insolation for a given location. From the profile, find the least insolation available \([122]\):

**Inputs:**

- Latitude, 'aLat'
- Day numbers, 'N'

**Outputs:**

- Annual profile of solar insolation in \(kWh/m^2/day\), 'Htc'
- Minimum solar insolation available in the year in \(kWh/m^2/day\), 'Htc_min'

**Design Constants:**

- Coefficients for a polynomial curve fit for precipitated water vapor content B:
  \[
  \begin{pmatrix}
  1.6204 & -0.2291 & -0.0068 \\
  1.6857 & -0.0073 & -0.0020 \\
  -1.2423 & -0.0810 & -0.0010 \\
  0.5626 & 0.0708 & 0.0019 \\
  -1.2140 & 0.0064 & 0.0014 \\
  -0.0990 & -0.0133 & -0.0004 \\
  0.4972 & 0.0605 & 0.0015
  \end{pmatrix}
  \]

- Coefficients for a polynomial curve fit for solar insolation A:
  \[
  \begin{pmatrix}
  0.5563 & 0.0089 & 0.0002 & 0.0743 & -0.0089 \\
  -0.2350 & 0.0119 & 0.0004 & 0.1473 & -0.0237 \\
  -0.1011 & -0.0091 & -0.0004 & 0.1029 & -0.0201 \\
  0.0136 & 0.0041 & 0.0002 & -0.0071 & 0.0010 \\
  0.1300 & -0.0133 & -0.0003 & -0.0848 & 0.0098 \\
  -0.0600 & 0.0048 & 0.0002 & 0.0733 & -0.0132 \\
  0.0970 & 0.0058 & 0.0002 & -0.0282 & 0.0010
  \end{pmatrix}
  \]

- Solar constant \(I_{sc} = 1.367 \text{ kWh/m}^2/\text{day}\)

**Governing Equations:**
• Day number of year = \( N = [15 46 74 105 135 166 196 227 258 288 319 34 9] \);

• Latitude = \( a\text{Lat} \)

\[
X = (a\text{Lat} - 35)
\]

Model for curve fit = \( M1 = [1; X; X^2] \) Fourier Coefficients:

\[
B1 = B(1, 1 : 3) \ast M1
\]

\[
B2 = B(2, 1 : 3) \ast M1
\]

\[
B3 = B(3, 1 : 3) \ast M1
\]

\[
B4 = B(4, 1 : 3) \ast M1
\]

\[
B5 = B(5, 1 : 3) \ast M1
\]

\[
B6 = B(6, 1 : 3) \ast M1
\]

\[
B7 = B(7, 1 : 3) \ast M1
\]

For all months

\{
\[
t = \frac{2\pi}{365} \left( N(month\_number) - 80 \right)
\]

\[
\delta = 23.45 \times \sin \left( 2\pi \left( \frac{N(month\_number) - 80}{365} \right) \right)
\]

\[
\phi = a\text{Lat} \left( \frac{\pi}{180} \right)
\]

\[
\omega_1 = -\tan(\phi) \times \tan \left( \frac{\pi}{180} \delta \right)
\]

\[
\omega = \acos(\omega_1)
\]

\[
I_o = I_{sc} \left[ 1 + 0.033 \cos \left( \frac{2\pi N(month\_number)}{365} \right) \right]
\]

\[
H_o(month\_number) = \frac{24 I_o}{\pi} \times \left[ \cos(\phi) \cos \left( \frac{\pi}{180} \delta \right) \sin(\omega) + \sin(\phi) \omega \sin \left( \frac{\pi}{180} \delta \right) \right]
\]

\[
w(month\_number) = B1 + B2 \sin t + B3 \sin 2t + B4 \sin 3t + B5 \cos t + B6 \cos 2t + B7 \cos 3t
\]

Model for curve fit \( M2 = [1; X; X^2; w(month\_number); w^2(month\_number)] \)

Fourier coefficients:

\[
A1 = A(1, 1 : 3) \ast M2
\]

\[
A2 = A(2, 1 : 3) \ast M2
\]

\[
A3 = A(3, 1 : 3) \ast M2
\]

\[
A4 = A(4, 1 : 3) \ast M2
\]

\[
A5 = A(5, 1 : 3) \ast M2
\]

\[
A6 = A(6, 1 : 3) \ast M2
\]

\[
A7 = A(7, 1 : 3) \ast M2
\]

Clearness Index\((month\_number) = A1 + A2 \sin t + A3 \sin 2t + A4 \sin 3t + A5 \cos t + A6 \cos 2t + A7 \cos 3t
\]

Solar Insolation = \( \text{Clearness Index}(month\_number) \times H_o(month\_number) \)

\}
\[ Minimum_{\text{insolation}} = \min(Solar\ Insolation) \]

### 6.5.2 Module 2

Calculates the daily average cooking load.

**Inputs:**

- number of LPG (Liquified Petroleum Gas) cylinders used per month, 'LPG'
- Percentage of energy saving from LPG, 'Per\_saving'
- Percentage of night cooking, 'Per\_night'

**Outputs:**

- Average energy required per day from LPG in kWh/day, 'Energy\_LPG'
- Average energy required per day from solar in kWh/day, 'Energy\_solar'
- Average energy required for night cooking from solar per day in kWh/day, 'Energy\_night'

**Design Constants:**

- Weight of LPG in a cylinder, 'LPG\_kg' in kg
- Efficiency of the LPG stoves, 'LPG\_eff'

**Governing Equations:**

- \[ LPG\_\text{day} = \frac{LPG\times LPG\_kg}{30} \]
- \[ Energy\_LPG = \frac{LPG\_\text{day} \times LPG\_\text{calorific} \times LPG\_eff}{1000 \times 3600} \times \frac{100 - Per\_saving}{100} \]
- \[ Energy\_solar = \frac{LPG\_\text{day} \times LPG\_\text{calorific} \times LPG\_eff}{1000 \times 3600} \times \frac{Per\_saving}{100} \]
- \[ Energy\_night = \frac{Energy\_solar \times Per\_night}{100} \]
6.5.3 Module 3

Calculates the area of the parabolic solar thermal collector

Inputs:

- Average energy required per day from solar in \( kWh/day \), 'Energy_solar'
- Minimum solar insolation available in the year in \( kWh/m^2/day \) from module-1, 'Htc_min'

Outputs:

- Area of the parabolic collector required in \( m^2 \), 'app_area'

Design Constants:

- Percentage of direct radiation assumed to be 70%, 'direct_radiation'
- Parabolic collector efficiency assumed to be 50% [129], 'eff_collector'

Governing Equations:

\[
\text{Energy}_{\text{collector}} = \frac{\text{Energy}_{\text{required}}}{\text{eff}_{\text{collector}} \times \text{direct}_{\text{radiation}}}
\]

\[
\text{app}_{\text{area}} = \frac{\text{Energy}_{\text{collector}}}{\text{Htc}_{\text{min}}}
\]

6.5.4 Module 4

Calculates volume of the heat storage tank required

Inputs:

- Average energy required for night cooking per day in \( kWh/day \) from module-2, 'Energy_night'
- Minimum solar insolation available in the year in \( kWh/m^2/day \), 'Htc_min'
- Maximum storage temperature in \( K \) or \( C \), 'T_max'
Chapter 6. *Life Cycle Costing*

- Minimum storage temperature in $K$ or $C$, 'T_min'

**Outputs:**

- Size of the storage tank in $m^3$, 'ST_vol'

**Design Constants:**

- Specific heat of the circulating fluid in $J/kgK$, 'C_p'
- Efficiency of storage tank (storing and taking out energy) assumed to be 70%, 'eff_storage'
- Density of the circulating fluid in $kg/m^3$, 'density'

**Governing Equations:**

- \[ \Delta T = T_{max} - T_{min} \]
- \[ ST_{vol} = \frac{Energy\_night\times3600\times1000}{C_p\times\Delta T\times eff\_storage\times density} \]

6.5.5 Module 5

Calculates the power rating of the pumps

**Inputs:**

- Maximum flow rate of the collector side circulation in $m^3/s$, 'Q1'
- Delivery head of the collector side in $m$, 'hd1'
- Maximum flow rate of the load side circulation in $m^3/s$, 'Q2'
- Delivery head of the load side in $m$, 'hd2'

**Outputs:**

- Power rating of the collector side centrifugal pump, 'P_2'
- Power rating of the load side centrifugal pump, 'P_3'
Design Constants:

- Efficiency of the pump (30%), 'eff_cp'
- Density of the circulating fluid in $kg/m^3$, 'density'

Governing Equations:

- $power_{cp1} = \rho \times Q1 \times g \times h d1$
- $power_{cp2} = \rho \times Q2 \times g \times h d2$
- $cpower\_rating1 = \frac{power\_cp1}{eff\_cp}$
- $cpower\_rating2 = \frac{power\_cp2}{eff\_cp}$

6.5.6 Module 6

Calculates the area of the PV panel required

Inputs:

- Power rating of linear actuator in W, 'P1'
- Number of hours linear actuator is powered per day, 'h1'
- Power rating of centrifugal pump-1 in W, 'P2'
- Number of hours pump-1 is powered per day, 'h2'
- Power rating of centrifugal pump-2 in W, 'P3'
- Number of hours pump-2 is powered per day, 'h3'
- Power rating of electronic control circuits in W, 'P4'
- Number of hours the electronic circuits are ON, 'h4'
- Number of no Sun days, 'd_ns'

Outputs:

- Total energy required from PV in Wh, 'E_rpv'
Chapter 6. *Life Cycle Costing*

- Area of the PV panel in $m^2$, 'area_pv'
- Energy required during no-sun days in Wh, 'W3'

**Design Constants:**

- Efficiency of PV panel (12%), 'eff_PV'
- Efficiency of battery charging and discharge circuits (70%), 'eff_battery'

