Performance of Solar Organic Rankine Cycle

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***Abstract*— In the present scenario of huge demand for energy and economy necessitates development of various energy resources either, conventional or nonconventional. Despite the rapid depletion of fossil fuel across the world, billions of people are yet devoid of the comfort offered by electricity. If the consumption of fossil fuel continues at the current rate, the future generation is bound to suffer from the acute shortage. The associated global warming and ozone layer depletion caused due to intensive application of fossil fuels forces us to look for solar based systems. One such system is an Organic Rankine Cycle (ORC) plant which is modular and scalable. It can be easily transported, assembled and commissioned rapidly at site, may it be in small industrial units or "micro-grids" for remote and isolated areas. The heat-energy converter of the ORC plant is a hermetically sealed unit with a few moving parts. This technology now turns out to be proven and available to all. The plant requires no operator, the maintenance cost is negligible over long periods, and the unit can be operated and monitored remotely. The design, technologies and materials proposed to be used are largely indigenous and it acquires a significant improvement over the traditional units used earlier for large plants, thus it provides acceptable performance at low capital cost. The solar energy available for almost 295 days a year in India is utilized by an array of sun-tracking parabolic-trough collectors. The functionality and performance of such newly developed low-temperature ORC unit comprising of helical coil solar cavity receiver based parabolic trough concentrator (PTC) was investigated at CERD, Mechanical Engineering, IIT(BHU), Varanasi. The PTC comprised of blackened helical coil made up of two concentric borosilicate glass cylinder with vacuum in annulus was kept at focal line that maximized the conversion of energy received from sun into useful heat and eventually electricity.**

Keywords-ORC (Organic Rankine Cycle), Scroll, Shell and Tube heat exchanger, Gear Pump

I. INTRODUCTION

ORCs are very important in case of both power generation and cooling when the exhaust-stream temperature drops below 370°C. However, recovering low-grade waste heat in power generation becomes economically feasible when using ORCs [1]. At low temperatures, organic fluids lead to higher cycle efficiency than water. In small plants, organic fluids are preferred, as fluid mechanics leads to high turbine efficiency also in partial load [1]. Organic Rankine cycles for low grade waste heat recovery are described with different working fluids. The effects of the thermodynamic parameters on the ORC performance are drastic, and the thermodynamic parameters of the ORC for each working fluid are needed to be optimized with exergy efficiency as an objective function by means of some method like hit and trial or by minimization of some function by algorithm [3].

The heat exchanger component is one of the important component in deciding efficiency there was some innovative research done by using plastic heat exchangers to reduce overall cost of plant [4]. There are again various methods of optimization like using MATLAB optimization environment [5].

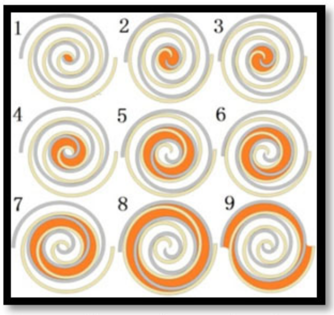

Figure 1. The various positions and expansions in a scroll expander [18]

The study of the main component the expander is the most difficult part especially when you are using scroll expander because there are no such correlations available as we have in normal turbine. Most of the studies in this subject were either done by simulations or by practical experiments and then optimization. Successful experimental studies on the scrolls that are reverse engineered to operate in the expansion mode recommend their use in the ORCs over conventional turbines when the power capacities lie in the range of 1–50 kW have been done. However, when it comes to the design of a new scroll expander for a particular ORC requirement, an ORC engineer hardly finds any guidelines in terms of selecting the geometric features of scroll such as height, involutes base circle radius, and number of expansion chambers[8].

II. Setup Description

The Solar ORC setup was built by an associate firm based on the details provided by us. The setup is designed to use R-123 as working fluid. In the setup we have used a Copeland Emerson Scroll (ZR36K3-TFD). It was basically a scroll compressor but we modified it to work as an expander. Both of the heat exchangers are based on relatively new design of shell and tube heat exchanger in which there are three parallel helical tubes running along the length. And the pump used was a gear pump.

The heating loop for the time being is using three electric heaters with water as working fluid. Sometime later water will be replaced by thermic oil and heating will be done by solar array present in the CERD lab.

The scroll machine was patented in 1905 by Creux [17].Scroll expanders are positive displacement machines have some advantages over turbines; first they are available at cheaper price (in the eventuality of mass-production such as in HVAC), their ability to handle a liquid phase (particularly interesting for Organic Rankine cycle applications) and their lower rotational speeds [9].