**Governing Equations:**

- $Energy\_required\_daytime = W1 = P1 \times h1 + P2 \times h2 + P3 \times h3 \times \frac{(100 - Per\_night)}{100} + P4 \times h4$
- $Energy\_required\_nighttime = W2 = \frac{P3 \times h3 \times Per\_night}{eff\_battery \times 100}$
- $Energy\_required\_no\_sun\_days = W3 = (W1 \times d\_ns)$
- $Energy\_required\_Total = E\_rpv = W1 + W2 + \frac{W3}{eff\_battery}$
- $Energy\_available = E\_apv = Htc\_min \times eff\_PV$
- $area\_pv = \frac{E\_rpv}{1000 \times E\_apv}$

### 6.5.7 Module 7

Calculates the capacity of the battery

**Inputs:**

- Power rating of centrifugal pump-2 in $W$, 'P3'
- Number of hours pump-2 is powered per day, 'h3'
- Power rating of electronic control circuits in $W$, 'P4'
- Number of hours the electronic circuits are ON, 'h4'
- Energy required during no-sun days in Wh, 'W3'
- Nominal voltage of the battery in volts, 'Vnom'
Chapter 6. Life Cycle Costing

Outputs:

- Capacity of the battery in Amp-hours (Ah), 'battery_capacity'

Design Constants:

- Efficiency of the battery (70%), 'eff_batt'
- Depth of discharge (DOD) of the battery, 'DOD'

Governing Equations:

- \( Wh_{bat} = P3 \times h3 \times \frac{Per_{night}}{100} + P4 \times h4 + W3 \)
- \( battery\_capacity = \frac{Wh_{bat}}{eff\_batt \times DOD \times V_{nom}} \)

6.5.8 Module 8

Calculates the life cycle costing and annual life cycle costing

Inputs:

- Apperture area of the parabolic collector, 'app_area'.
- Reflector material rate in Rs/m², 'RT2'.
- Total energy required from PV in Wh, 'E_rpv'.
- Cost of PV per peak watt, 'RT3'.
- Volume of the heat storage tank, 'ST_vol'.
- Cost of the heat storage tank material per m², 'ST'.
- Cost of insulation for the heat storage tank per m², 'ST_insu'.
- Cost of the circulating fluid per m³, 'Oil'.
- Cost of the linear actuator, 'CP7'.
- Cost of the pump-1, 'CP8'.
Chapter 6. Life Cycle Costing

- Cost of the pump-2, 'CP9'.
- Cost of the temperature sensor, 'CP10'.
- Cost of the heat exchanger, 'CP11'.
- Cost of the battery per $Ah$, 'C_bat'.
- Cost of the flow meter, 'CP13'.
- Cost of the pipes and fitting, 'CP14'.
- Cost of the electronic circuits, 'CP15'.
- Annual maintenance cost, 'AM'.
- Salvage value, 'S1'.
- Replacement period, 'N1' years.
- Life cycle period, 'N2' years.
- Capacity of the battery, 'battery_capacity'.
- Cost of conventional energy source in $Rs/kWh$, 'RT4'
- Average energy required per day from LPG in $kWh/day$, 'Energy_LPG'
- Average energy required per day from solar in $kWh/day$, 'Energy_solar'

Outputs:

- Life cycle cost, 'LCC'
- Annual life cycle cost, 'ALCC'
- Cost per unit in $Rs/kWh$, 'cost'

Design Constants:

- Rate of interest, 'i'.
- Rate of inflation, 'f'.
- Focal length to diameter ratio of paraboloid, 'fd1'.

6-20
Diameter to length ratio of the heat storage tank, 'ST_DL'.

Amount of carbon emission from conventional energy in kg/kWh, 'carbon_emission'.

Carbon credit in Rs/ton, 'carbon_credit'.

**Governing Equations:**

- Diameter of the paraboloid $D_p = \sqrt{\frac{4 \text{app area}}{\pi}}$
  Focal length $f_l = fd1 \times D_p$
  Surface area $SA = \frac{8\pi f_l^2}{3} \times \left\{ \left( \frac{D}{4\sqrt{f_l}} \right)^2 + 1 \right\}^\frac{3}{2} - 1$
  Cost of the paraboloid $CP1 = SA \times RT2$

- Cost of the solar collector, frame and mount $CP2 = 0.6 \times CP1$

- Cost of the PV panel $CP3 = P_{-pv} \times RT3$

- Cost of the PV panel structure, mounting and wiring $CP4 = 0.3 \times CP3$

- Diameter of the heat storage tank $D = \sqrt[3]{\frac{4 \times ST_{\_DL} \times ST_{\_vol}}{\pi}}$
  Height of the tank $L = \frac{4 \times ST_{\_vol}}{\pi \times D^2}$
  Surface area of the tank $SA1 = ST_{\_vol} \times \left( \frac{2}{L} + \frac{4}{D} \right)$
  Cost of the tank $CP5 = SA1 \times ST + SA1 \times ST_{\_insu}$

- Cost of the fluid $CP6 = Oil \times ST_{\_vol}$

- Cost of the battery $CP12 = C_{\_bat} \times battery_{\_capacity}$

- Annual energy cost $EC = Energy_{\_LPG} \times RT4 \times 365$

- Total annual maintenance and energy cost $ME = AM + EC$

- Annual Carbon credit $CC = Energy_{\_solar} \times carbon_{\_emission} \times 365 \times \frac{carbon_{\_credit}}{1000}$

- Replacement cost $R = 0.93 \times CP12 + 0.95 \times CP6$

- Total capital cost $K1 = CP1 + CP2 + CP3 + CP4 + CP5 + CP6 + CP7 + CP8 + CP9 + CP10 + CP11 + CP12 + CP13 + CP14 + CP15$

- Present worth of capital cost $P1 = K1$
Chapter 6. Life Cycle Costing

- Present worth of replacement cost of battery after N1 years \( P_2 = R \times \left( \frac{1+f}{1+i} \right)^{N1} \)
- Present worth of replacement cost of battery after 2*N1 years \( P_3 = R \times \left( \frac{1+f}{1+i} \right)^{2N1} \)
  ...........
- Present worth of replacement cost of battery after 5*N1 years \( P_6 = R \times \left( \frac{1+f}{1+i} \right)^{5N1} \)
- Present worth of annual maintenance cost \( P_7 = ME \times \frac{1+f}{i-f} \times \left( 1 - \frac{(1+f)^N}{(1+i)^N} \right) \)
- Present worth of the salvage value \( P_8 = S1 \times \frac{(1+f)^N}{(1+i)^N} \)
- Carbon credit value \( P_9 = CC \times \left( \frac{1+f}{1+i} \right)^{PB} \times \left[ \frac{1-(\frac{1+f}{1+i})^{N2-PB}}{1-(\frac{1+f}{1+i})} \right] \)
- \( LCC = P1 + P2 + P3 + P4 + P5 + P6 + P7 - P8 - P9 \)
- \( ALCC = \frac{LCC}{(\frac{1+f}{1+i})[1-(\frac{1+f}{1+i})^{N2}]} \)
- Cost of unit energy from the system \( \text{cost} = \frac{ALCC}{\text{Energy_solar} \times 365} \)

6.6 Toolbox structure

Programs for the above explained toolbox have been written in Matlab. The main program of the toolbox is in a file called system_design.m. This program calls the following function for calculating different design parameters of the system. All the m-files are given in appendix B. Order of the functions called are given below,

1. spec: Calls spec.m file containing specifications of the system
2. \([Htc\_min, Htc, N] = insolation(N, aLat)\): Calls insolation.m module
3. \([\text{Energy\_LPG}, \text{Energy\_solar}, \text{Energy\_night}] = \text{loading}(\text{LPG, Per\_saving, Per\_night})\): Calls loading.m module
4. \([\text{area\_collector}] = \text{collector\_sizing}(\text{Energy\_solar, Htc\_min})\):
   Calls collector_sizing.m module
5. \([\text{storage\_size}] = \text{storage\_sizing}(\text{Energy\_night, Htc\_min, T\_max, T\_min})\):
   Calls storage_sizing.m module
6. \([\text{cpower\_rating1, cpower\_rating2}] = \text{cp\_rating}(Q1, hd1, Q2, hd2)\): Calls cp_rating.m module
Chapter 6. *Life Cycle Costing*

7. \[E_{rpv, area\_pv, W3} = pv\_sizing(P1, h1, P2, h2, P3, h3, P4, h4, d\_ns, Htc_{min}, Per\_night)\): Calls \textit{pv\_sizing.m} module

8. \[\text{battery\_capacity} = battery\_sizing(P3, h3, P4, h4, W3, V_{nom}, Per\_night)\): Calls \textit{battery\_sizing.m} module

9. \[LCC, ALCC, cost = system\_costing(LCC\_spec)\): Calls \textit{system\_costing.m} module

### 6.7 Sample design

Specifications:

**Insolation module:**
Days of the year for insolation estimation, \(N = [15 46 74 105 135 166 196 227 258 288 319 349]\)
Latitude of the location, in degrees=13

**Loading module:**
Number of LPG cylinders used per month=1
Percentage of energy saving from LPG=90
Percentage of night cooking=50

**Storage tank sizing module:**
Maximum storage temperature in C, \(T_{\text{max}}=200\)
Minimum storage temperature in C, \(T_{\text{min}}=80\)

**Pump power rating module:**
Maximum flow rate of the collector side circulation in \(m^3/s, Q_1=2/60000\)
Delivery head of collector side in meters, \(hd_1=2\)
Maximum flow rate of the load side circulation in \(m^3/s, Q_2=2/60000\)
Delivery head of load side in meters, \(hd_2=10\)

**PV sizing module:**
Power rating of linear actuator in W, \(P_1=50\)
Number of hours linear actuator is powered per day in hours, \(h_1=1/60\)
Chapter 6. Life Cycle Costing

Number of hours pump-1 is powered per day in hours, $h_2 = 8$
Number of hours pump-2 is powered per day in hours, $h_3 = 2$
Power rating of electronic control circuits in W, $P_4 = 10$
Number of hours the electronic circuits are ON in hours, $h_4 = 8$
Number of no Sun days, $d_{ns} = 0$