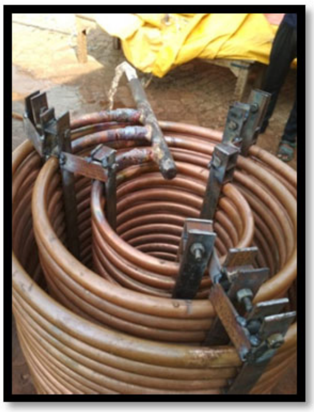

Figure 2. The inside view of triple coil helical shell and tube type heat exchanger [20]

III. Costs Related to Setup

Things bought from the market included Copeland scroll compressor, Gear pump, Piping, Sheet metal, Various Gauges. They were assembled by company named Swaraj Herbals situated in Barabanki, UP, India. The Lump-sum cost came near-about 3.5 lakh Indian rupees. For the solar collectors the cost was around 20,000 INR.

Later on R-123 was purchased from local market. The fabricating cost was kept low by self fabrication of heat exchangers. Now generally once commissioned there will be absolutely nil working costs. The system is stand-alone one. The only possible maintainace costs will arrive due to fouling of heat transfer surfaces in heat exchangers and also due to partial pressure loss due to gas leakage.

The solar array was bought from the local market from the lowest bidder. And these arrays well function for years if no damage is caused due to impacts. The whole of the system is quiet portable and its assembly is easy using the blueprints. Hence they are completely adequate to get seted up in rural areas. Once commissioned, they can perform well for years with minimum maintainace.

IV. Mathematical Model

The model of the scroll expander worked upon was very similar to the one developed by lamort. In which he divided the ongoing complex process into a series of simple ones. The semi-empirical model of a scroll expander described here under was deduced from the one made by Winandy et al.[19] for scroll compressors. The model was already been partially validated by tests with steam as working fluid. The model parameters are identified for the expander under investigation, integrated into a Rankine cycle and fed with R-256fa. The main method of the expander modeling is shown below. In this model, the evolution of the fluid through the expander is decomposed into the following consecutive steps:

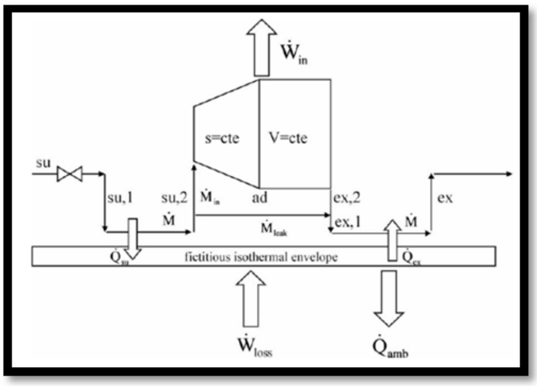

Figure 3. The actual flow process assumed in scroll expander [21]

(a) Adiabatic supply pressure drop (su - su_1)
(b) Isobaric supply cooling-down (su_1 - su_2)
(c) Adiabatic and reversible expansion to the ‘‘adapted” pressure imposed by the built-in volume ratio of the machine (su_2 – ad)
(d) Adiabatic expansion at a constant machine volume (ad – ex_2)
(e) Adiabatic mixing between supply and leakage flows (ex_2 – ex_1)
(f) Isobaric exhaust cooling-down or heating-up (ex_1 - ex)

From the description given , it can be observed that the heat transfers, the supply pressure drop and the internal leakage are fictitiously dissociated from the actual expansion process (su_2 – ex_2). All the processes mentioned here under are described in the following paragraphs.

1)The supply pressure drop (Psu - Psu_1)

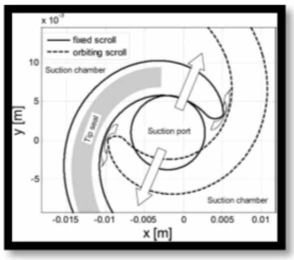

Figure 4. Inlet valve view of scroll expander showing the variation of area at inlet port

It accounts for all pressure losses encountered by the fluid from the suction line to the suction chamber. The lumped supply pressure drop is computed by comparison to the isentropic flow through a converging nozzle, whose cross-sectional area A_su is a parameter to identify. Because of the steady-state nature of the model, this cross-sectional area represents an average value of the suction port effective area over the entire suction process (that extends over one shaft revolution).

$$m = \frac{A_{su}}{v_{thr}} \times \sqrt{2}(h_{su} - h_{thr})$$

2) Supply and exhaust heat transfers

The main heat transfer mechanisms inside the scroll expander occur between: (1) the expander shell and the fluid in the supply and exhaust pipes; (2) the scrolls (fixed and orbiting) and the fluid in the suction, expansion and discharge chambers; (3) between the

Shell and the ambient. Both supply and exhaust heat transfers are computed by introducing

a fictitious metal envelope of uniform temperature T_w. This fictitious envelope represents the metal mass associated to the expander shell, the fixed and the orbiting scrolls. The supply heat transfer is given by:

$$Q_{su} = (1 - e^{\left(\frac{-AU_{su}}{m*C_p}\right)}) \times m \times C_p \times (T_{su1} - T_w)$$

3) Internal leakage

There are two different leakage paths in a scroll compressor/expander: the radial leakage is due to a gap between the bottom or the top plate and the scrolls and the flank leakage results from a gap between the flanks of the scrolls. In this modeling, all the leakage paths are lumped into one unique fictitious leakage clearance, whose cross-sectional area A_leak is a parameter to identify.