Battery sizing module:
Number of hours pump-2 is powered per day in hours, $h_3 = 2$
Power rating of electronic control circuits in W, $P_4 = 10$
Number of hours the electronic circuits are ON in hours, $h_4 = 8$
Percentage of night cooking, $P_{er_{night}} = 50$
Nominal voltage of the battery in volts, $V_{nom} = 12$

Specifications for LCC module:
Cost of reflector material used in paraboloid in $Rs/m^2$, $RT_2 = 500$
Cost of PV per peak watt in $Rs$, $RT_3 = 120$
Cost of the heat storage tank per $m^2$ in $Rs$, $ST = 15e3$
Cost of insulation for the heat storage tank per $m^2$ in $Rs$, $ST_{insu} = 4e3$
Cost of circulating fluid per $m^3$ in $Rs$, Oil = $80*1000$
Cost of linear actuator in $Rs$, $CP_7 = 5000$
Cost of pump-1 in $Rs$, $CP_8 = 8000$
Cost of pump-2 in $Rs$, $CP_9 = 8000$
Cost of temperature sensors in $Rs$, $CP_{10} = 500*6$
Cost of heat exchanger in $Rs$, $CP_{11} = 5000$
Cost of battery per $Ah$ in $Rs$, $C_{bat} = 50$
Cost of flow meter in $Rs$, $CP_{13} = 2000*2$
Piping cost for 20m in $Rs$, $CP_{14} = 300*20$
Cost of other electronics in $Rs$, $CP_{15} = 1000$
Annual maintenance cost in $Rs$, $AM = 2000$
Salvage in $Rs$, $S_1 = 25000$
Replacement interval in years, $N_1 = 4$
Life of the system in years, $N_2 = 25$
Cost of conventional energy source in $Rs/kWh$, $RT_4 = 7$

Design Results:
Minimum solar insolation over the year in $kwh/m^2/day$, $Htc_{min} = 4.7729$
Aperture area of paraboloid collector in \( m^2 \), \( \text{app\textunderscore area} = 2.14 \)
Volume of storage tank in \( m^3 \), \( \text{ST\textunderscore vol} = 0.0402 \)
Power rating of pump-1 in \( W \), \( P_2 = 1.96 \)
Power rating of pump-2 in \( W \), \( P_3 = 9.81 \)
Area of PV panel in \( m^2 \), \( \text{area\textunderscore pv} = 0.0689 \)
Capacity of battery in \( Ah \), \( \text{battery\textunderscore capacity} = 26.73 \)
Capital investment in \( Rs \), \( \text{cap} = 63,130 \)
Life cycle costing in \( Rs \), \( \text{LCC} = 91,487 \)
Annual life cycle costing in \( Rs \), \( \text{ALCC} = 5,808 \)
Cost per unit energy in \( Rs/kW\text{h} \), \( \text{cost} = 4.485 \)
Amount of savings per year in \( Rs \), \( \text{save} = 3,257 \)

In a typical Indian household of five members, one LPG cylinder lasts approximately a month. One LPG cylinder containing 14 \( kg \) of LPG contains 197 \( kW\text{h} \) of energy. Thus, in a month 118 \( kW\text{h} \) of energy is utilized for cooking purposes by a typical Indian household considering efficiency of LPG stove as 60%. At an average irradiance of 4.77 \( kW\text{h}/m^2/day \), the amount of energy incident on the surface for a month of 30 days is 306 \( kW\text{h} \). Using a paraboloid having cross section area of around 2.14 \( m^2 \) 90% of LPG can be saved. From the sample design example, capital cost came up to 63,000 \( Rs \). Annual life cycle cost of the system is 5,808 \( Rs \) making per unit cost as 4.5 \( Rs \) which is much below the commercial price of LPG (which turns out to be more than 7 \( Rs \) per unit). For the above mentioned system, annual saving turns out to be 3,257 \( Rs \) having pay back time of 12 years.

6.8 Conclusion

One of the important measures of a renewable energy system is the estimation of the payback period. This chapter presents the life cycle costing approach to estimate the energy payback period. The different components of the life cycle costing are presented in this chapter. It should be noted that a significant issue with life cycle costing is that it is closely linked with the design of the system to a given set of specifications. This implies that system design and life cycle costing must be performed together in order to estimate a reasonably accurate energy cost. As a consequence, a design toolbox has been developed specifically to design the proposed hybrid solar cooking system with life
cycle costing analysis integrated within it. A general procedure for calculating LCC is explained in detail. The bill of materials for setting up of a solar cooking system is listed with classifying them into a. one dependent on the production requirement and b. the other independent of the production requirement. This results in a life cycle costing that can be scaled to the same solar cooking system of different capacities.

The latter part of the chapter focuses on the design toolbox. Calculation of life cycle costing is formulated and structured into various interlinked modules. The design toolbox also integrates within it the estimation of the carbon credits and its reflection on the overall energy costs. This chapter presents detailed inputs, outputs, design constraints and the governing equations for each module. This toolbox is used to for designing a system for a sample specification. This sample design shows the design parameters for a typical home requirement.

The thesis is concluded in the next chapter.
Chapter 7

Conclusion and Future Scope

7.1 Summary

Due to exponential raise in the world’s population and resulting growth of industrial activities, the energy requirements are such that fossil fuel cannot be the only source on a sustainable basis. This implies that one has to look for alternate sources of energy which are more environmental friendly, cleaner and renewable. As cooking is an integral part of every human being, it consumes a major fraction of the household energy requirement. Existing cooking fuels are either derived from fossil fuels like LPG, kerosene etc. or air polluting like firewood, crop residue, cow dung etc. In this respect, solar cooking is a very simple, clean and environment friendly alternative.

Till now, solar based cooking has never been a strong contender in the commercial market or even close to being a preferred method of cooking. They have been relegated to demonstration appliances to show case the solar based concepts. In this mode, cooking is no longer a time independent activity that can be performed at any time of day. One is forced to cook only at certain times when there is sufficient insolation. The geography of the cooking activity also shifts away from the kitchen. The kitchen is no longer the hearth of the home as the actual cooking activity shifts to the roof tops or high insolation platforms. This further adds to the inconvenience apart from being unable to cook at night or during cloudy conditions or during most of the winter days.

This thesis proposes a method alleviate few of the problems faced in solar based cooking. The objective here is to transfer solar thermal energy to the kitchen to act as a supplement
for the conventional source of energy like the LPG or other sources that are traditionally being used in the households. The method proposed is in fact a hybrid scenario wherein the solar thermal is used to supplement the traditional source. Solar photovoltaic cells are also used to power the electronics and apparatus proposed in this thesis. This thesis addresses in detail the issues in analysis, modeling, designing and fabrication for the proposed hybrid solar cooking topology.

The main goal of the proposed system is to transfer heat from sun to the cooking load that is located in the kitchen. The topology includes an additional feature for storing the energy in a buffer. The heat is first transferred from the solar thermal collector to a heat storage tank (that acts as the buffer) by circulating the heat transfer fluid at a specific flow rate that is controlled by a pump. The stored heat energy that is collected in the buffer is directed into the kitchen by circulating the heat transfer fluid into the heat exchanger, located in the kitchen. This is accomplished by controlling the flow rate using another pump. This can be implemented for any scale from small scale household to large scale community level application. The significant contributions of the research work in this thesis are summarized below.

- **Energy transport**: Major challenge in energy transport is to bring heat energy obtained from sun to the kitchen for cooking. Energy transferred from solar insolation to the cooking load has to be optimized to maximize the overall efficiency. This is split into two parts, (a) optimizing efficiency of energy transferred from the collector to the energy buffer tank, (b) optimizing efficiency of energy transferred from the buffer tank to the load. The optimization is performed by means of a maximum power point tracking (MPPT) algorithm for a specific performance index.

The method proposed in this thesis is based on the hybrid utilization of solar energy wherein the solar thermal is used to supplement the conventional or auxiliary source of energy and the solar photovoltaic cells that are used to power the pump drives, tracking and monitoring electronics. In order to maximize the energy transferred to the load; this thesis discusses the concept of maximum power point tracking as a means to optimize the flow rate of the thermic fluid. The concept of maximum power point tracking ensures continuous and dynamic control of the thermic fluid flow rate to maximize the energy collected from the sun. Additionally, the choice of the pipe diameter is also an exercise in solving conflicting constraints viz. heat loss.
from the pipe surface area and reduction in pumping effort with increase in pipe diameter. The concepts of optimization for the proposed system have been verified by simulation and experiment.

- **Modeling of cooking system:** There are several domains that exist in the solar cooking system such as electrical domain, thermal domain, and hydraulic domain. The analysis of power/energy flow across all these domains presents a challenging task in developing a model of the hybrid cooking system.

The proposed system is modeled by primarily using the method of bond graphs which is a multi-energy domain modeling approach. The system comprises of energy interactions in the hydraulic domain which is interlinked with the energy interactions in the thermal domain. The thermal domain further complicates the issue as it includes non-linearities. The bond graph model inherently handles all these energy transactions. The energy storage and energy loss based on the entropy flow rate are modeled as field elements. Without loss of generality, all elements including the field elements always maintain the fundamental energy conservation principle. The thermal domain is modeled primarily by using R-fields and C-fields. The dynamic model is a non-linear model and it includes the dynamics of both the hydraulic and thermal domains.

This developed model of the proposed system is simulated and validated through experimentation. The effects of the flow rate and pipe diameter on the solar power collected and the utilized power are discussed. These results are shown by simulating the bond graph model. The method to estimate the various parameters of the practical system is explained. The simulation using these estimated parameters closely match with the experimental results thereby validating the proposed bond graph model. Thus the proposed mathematical model can be used for both analysis and synthesis of a hybrid solar cooking system with different specifications.

- **Flow measurement:** A novel method of low fluid mass flow measurement by compensating the pressure drop across the ends of measuring unit using a compensating pump has been proposed. The pressure drop due to flow is balanced by feedback control loop. This is a null-deflection type of measurement. As insertion of such a measuring unit does not affect the functioning of the systems, this is also a non-disruptive flow measurement method. This allows the measurement of very
low flow rate at very low resolution. Implementation and design of such a unit are discussed. The system is modeled using bond graph technique and then simulated. The flow meter is fabricated and the model is experimentally validated.

- **Design Toolbox:** Design of hybrid cooking system involves design of multi domain systems. The design becomes much more complex if the energy source to operate the system is hybrid solar based. The energy budget has to be evaluated considering the worst case conditions for the availability of the solar energy. The design toolbox helps in assessing the user requirement and help designing the cooking system to fulfill the user requirement. A detailed toolbox is proposed to be developed that can be used in designing/selecting sub-systems like collector, concentrator, tracking system, buffer tank, heat exchanger, PV panel, batteries etc. The toolbox can also be used for performing life cycle costing.

### 7.2 Conclusion

Fossil fuel based energy sources are very crucial with depleting limited resources available for future. The gap between the energy demand and the available resources is increasing exponentially due to the fast industrial growth and the population explosion. Green house gases emitted from burning fossil fuels not only present health hazards, but also cause increase in the temperature of earth’s atmosphere. In this regard, utilization of renewable energy sources like solar, wind, hydro, tidal, wave etc present a possibility to reduce the entropy rate. Solar cooking proposed in this thesis is one of the long term solutions for the issues mentioned above.

Solar energy is available abundantly in and around the equatorial region and it is free of cost. Solar cooking gives not only relief from the indoor air pollution but also decreases felling of trees for cooking purpose. In urban areas, LPG, kerosene or electricity is used for cooking which are mainly derived from fossil fuels. Proposed system is going to be a solution for sustainable living without depending much on conventional energy resources.

Existing solar cooking methods are not sophisticated as cooking has to be carried outdoor. Time taken for cooking varies from 2-5 hours depending on the availability of sunlight. The proposed cooking system transports thermal energy from sun directly to the kitchen.
More number of households may be interested to use this type of system as there is a control over the rate of cooking and cooking can be carried out at any time of the day or night. The conventional cooking source is aided by additional energy from the sun reducing the consumption of traditional cooking source.

The solar cooking system proposed in this thesis is not only suitable for household application but also for community level cooking. The solar thermal collector and the storage tank can be combined together for a group of houses and hot fluid outlets can be provided for various nearby houses. This reduces the capital cost of building and setting up of solar collector and buffer tank in individual houses.

Scaling of the existing cooking system can be done by adding extra solar thermal collectors to the heat storage tank. This stored energy can be used for different applications just by tapping the hot oil from the buffer tank. Multiple heat exchangers can also be included on the load side according to the user requirement.

Since maintenance of the cooking system is very less, life of the system is as large as 25-30 years. With automation of the entire system, sophistication level can be improved to reduce inconvenience. Nutrition value of the food is preserved by controlling the temperature of the food at moderate value. Solar cooking method proposed in this thesis is the ideal solution for both rural and urban areas where sufficient sunlight is available.

In a typical Indian household of five members, one LPG cylinder lasts approximately a month. One LPG cylinder containing 14 kg of LPG contains 197 kWh of energy. Thus, in a month 118 kWh of energy is utilized for cooking purposes by a typical Indian household considering efficiency of LPG stove as 60%. At an average irradiance of 4.77 kWh/m²/day, the amount of energy incident on the surface for a month of 30 days is 306 kWh. Using a paraboloid having cross section area of around 2.14 m² 90% of LPG can be saved. From the sample design example, capital cost came up to 63,000 Rs. Annual life cycle cost of the system is 5,808 Rs making per unit cost as 4.5 Rs which is much below the commercial price of LPG (which turns out to be more than 7 Rs per unit). For the above mentioned system, annual saving turns out to be 3,257 Rs having pay back time of 12 years.
Chapter 7. Conclusion and Future Scope

7.3 Future Scope

There are many issues which can be improved in the future design of the solar cooking system which are listed below.

- Design of the heat exchanger can be improved or redone such that cooking can be done using vessels of different sizes without changing the heat exchanger. Provision for using conventional cooking stove along with the heat exchanger should be thought of. For household applications, cooking involves not only boiling, but also frying, stewing, roasting, baking, grilling, braising and other techniques of cooking. Hence, design of the heat exchanger should take care of all these issues.

- Existing solar thermal concentrating collectors can absorb only beam radiation and fails to convert diffused radiation into heat energy. Hence it can not be used during cloudy days. There is a possibility of redesign of thermal collector which can address even diffused radiation. Since diffused radiation is 20-40% of total radiation, efficiency of the collector can be improved a lot by absorbing diffused radiation.

- Better ways of storing thermal energy has to be looked in detail in order to minimize the cost without compromising on the performance. Comparison of performance has to be analyzed for different energy storage techniques.

- Thermal insulator that is put around the pipes and the heat storage tank in the prototype has to be improved such that the temperature drop along the pipe carrying hot fluid is minimized. Even small improvement of thermal insulation significantly increases the overall efficiency of the system.

Concepts and techniques proposed in this thesis can be extended to some other applications. Few of them are listed below.

- Heat energy obtained from the sun is optimized by choosing proper flow rate dynamically. Maximum power point tracking, which is proposed in this thesis, can be applied to any other active solar thermal systems like solar water heater, space
heating/cooling etc. to improve the efficiency of the system.

- Solar cooking system can be combined with other thermal applications. Single thermal collector and heat storage tank can be used for different applications like solar cooking, water heating, desalination, space heating/cooling etc. This can also be extended to industrial food processes and food drying with modifications on the load side.

- Flow measuring technique proposed in the thesis is a novel technique to inexpensively measure very low flow rate without disturbing the system. This has vast applications in many different areas like custody transfer, chemical dosing, biomedical applications etc. Instantaneous mileage of a vehicle can be calculated by measuring fuel flow rate into the IC engine and hence optimal speed at which fuel consumption is minimal can be addressed.
Appendix A

Simulation of Linear Parabolic Collector

% this model is for different flow rate
% this program calculates different parameters like collector efficiency,
% heat removal factor, outlet temperature, instantaneous efficiency..etc. for
% different flow rates of linear parabolic collector

clear
clc

% fixed values
S = 486.03; % incident solar flux absorbed by the plate
G = 705*1.0143+244*.993; % Ib*Rb+Id*Rd
Do = 0.081; % outer diameter of absorber tube
Di = 0.075; % inner diameter of absorber tube
Dco = 0.15; % outer diameter of glass cover
Dci = 0.144; % inner diameter of glass cover
Ta = 25+273.2; % ambient temperature
Tsky = Ta-6;
Tfi = 150+273.2; % inlet fluid temperature
g = 9.81; % gravity constant
W = 1.25; % width of the aperture
L = 3.657; % length of the aperture
Ap = pi*Do*L; % absorber area
CR = (W-Do)/(pi*Do); % concentration ratio
spc1 = (Dci-Do)/2; % spacing between tube and cover
Appendix A. Simulation of Linear Parabolic Collector

\[ V_w = 4; \] wind velocity in m/s
\[ \varepsilon_p = 0.15; \] emissivity of absorber tube surface
\[ \varepsilon_c = 0.88; \] glass cover emissivity
\[ \sigma = 5.67 \times 10^{-8}; \] Stefan-Boltzmann constant
\[ \tau_r = 4; \] tape twist ratio

\[ k_1 = \sigma \pi D_0 / \left( 1/\varepsilon_p + (D_0/D_{ci}) \right) (1/\varepsilon_c - 1); \]
\[ k_2 = \sigma \pi D_{co} \varepsilon_c; \]
\[ k_{pump} = 2.4000 \times 10^6; \] centrifugal pump constant

% properties of dry air
\[ T = [0 10 20 30 40 50 60 70 80 90 100 120 140 160 180 200 250 300 350 400 500 600 700 800 900 1000]; \]
\[ T = T + 273.2; \] Temperature in Kelvin
\[ \rho = [1.293 1.247 1.205 1.165 1.128 1.093 1.06 1.029 1.000 0.972 0.946 0.919 0.898 0.879 0.861 0.844 0.829 0.814 0.801 0.789 0.778 0.768]; \]
\[ C_p = 1000 \times [1.005 1.005 1.005 1.005 1.005 1.005 1.000 0.999 0.999 1.009 1.009 1.009 1.013 1.017 1.022 1.026 1.038 1.047 1.059 1.068 1.093 1.114 1.135 1.156 1.176 1.185]; \]
\[ k = [0.0244 0.0251 0.0259 0.0267 0.0275 0.0283 0.0291 0.0297 0.0305 0.0313 0.0321 0.0334 0.0349 0.0364 0.0378 0.0393 0.0427 0.0461 0.0491 0.0521 0.0575 0.0622 0.0671 0.0718 0.0763 0.0817]; \]
\[ Pr = [0.707 0.705 0.703 0.701 0.699 0.698 0.696 0.694 0.692 0.690 0.688 0.686 0.684 0.682 0.681 0.680 0.677 0.674 0.676 0.678 0.687 0.699 0.706 0.713 0.717 0.719]; \]
\[ v = 1 \times 10^{-6} \times [13.28 14.16 15.06 16 16.96 17.95 18.97 20.02 21.09 22.10 23.13 25.45 27.8 30.09 32.49 34.85 40.61 48.33 55.46 63.09 79.38 96.89 115.4 134.8 155.1 177.1]; \]

% properties of water
\[ T_w = [0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200]; \]
\[ T_w = T_w + 273.2; \]
\[ \rho_w = [999.9 999.7 998.2 995.7 992.2 988.1 983.2 977.8 971.8 965.3 958.4 951 943.1 934.8 926.1 917 907.4 897.3 886.9 876 863]; \]
\[ v_w = 1 \times 10^{-6} \times [1.789 1.306 1.006 0.805 0.659 0.556 0.478 0.415 0.365 0.326 0.295 0.272 0.252 0.233 0.217 0.203 0.191 0.181 0.171 0.165 0.158]; \]
% kinematic viscosity of water in centistrokes ie mm²/s
Appendix A. Simulation of Linear Parabolic Collector

kw=[.551 .575 .599 .618 .634 .648 .659 .668 .675 .68 .683 .685 .686 .686 .685 .684 .683 .679 .675 .67 .663];

Tpm=Tfi+20; %absorber tube temperature, initial value

for i=1:50,
m_dot(i)=i*15/3600;
%First iteration values
Ql(1)=0;
Ql(2)=200;
Tfo=Tfi+5;
while abs(Ql(2)-Ql(1))>0.01
Ql(1)=Ql(2);
Ul=Ql(1)/(pi*Do*(Tpm-Ta));
Tfio=(Tfo+Tfi)/2;
%calculation of Hf
rhow1=interp1(Tw,rhow,Tfio,'linear');
Cpw1=interp1(Tw,Cpw,Tfio,'linear');
vw1=interp1(Tw,vw,Tfio,'linear');
kw1=interp1(Tw,kw,Tfio,'linear');
V=m_dot(i)/(pi*Diˆ2*rhow1/4);
Rey=V*Di/vw1;
Prd=Cpw1*vw1*rhow1/kw1;
Nul=5.172*sqrt((1+.005484*(Prd*((Rey/Trr)ˆ1.78))ˆ0.7));
hf=Nul*kw1/Di;

F_1=1/(Ul*(1/Ul+Do/(hf*Di))); %collector efficiency factor
FR=Cpw1*m_dot(i)*(1-exp(-F_1*Ul*Ap/(Cpw1*m_dot(i))))/(Ul*Ap); %heat removal factor
qu=FR*(W-Do)*L*(S-Ul*(Tfi-Ta))/CR; %useful energy
ql=(W-Do)*L*S-qu; %heat lost
Tpm=ql/(Ap*Ul)+Ta; %plate temperature

Tc=310; %1st iteration value

x1=[0;0];
x2=[220;Tc];

while max((x2-x1))>0.1
    x1=x2;
    Ul=x2(1)/(pi*Do*(Tpm-Ta));
    Tc=x1(2);

    %calculation of hp-c
    Tpc=0.5*(Tpm+Tc);
    Pr1=interp1(T,Pr,Tpc,'linear');
    k1=interp1(T,k,Tpc,'linear');
    v1=interp1(T,v,Tpc,'linear');
    Ral=g*(Tpm-Tc)*spc1^3*Pr1/(Tpc*v1^2);
    keff=k1*0.317*log((Dci/Do))*Ral^0.25/((spc1^0.75)*((1/Do^0.6+1/Dci^0.6)^1.25));

    hpc=2*keff/(Do*log(Dci/Do))

    %calculation of hw
    Tca=0.5*(Tc+Ta)
    v1=interp1(T,v,Tca,'linear')
    k1=interp1(T,k,Tca,'linear')
    L_char=Dco;
    ReL=Vw*L_char/v1

    if ReL<=4000
        Nul=0.615*ReL^0.466;
        elseif ReL<=40000
        Nul=0.174*ReL^0.618;
        elseif ReL<=400000
        Nul=0.0239*ReL^0.805;
    end

    hw=Nul*k1/0.15

    J=[-1 -hpc*pi*Do-4*k_1*x1(2)^3; -1 hw*pi*Dco+4*k_2*x1(2)^3]
    C=[hpc*(Tpm-x1(2))*pi*Do+k_1*(Tpm-4*x1(2)^4)-x1(1);]
Appendix A. Simulation of Linear Parabolic Collector

\[ hw*(x_1(2)-T_{a})*\pi*D_{co}+k_2*(x_1(2)^4-T_{sky}^4)-x_1(1) \]

\[
x_2=x_1-inv(J)*C
\]

end

\[ Q_l(2)=x_2(1) \]

\[ U_l=Q_l(2)/(\pi*D_0*(T_{pm}-T_{a})); \]

\[ qu=FR*(W-D_0)*L*(S-U_l*(T_{fi}-T_{a})/CR); \text{\%useful energy} \]

\[ T_{fo}=T_{fi}+qu/((m_{dot}(i))*C_{pw1}); \text{\%outlet water temperature} \]

end

UL_ans(i)=Ul; \text{\%heat loss coefficient}
FR_ans(i)=FR; \text{\%Heat removal factor}
F_1ans(i)=F_1; \text{\%collector efficiency factor}
q_u(i)=qu; \text{\%useful heat gain in W}
q_l(i)=ql; \text{\%heat lost (in W)}
q_pump(i)=k_pump*m_{dot}(i)^3; \text{\%power required to pump water}
n_i(i)=qu/(G*W*L); \text{\%instantaneous efficiency}
Tfo_ans(i)=T_{fi}+qu/(m_{dot}(i)*C_{pw1}); \text{\%outlet water temperature}
Tw1(i)=0.5*(T_{fo}+T_{fi}); \text{\%average water temperature}

end

plot(m_{dot}, n_i)
Appendix B

Design Toolbox

This is the main program called as system_design.m which calls several functions given as m-files. The functions are called in the sequence given in the following main program:

```matlab
%**************************************************************************
% This is the main program called as system_design.m which calls several functions given as m-files. The functions are called in the sequence given in the following main program:
%**************************************************************************
% This is the main program for the solar cooking system design.
% This will call all the other modules of the toolbox
% Copyright: CEDT, Indian Institute of Science
% Date : June 2010
%**************************************************************************
%
clear
clc

spec; %calls the spec.m module
[Htc_min,Htc,N] = insolation(N,aLat) %calls the insolation.m module
[Energy_LPG,Energy_solar,Energy_night]=loading(LPG,Per_saving,Per_night) %calls the loading.m module
[app_area]=collector_sizing(Energy_solar,Htc_min) %calls the collector_sizing.m module
[ST_vol]=storage_sizing(Energy_night,Htc_min,T_max,T_min) %calls the storage_sizing.m module
[P2,P3]=cp_rating(Q1,hd1,Q2,hd2) %calls the cp_rating.m module
```
Appendix B. Design Toolbox

[E_rpv, area_pv, W3] = pv_sizing(P1, h1, P2, h2, P3, h3, P4, h4, d_ns, Htc_min, Per_night) % calls the pv_sizing.m module

[battery_capacity] = battery_sizing(P3, h3, P4, h4, W3, Vnom, Per_night) % calls up battery_sizing.m module

spec_for_LCC; % Calls the spec_for_LCC.m module

[cap, LCC, ALCC, cost] = system_costing(LCC_spec) % calls system_costing.m module

spec.m

Module that gives specifications

% Specifications for the Solar cooking system

% Specifications for insolation.m module
N = [15 46 74 105 135 166 196 227 258 288 319 349];
aLat = 13;

% Specifications for loading.m module
LPG = 1;
Per_saving = 90;
Per_night = 50;

% Specifications for storage_sizing.m module
T_max = 200;
T_min = 80;

% Specifications for cp_rating.m module
Q1 = 2/60000; % 2 LPM
hd1 = 2;
Q2 = 2/60000; % 2 LPM
hd2 = 10;

% Specifications for pv_sizing.m module
P1 = 50;
h1 = 1/60;
%Specifications for battery_sizing.m module
h3=2;
P4=10;
h4=8;
Per_night=50;
Vnom=12;

%Specifications for spec_for_LCC.m module
RT2 = 500; %reflector material rate in Rs/m^2
RT3 = 120; %cost of PV per peak watt
ST = 15e3; %cost of the heat storage tank per m^2
ST_insu = 4e3; %cost of insulation for the heat storage tank per m^2
Oil = 80*1000; %cost of circulating fluid per m^3, 80Rs/litre
CP7 = 5000; %linear actuator
CP8 = 8000; %pump-1
CP9 = 8000; %pump-2
CP10 = 500*6; %cost of temperature sensors
CP11 = 5000; %cost of heat exchanger
C_bat = 50; %cost of battery per Ah
CP13 = 2000*2; %cost of flow meter
CP14 = 300*20; %piping cost for 20m
CP15 = 1000; %other electronics
AM = 2000; %annual maintenance cost
S1 = 25000; %salvage
N1 = 4; %replacement interval
N2 = 25; %life of the system
RT4 = 7; %Cost of conventional energy source in Rs/kWh
insolation.m

Module that returns annual insolation profile and minimum insolation in year

function [Htc_min,Htc,N]=insolation(N,aLat)
%Calculates Global Solar Radiation Htc at any location on any day of the YEAR
%
%SYNTAX: [Htc_min,Htc,N]=insolation(N,aLat)
%
% where
% Htc_min = KWh/m^2/day - minimum energy/unit area/day in the year
% Htc = array of montly mean energy/unit area/day
% N = array of day number
% aLat = Latitude of the place in degrees
%
%Copyright: CEDT, Indian Institute of Science
%Date: June 2010
%
error(nargchk(2,2,nargin));

Beta=(pi/180)*0;
Phi=(pi/180)*aLat;

B=[1.6204 -0.2291 -0.0068;
  1.6857 -0.0073 -0.0020;
  -1.2423 -0.0810 -0.0010;
  0.5626 0.0708 0.0019;
  -1.2140 0.0064 0.0014;
  -0.0990 -0.0133 -0.0004;
  0.4972 0.0605 0.0015];

x=(aLat-35);
X=[1;x;x.*x];
B1=B(1,1:3)*X;
B2=B(2,1:3)*X;
B3=B(3,1:3)*X;
\[ B4 = B(4,1:3) \times X; \]
\[ B5 = B(5,1:3) \times X; \]
\[ B6 = B(6,1:3) \times X; \]
\[ B7 = B(7,1:3) \times X; \]

\[
A = \begin{bmatrix}
0.5563 & 0.0089 & 0.0002 & 0.0743 & -0.0089 \\
-0.2350 & 0.0119 & 0.0004 & 0.1473 & -0.0237 \\
-0.1011 & -0.0091 & -0.0004 & 0.1029 & -0.0201 \\
0.0136 & 0.0041 & 0.0002 & -0.0071 & 0.0010 \\
0.1300 & -0.0133 & -0.0003 & -0.0848 & 0.0098 \\
-0.0600 & 0.0048 & 0.0002 & 0.0733 & -0.0132 \\
0.0970 & 0.0058 & 0.0002 & -0.0282 & 0.0010
\end{bmatrix}
\]

for \( i = 1:12 \)
\[
t = \frac{(2\pi)}{365} \times (N(i) - 80);\]
\[
\Delta = 23.45 \times \sin\left(\frac{(2\pi)}{365} \times (N(i) - 80)\right);\]
\[
\Omega_1 = \frac{-\tan(\Phi)}{\tan\left(\frac{\pi}{180} \times \Delta\right)};\]
\[
\Omega = \arccos(\Omega_1);\]
\[
I_0 = 1.367 \times (1 + 0.033 \times \cos((2\pi) \times N(i) / 365));\]
\[
H_0(i) = \left[ \frac{24 \times I_0}{\pi} \right] \times \left[ \cos(\Phi - \Beta) \times \cos\left(\frac{\pi}{180} \times \Delta \right) \sin(\Omega) \right] \\
+ \left[ \sin(\Phi - \Beta) \times \sin\left(\frac{\pi}{180} \times \Delta \right) \sin(\Omega) \right];\]
\[
W(i) = B1 + B2 \times \sin(t) + B3 \times \sin(2t) + B4 \times \sin(3t) + B5 \times \cos(t) + B6 \times \cos(2t) + B7 \times \cos(3t);\]
\[
Y = [1; x; x \times x; W(i); W(i) \times W(i)];\]
\[
A1 = A(1,1:5) \times Y; \]
\[
A2 = A(2,1:5) \times Y; \]
\[
A3 = A(3,1:5) \times Y; \]
\[
A4 = A(4,1:5) \times Y; \]
\[
A5 = A(5,1:5) \times Y; \]
\[
A6 = A(6,1:5) \times Y; \]
\[
A7 = A(7,1:5) \times Y; \]
\[
K_t(i) = A1 + A2 \times \sin(t) + A3 \times \sin(2t) + A4 \times \sin(3t) + A5 \times \cos(t) + A6 \times \cos(2t) + A7 \times \cos(3t);\]
\[
H_{tc}(i,;) = K_t(i) \times H_0(i);\]
\]
end
\[
H_{tc\_min} = \min(H_{tc});\]

return
loading.m

Module that calculates daily cooking load

function [Energy_LPG, Energy_solar, Energy_night]=loading(LPG,Per_saving,Per_night)
%%Calculates the daily average cooking load
%
%%SYNTAX:
%[Energy_LPG, Energy_solar, Energy_night]=loading(LPG,Per_saving,Per_night)
%
% where
% LPG = number of LPG (Liquified Petroleum Gas) cylinders used per month
% Per_saving = Percentage of energy saving from solar
% Per_night = Percentage of night cooking
% Energy_LPG = Average energy required per day from LPG in k Wh/day
% Energy_solar = Average energy required per day from solar in k Wh/day
% Energy_night = Average energy required for night cooking per day in k Wh/day
%
%%Copyright: CEDT, Indian Institute of Science
%%Date: June 2010
%
error(nargchk(3,3,nargin));

LPG_kg=14.3; %weight of LPG in a cylinder in kg
LPG_calorific=49.62e6; %Calorific value of LPG in J/kg
LPG_eff=0.6; %efficiency of the LPG stoves

LPG_day = LPG*LPG_kg/30; %LPG used per day in kg
Energy_LPG=LPG_day*LPG_calorific*LPG_eff*(100-Per_saving)/(1000*3600*100); % Energy required per day from LPG in kWh/day
Energy_solar=LPG_day*LPG_calorific*LPG_eff*Per_saving/(1000*3600*100); % Energy required per day from solar in kWh/day (considering the efficiency of the LPG stove)
Energy_night=Energy_solar*Per_night/100; %Energy required for night cooking in kWh/day
Appendix B. Design Toolbox

return

collector_sizing.m

Calculates Area of the parabolic solar thermal collector

function [area_collector]=collector_sizing(Energy_solar,Htc_min)
%Calculates Area of the parabolic solar thermal collector
%
%SYNTAX: [area_collector]=collector_sizing(Energy_solar,Htc_min)
%
% where
% Energy_solar = Average Energy required for cooking from solar in kWh/day
% Htc_min = kWh/m^2/day - minimum energy/unit area/day in the
% year for a given location
% area_collector = Area of the parabolic collector required in m^2
%
%Copyright: CEDT, Indian Institute of Science
%Date: June 2010
%
error(nargchk(2,2,nargin));

direct_radiation=0.7; %percentage of direct radiation assumed to be 70%
eff_collector=0.5; %parabolic collector efficiency assumed to be 50%

Energy_collector=Energy_solar/(eff_collector*direct_radiation); %Energy input
required for the collector in kWh/day
area_collector=Energy_collector/Htc_min; %area of the collector in m^2

return

storage_sizing.m

This module calculates volume of the storage tank
function [storage_size] = storage_sizing(Energy_night, Htc_min, T_max, T_min)
% Calculates volume of the storage tank
% SYNTAX: [storage_size] = storage_sizing(Energy_night, Htc_min, T_max, T_min)
% where
% Energy_night = Average Energy required for cooking in the night kWh/day
% Htc_min = kWh/m^2/day - minimum energy/unit area/day in the
% T_max = Maximum storage temperature in K or C
% T_min = Minimum storage temperature in K or C
% storage_size = Size of the storage tank in m^3
%
% Copyright: CEDT, Indian Institute of Science
% Date: June 2010
%
error(nargchk(4,4,nargin));

C_p=2100; % specific heat of the circulating fluid in J/kgK
eff_storage=0.7; % efficiency of storage tank (storing and taking out energy) assumed to be 70%
density=900; % density of the circulating fluid in kg/m^3

delta_T=T_max-T_min; % temperature difference between maximum and minimum storage temperature in K
storage_size=Energy_night*3600000/(C_p*delta_T*eff_storage*density); % volume in m^3 (3600000 conversion from kWh to MJ)

return

cp_rating.m

This module calculates power rating of centrifugal pump

function [cpower_rating1, cpower_rating2] = cp_rating(Q1, hd1, Q2, hd2)
% Calculates Power rating of centrifugal pump
Appendix B. Design Toolbox

% SYNTAX: [cpower_rating1,cpower_rating2]=cp_rating(Q1,hd1,Q2,hd2)
% where
% Q1 = Maximum flow rate of the collector side
% circulation in m^3/s
% hd1 = Delivery head of collector side in meters
% Q2 = Maximum flow rate of the load side
% circulation in m^3/s
% hd2 = Delivery head of load side in meters
% cpower_rating1 = Power rating of the collector side centrifugal pump
% cpower_rating2 = Power rating of the load side centrifugal pump
%
% Copyright: CEDT, Indian Institute of Science
% Date: June 2010
%
error(nargchk(4,4,nargin));

eff_cp=0.3; % efficiency of the centrifugal pump
rho=900; % density of the circulating fluid in kg/m^3

\[ g=9.81; \]

\[ \text{power}_\text{cp1}=\rho Q1^2 g^*hd1; \]
\[ \text{power}_\text{cp2}=\rho Q2^2 g^*hd2; \]
\[ \text{cpower}_\text{rating1}=\text{power}_\text{cp1}/\text{eff}_\text{cp}; \]
\[ \text{cpower}_\text{rating2}=\text{power}_\text{cp2}/\text{eff}_\text{cp}; \]

return

\textbf{pv\_sizing.m}

Module that calculates area of PV panel

function [E_rpv,area_pv,W3]=pv_sizing(P1,h1,P2,h2,P3,h3,P4,h4,d_ns,Htc_min,
Per\_night)
Appendix B. Design Toolbox

%Calculates Area of the PV panel
%
%SYNTAX:
%[E_rpv,area_pv,W3]=pv_sizing(P1,h1,P2,h2,P3,h3,P4,h4,d_ns,Htc_min,Per_night)
%
% where
% P1 = Power rating of linear actuator in W
% h1 = Number of hours linear actuator is powered per day in hours
% P2 = Power rating of centrifugal pump-1 in W
% h2 = Number of hours pump-1 is powered per day in hours
% P3 = Power rating of centrifugal pump-2 in W
% h3 = Number of hours pump-2 is powered per day in hours
% P4 = Power rating of electronic control circuits in W
% h4 = Number of hours the electronic circuits are ON in hours
% d_ns = Number of no Sun days
% E_rpv = Total energy required from PV in Wh
% area_pv = area of the PV panel in m^2
% W3 = Energy required during no-sun days in Wh
% Htc_min = array of monthly mean energy/unit area/day
% Per_night = Percentage of night cooking
%
%Copyright: CEDT, Indian Institute of Science
%Date: June 2010
%
error(nargchk(11,11,nargin));

eff_PV=0.12; %efficiency of PV panel
eff_battery=0.7; %efficiency of battery charging and discharge circuits

W1=P1*h1+P2*h2+P3*h3*(100-Per_night)/100+P4*h4; %day time energy requirement
W2=P3*h3*Per_night/(eff_battery*100); %night time energy requirement
W3=(W1*d_ns); %Energy requirement for no sun days
E_rpv=(W1+W2+W3/eff_battery); %Total energy required in Wh
E_apv=Htc_min*eff_PV; %Energy available from PV kWh/m^2/day
area_pv=E_rpv/1000*E_apv; %area of PV panel in

B-10
function [battery_capacity] = battery_sizing(P3,h3,P4,h4,W3,Vnom,Per_night)

% Calculates Capacity of the battery
% 
% SYNTAX: [battery_capacity] = battery_sizing(P3,h3,P4,h4,W3,Vnom,Per_night)
% 
% where
% P3 = Power rating of centrifugal pump-2 in W
% h3 = Number of hours pump-2 is powered per day in hours
% P4 = Power rating of electronic control circuits in W
% h4 = Number of hours the electronic circuits are ON in hours
% W3 = Energy required during no-sun days in Wh % Vnom = Nominal voltage of the battery in volts
% battery_capacity = Capacity of the battery in Amp-hours
% Per_night = Percentage of night cooking
% 
% Copyright: CEDT, Indian Institute of Science
% Date: June 2010
% 
error(nargchk(7,7,nargin));

eff_batt=0.7;
DOD=0.4;

Wh_bat=P3*h3*Per_night/100+P4*h4+W3; % energy need to be stored in battery in Wh
battery_capacity=(Wh_bat/eff_batt)/(DOD*Vnom);

return
spec_for_LCC.m

Module that returns specifications for LCC calculations

% Specifications for LCC calculation given as an array
%
% Copyright: CEDT, Indian Institute of Science
% Date: June 2010
%
LCC_spec(1)=app_area; % Apperture area of the parabolic collector
LCC_spec(2)=RT2; % Reflector material rate in Rs/m^2
LCC_spec(3)=E_rpv; % Energy required from PV panel in watts
LCC_spec(4)=RT3; % Cost of PV per peak watt
LCC_spec(5)=ST_vol; % Volume of the heat storage tank
LCC_spec(6)=ST; % Cost of the heat storage tank per m^2
LCC_spec(7)=ST_insul; % Cost of insulation for the heat storage tank per m^2
LCC_spec(8)=Oil; % Cost of circulating fluid per m^3
LCC_spec(9)=CP7; % Cost of linear actuator
LCC_spec(10)=CP8; % Cost of pump-1
LCC_spec(11)=CP9; % Cost of pump-2
LCC_spec(12)=CP10; % Cost of temperature sensors
LCC_spec(13)=CP11; % Cost of heat exchanger
LCC_spec(14)=C_bat; % Cost of battery per Ah
LCC_spec(15)=battery_capacity; % Capacity of battery in Ah
LCC_spec(16)=CP13; % Cost of flow meter
LCC_spec(17)=CP14; % Piping cost
LCC_spec(18)=CP15; % Cost of other electronics
LCC_spec(19)=AM; % Annual maintenance cost
LCC_spec(20)=S1; % Salvage
LCC_spec(21)=N1; % Replacement interval
LCC_spec(22)=N2; % Life of the system
LCC_spec(23)=Energy_LPG;
LCC_spec(24)=RT4; % Cost of conventional energy source in Rs/kWh
LCC_spec(25)=Energy_solar;


**system_costing.m**

Module to calculate life cycle costing and annual life cycle costing

```matlab
function [Cap,LCC,ALCC,cost]=system_costing(LCC_spec)
%Calculates Life cycle costing and annual life cycle costing
%
%SYNTAX: [Cap,LCC,ALCC,cost]=system_costing(LCC_spec)
%
% where
% LCC_spec = Array of variables
% Cap = Capital cost
% LCC =Life cycle cost
% ALCC =Annual life cycle cost
% cost = Cost of unit energy in Rs/kWh
%
%Copyright: CEDT, Indian Institute of Science
%Date: June 2010
%
error(nargchk(1,1,nargin));

app_area= LCC_spec(1) ; %apperture area of the parabolic collector
RT2 = LCC_spec(2) ; %reflector material rate in Rs/m^2
E_rpv = LCC_spec(3) ; %Energy required from PV panel
RT3 = LCC_spec(4) ; %cost of PV per peak watt
ST_vol = LCC_spec(5) ; %volume of the heat storage tank
ST = LCC_spec(6) ; %cost of the heat storage tank per m^2
ST_insu = LCC_spec(7) ; %cost of insulation for the heat storage tank per m^2
Oil = LCC_spec(8) ; %cost of circulating fluid per m^3
CP7 = LCC_spec(9) ; %linear actuator
CP8 = LCC_spec(10) ; %pump-1
CP9 = LCC_spec(11) ; %pump-2
CP10 = LCC_spec(12) ; %cost of temperature sensors
CP11 = LCC_spec(13) ; %cost of heat exchanger
CP12 = LCC_spec(14)*LCC_spec(15); %cost of battery
CP13 = LCC_spec(16) ; %cost of flow meter
```
Appendix B. Design Toolbox

CP14 = LCC_spec(17); %piping cost
CP15 = LCC_spec(18); %other electronics
AM = LCC_spec(19); %annual maintenance cost
S1 = LCC_spec(20); %salvage
N1 = LCC_spec(21); %replacement interval
N2 = LCC_spec(22); %life of the system
RT4 = LCC_spec(24); %cost of conventional energy source
Energy_LPG = LCC_spec(23); % Energy required from conventional source
Energy_solar = LCC_spec(25); %Energy required from solar in kWh/day

i=0.06;
f=0.02;
fd1=0.4;
ST_DL=0.5; %D/L ratio of the heat storage tank
carbon_emission=1.57; %CO2 emission in kg/kWh
carbon_credit=1650; %Carbon credit in Rs/ton
LPG_cost=RT4; %cost of LPG in Rs/kWh
PB=12; %payback time

D=sqrt((4*app_area)/pi);
fl=fd1*D;
SA=((8*pi*fl^2)/3)*(((D/(4*fl))/2+1)^1.5-1);
CP1=SA*RT2;
CP2=0.6*CP1 %60% of paraboloid material cost
P_pv=E_rpv/5;
CP3=P_pv*RT3;
CP4=CP3*0.3 %30% of PV cost is considered
D=power((4*ST_DL*ST_vol/pi),1/3);
L=4*ST_vol/(pi*D*D);
CP5=(ST+ST_insu)*(2/L+4/D)*ST_vol; %cost of heat storage tank and insulation
CP6=Oil*ST_vol; %cost of oil
EC=Energy_LPG*RT4*365; %Annual energy cost
ME=AM+EC; %Total annual maintenance and energy cost
CC=Energy_solar*carbon_emission*365*carbon_credit/1000; %Annual Carbon credit in Rs
R=CP12*0.93+CP6*.95; %replacement cost
Appendix B. Design Toolbox

\[ \text{cost}_\text{LPG} = (\text{Energy}_\text{LPG} + \text{Energy}_\text{solar}) \times \text{LPG}_\text{cost} \times 365; \% \text{Annual cost of LPG} \]

\[ K1 = (\text{CP1} + \text{CP2} + \text{CP3} + \text{CP4} + \text{CP5} + \text{CP6} + \text{CP7} + \text{CP8} + \text{CP9} + \text{CP10} + \text{CP11} + \text{CP12} + \text{CP13} + \text{CP14} + \text{CP15}); \]
\[ P1 = K1; \]
\[ P2 = R \times \left( \frac{(1+f)}{(1+i)} \right)^{N1}; \]
\[ P3 = R \times \left( \frac{(1+f)}{(1+i)} \right)^{(2 \times N1)}; \]
\[ P4 = R \times \left( \frac{(1+f)}{(1+i)} \right)^{(3 \times N1)}; \]
\[ P5 = R \times \left( \frac{(1+f)}{(1+i)} \right)^{(4 \times N1)}; \]
\[ P6 = R \times \left( \frac{(1+f)}{(1+i)} \right)^{(5 \times N1)}; \]
\[ P7 = \text{ME} \times \left( \frac{(1+f)}{(1-i)} \right) \times (1 - \left( \frac{(1+f)}{(1+i)} \right)^{N2}); \]
\[ P8 = S1 \times \left( \frac{(1+f)}{(1+i)} \right)^{N2}; \]
\[ P9 = \text{CC} \times \left( \frac{(1+f)}{(1+i)} \right)^{-\text{PB}} \times (1 - \left( \frac{(1+f)}{(1+i)} \right)^{(N2 - \text{PB} + 1)}/(1 - \left( \frac{(1+f)}{(1+i)} \right)); \]
\[ \text{LCC} = (P1 + P2 + P3 + P4 + P5 + P6 + P7 - P8 - P9); \]
\[ \text{DEN} = \left( \frac{(1+f)}{(1-i)} \right) \times (1 - \left( \frac{(1+f)}{(1+i)} \right)^{N2}); \]
\[ \text{ALCC} = \text{LCC} / \text{DEN}; \]
\[ \text{Cap} = P1; \]
\[ \text{cost} = \frac{\text{ALCC}}{(\text{Energy}_\text{solar} \times 365)}; \]

\[ \text{return} \]
Appendix C

Experimental Results

Characteristic of storage tank:

<table>
<thead>
<tr>
<th>Time in hours</th>
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Table C.1: Temperature of storage tank

Characteristic of pumps:
### Appendix C. Experimental Results

#### Table C.2: Characteristic of pump-1

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<th>$V_{p1}$</th>
<th>$I_{p1}$</th>
<th>Flow in $V$</th>
<th>Flow in $kg/s$</th>
<th>Power in $W$</th>
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<td>0.0093</td>
<td>8.274</td>
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#### Table C.3: Characteristic of pump-2

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<th>Power in $W$</th>
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**Performance with constant input power:**

Result for the experiment conducted on 05-03-2010 with constant input power of 1225W with only collector side circulation is given in table C.4.

Result for the experiment conducted on 06-03-2010 with constant input power of 1600W and circulation on both the collector side and the load side is given in table C.5.
### Appendix C. Experimental Results

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<th>$T_{\text{cout}}$ °C</th>
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<th>$T_{\text{s2}}$ °C</th>
<th>$T_{\text{a}}$ °C</th>
<th>Flow in V</th>
<th>Flow in kg/s</th>
<th>$V_{\text{in}}$ V</th>
<th>$P_{\text{in}}$ W</th>
<th>$V_{\text{p1}}$ V</th>
<th>$I_{\text{p1}}$ A</th>
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*Table C.4: Collector side circulation with constant input power*
Appendix C.
Experimental Results

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<th>Time</th>
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<th>$T_{c_o}$ °C</th>
<th>$T_{s_l}$ °C</th>
<th>$T_{s_b}$ °C</th>
<th>$T_a$ °C</th>
<th>Flow $\dot{q}_1$ kg/s</th>
<th>$V_{p_1}$ V</th>
<th>$I_{p_1}$ A</th>
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<th>$T_{x_o}$ °C</th>
<th>$T_{l_i}$ °C</th>
<th>$T_{l_o}$ °C</th>
<th>$V_{p_2}$ V</th>
<th>$I_{p_2}$ A</th>
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Table C.5: Both side circulation with constant input power, $P_{in}=1600$W
Appendix C. **Experimental Results**

Result for the experiment conducted on 07-03-2010 with circulation on only load side is given in table C.6.

<table>
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<th>$T_{s_b}$ $^\circ C$</th>
<th>$T_a$ $^\circ C$</th>
<th>$T_{x_l}$ $^\circ C$</th>
<th>$T_{x_o}$ $^\circ C$</th>
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Table C.6: With only load-side circulation

**Results for variable input power:**

Result for the experiment conducted on 18-03-2010 with only collector side circulation is given in table C.7.

Result for the experiment conducted on 23-03-2010 with both the collector side and the load side circulation is given in table C.8.
## Appendix C. Experimental Results

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<tr>
<td>17:00</td>
<td>950</td>
<td>90.7</td>
<td>121.4</td>
<td>113.4</td>
<td>99.7</td>
<td>30.7</td>
<td>0.0114</td>
<td>23.6</td>
<td>0.56</td>
<td>85.9</td>
<td>61.2</td>
<td>74.2</td>
<td>69.4</td>
<td>46.6</td>
<td>0.35</td>
<td>0.658</td>
<td>0.0183</td>
</tr>
</tbody>
</table>

Table C.8: Both side circulation with variable input power
Appendix C. Experimental Results

Table C.9 gives the result for the experiment conducted on 22-03-2010 with only collector side circulation keeping flow rate constant.

<table>
<thead>
<tr>
<th>Time</th>
<th>$T_{c_i} \degree C$</th>
<th>$T_{c_o} \degree C$</th>
<th>$T_{s_l} \degree C$</th>
<th>$T_{s_b} \degree C$</th>
<th>$T_{a} \degree C$</th>
<th>$q_1$ kg/s</th>
<th>$V_p$ V</th>
<th>$I_p$ A</th>
</tr>
</thead>
<tbody>
<tr>
<td>8:00</td>
<td>29.2</td>
<td>26</td>
<td>38.9</td>
<td>37.2</td>
<td>27.8</td>
<td>0.747</td>
<td>0.0206</td>
<td>32.7</td>
</tr>
<tr>
<td>8:30</td>
<td>38.3</td>
<td>72.6</td>
<td>61.8</td>
<td>38.9</td>
<td>30</td>
<td>0.48</td>
<td>0.0128</td>
<td>41.8</td>
</tr>
<tr>
<td>9:00</td>
<td>40.4</td>
<td>84.7</td>
<td>85.7</td>
<td>41.1</td>
<td>31.3</td>
<td>0.421</td>
<td>0.0105</td>
<td>42</td>
</tr>
<tr>
<td>9:30</td>
<td>41.9</td>
<td>88.2</td>
<td>89.1</td>
<td>42.5</td>
<td>31.7</td>
<td>0.432</td>
<td>0.0109</td>
<td>42.9</td>
</tr>
<tr>
<td>10:00</td>
<td>57.2</td>
<td>104.3</td>
<td>103.9</td>
<td>57.5</td>
<td>32.9</td>
<td>0.434</td>
<td>0.011</td>
<td>35.9</td>
</tr>
<tr>
<td>10:30</td>
<td>85.7</td>
<td>112.3</td>
<td>110.2</td>
<td>88.8</td>
<td>34.9</td>
<td>0.48</td>
<td>0.0128</td>
<td>27.7</td>
</tr>
<tr>
<td>11:00</td>
<td>95.1</td>
<td>124.9</td>
<td>121.5</td>
<td>102.3</td>
<td>37.9</td>
<td>0.429</td>
<td>0.0108</td>
<td>24.8</td>
</tr>
<tr>
<td>11:30</td>
<td>104</td>
<td>127.8</td>
<td>141.2</td>
<td>111.4</td>
<td>34.8</td>
<td>0.442</td>
<td>0.0113</td>
<td>25.3</td>
</tr>
<tr>
<td>12:00</td>
<td>112.6</td>
<td>141.5</td>
<td>153.8</td>
<td>122.4</td>
<td>35.4</td>
<td>0.43</td>
<td>0.0109</td>
<td>24</td>
</tr>
<tr>
<td>12:30</td>
<td>118.1</td>
<td>142.2</td>
<td>163.2</td>
<td>129.2</td>
<td>34.8</td>
<td>0.432</td>
<td>0.0109</td>
<td>23.8</td>
</tr>
<tr>
<td>13:00</td>
<td>130</td>
<td>150.9</td>
<td>168.4</td>
<td>145.2</td>
<td>34.1</td>
<td>0.453</td>
<td>0.0118</td>
<td>24.6</td>
</tr>
<tr>
<td>13:30</td>
<td>131.5</td>
<td>158.1</td>
<td>175.2</td>
<td>148.2</td>
<td>33.9</td>
<td>0.422</td>
<td>0.0105</td>
<td>22.1</td>
</tr>
</tbody>
</table>

Table C.9: With constant flow rate

Calibration of flow meters

<table>
<thead>
<tr>
<th>Flow rate 0.01 kg/s</th>
<th>Output voltage in V</th>
<th>Flow rate 0.01 kg/s</th>
<th>Output voltage in V</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.255</td>
<td>0</td>
<td>0.578</td>
<td>1.675</td>
</tr>
<tr>
<td>0.324</td>
<td>0.62</td>
<td>0.598</td>
<td>1.681</td>
</tr>
<tr>
<td>0.325</td>
<td>0.541</td>
<td>0.611</td>
<td>1.712</td>
</tr>
<tr>
<td>0.409</td>
<td>0.841</td>
<td>0.62</td>
<td>1.861</td>
</tr>
<tr>
<td>0.425</td>
<td>1.013</td>
<td>0.627</td>
<td>1.727</td>
</tr>
<tr>
<td>0.436</td>
<td>1.174</td>
<td>0.644</td>
<td>1.806</td>
</tr>
<tr>
<td>0.446</td>
<td>1.005</td>
<td>0.645</td>
<td>1.782</td>
</tr>
<tr>
<td>0.463</td>
<td>1.213</td>
<td>0.674</td>
<td>1.8524</td>
</tr>
<tr>
<td>0.476</td>
<td>1.253</td>
<td>0.691</td>
<td>1.896</td>
</tr>
<tr>
<td>0.498</td>
<td>1.382</td>
<td>0.704</td>
<td>1.902</td>
</tr>
<tr>
<td>0.502</td>
<td>1.360</td>
<td>0.704</td>
<td>1.933</td>
</tr>
<tr>
<td>0.503</td>
<td>1.375</td>
<td>0.739</td>
<td>1.959</td>
</tr>
<tr>
<td>0.504</td>
<td>1.369</td>
<td>0.755</td>
<td>2.009</td>
</tr>
<tr>
<td>0.556</td>
<td>1.6</td>
<td>0.811</td>
<td>2.174</td>
</tr>
<tr>
<td>0.571</td>
<td>1.652</td>
<td>0.828</td>
<td>2.279</td>
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</tbody>
</table>

Table C.10: Calibration of flow meter-1
### Appendix C. Experimental Results

<table>
<thead>
<tr>
<th>Flow rate 0.01 kg/s</th>
<th>Output voltage in V</th>
<th>Flow rate 0.01 kg/s</th>
<th>Output voltage in V</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.255</td>
<td>0</td>
<td>0.402</td>
<td>1.047</td>
</tr>
<tr>
<td>0.302</td>
<td>0.655</td>
<td>0.423</td>
<td>1.136</td>
</tr>
<tr>
<td>0.305</td>
<td>0.705</td>
<td>0.424</td>
<td>1.107</td>
</tr>
<tr>
<td>0.313</td>
<td>0.698</td>
<td>0.455</td>
<td>1.262</td>
</tr>
<tr>
<td>0.318</td>
<td>0.741</td>
<td>0.481</td>
<td>1.299</td>
</tr>
<tr>
<td>0.321</td>
<td>0.762</td>
<td>0.486</td>
<td>1.352</td>
</tr>
<tr>
<td>0.338</td>
<td>0.787</td>
<td>0.517</td>
<td>1.47</td>
</tr>
<tr>
<td>0.362</td>
<td>0.879</td>
<td>0.547</td>
<td>1.54</td>
</tr>
<tr>
<td>0.364</td>
<td>0.886</td>
<td>0.58</td>
<td>1.641</td>
</tr>
<tr>
<td>0.366</td>
<td>0.897</td>
<td>0.605</td>
<td>1.722</td>
</tr>
<tr>
<td>0.367</td>
<td>0.946</td>
<td>0.639</td>
<td>1.813</td>
</tr>
<tr>
<td>0.389</td>
<td>1.057</td>
<td>0.644</td>
<td>1.85</td>
</tr>
<tr>
<td>0.393</td>
<td>1.013</td>
<td>0.692</td>
<td>1.995</td>
</tr>
<tr>
<td>0.395</td>
<td>1.015</td>
<td>0.771</td>
<td>2.175</td>
</tr>
<tr>
<td>0.398</td>
<td>1.004</td>
<td>0.779</td>
<td>2.164</td>
</tr>
</tbody>
</table>

**Table C.11: Calibration of flow meter-2**
Appendix D

List of Publications

The following is the list of publications out of this thesis work.


Patent:

Appendix E

Photograph of the experimental setup

Figure E.1: Photograph of the lab prototype
Figure E.2: Photograph of the low flow measuring system
Figure E.3: Photograph of the setup for calibration of the flow meter
Appendix E. *Photograph of the experimental setup*

**Figure E.4:** Photograph of the paraboloid collector

**Figure E.5:** Photograph of the experimental setup
Figure E.6: Photograph of the heat exchanger in kitchen

Figure E.7: Thermocouple, used for temperature measurement

Figure E.8: Photograph of the circulating pump
Bibliography