The leakage flow rate can be computed by reference to the isentropic flow through a simply convergent nozzle, whose throat area is A_leak. The pressure at the inlet of the nozzle is P_su2 The throat pressure corresponds to the maximum between exhaust and critical

Pressures:

$$m_{leak} \times v_{ex2} = A_{leak} \times ((2 \times (h_{su2} - h_{ex2})^{0.5}))$$

4) Displaced mass flow rate

As shown, the internal mass flow rate M_ in is the difference between the mass flow rate M_ entering the expander and the leakage mass flow rate M_ leak. The internal mass flow rate is the volume flow rate V_ exp divided by the specific volume of the fluid v_su2 after pressure drop and cooling down. The volume flow rate is the swept volume V_exp multiplied by the expander rotational speed N. The swept volume in expander mode is equal to the one in compressor mode V_cp divided by the built-in volume ratio of the machine r_v.

$$m = m_{in} + m_{leak}$$

$$m_{in} = ((N * V_{s.CP})/(v_{su2} * r_v))$$

5) Internal expansion

One working cycle of the scroll expander includes three processes: suction, expansion and discharge. During the suction process, the suction chamber is in communication with the suction line and the fluid flows into the chamber. The expansion process is initiated when the suction chamber ceases to be in communication with the suction line. The discharge process begins when the discharge chambers enter in communication with the discharge line.

$$W_{in} = m_{in} \times (h_{su2} - h_{ex2})$$

6) Shaft power

Mechanical losses W_ loss is due to friction between the scrolls and losses in the bearings. In the present modeling, all these losses are lumped into one unique mechanical loss torque T_loss that is a parameter to identify. The modeling assumes that this torque is independent of the rotational speed. Accordingly, the expander shaft power can be defined by:

$$W_{sh} = W_{in} - 2 \times 3.14 \times N \times T_{loss}$$

7) Heat balance over the expander

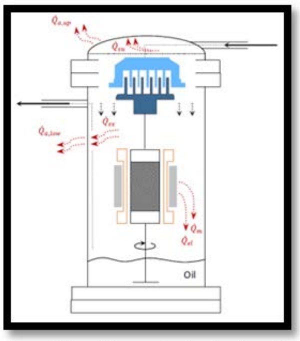

Figure 5. The ambient losses in the body of expander [21]

The ambient losses are computed by introducing a global heat transfer coefficient AU_amb between the envelope and the ambient:

$$W_{loss} - Q_{ex} + Q_{su} - Q_{amb} = 0$$

8) Identification of the parameters of the model

First method deals with a trial and error sort of methodology. In which the help of P-h chart is taken. Points are marked based on actual expander's performance and the idealized model. Various processes are, marked on the chart and the parameters are identified based on the best possible curve fitting. This is a very tedious task and a number of experiments are need to be done. Moreover the results are valid in only nearby regions of p-h chart and the performance can't be predicted far off.

The second method is a way broader method in comparison to the first one in which the parameter identification process needs input parameter values like mass flow rate, inlet and outlet pressures with the detailed information regarding the physical setup like areas at various crossection, the volume ratio etc. Then various experiments are on a broader scale and a minimizing algorithm of EES is incorporated which decides which parameters are most affecting the expanders performance and which are to be minimized. A minimization function is designed whose value needs to minimize by varying the parameters and making some parameters fixed. The model calculates the work output by the expander, the delivered mechanical power with the actual performance parameters. The model only necessitates eight parameters, which are given in table. The nominal mass flow rate M_ n is only introduced as a reference to define the nominal heat transfer coefficients AU_sun and AU_exn. Imposing the supply pressure as an input variable and the mass flow rate as an output variable is purely a convention. In fact, the mass flow rate could be imposed as an input and the supply pressure would be predicted by the model.

Figure 6. Side view of HE showing the length, spacing of tubes

It is basically a modified shell and tube type heat exchanger. By coiling a tube, heat transfer may be enhanced without turbulence or additional heat transfer surface area [12][13]. In this case, centrifugal forces within the fluid induce a secondary flow consisting of a pair of longitudinal vortices that, in contrast to conditions in a straight tube, can result in highly non-uniform local heat transfer coefficients around the periphery of the tube. Hence, local heat transfer coefficients vary with angle as well as length. If constant heat flux conditions are applied, the mean fluid temperature can be calculated using the conservation of energy principle [11].

For situations where the fluid is heated, maximum fluid temperatures occur at the tube wall, but calculation of the maximum local temperature is not straightforward because of the angular-dependence of the heat transfer coefficient. Therefore, correlations for the peripherally averaged Nusselt number are of little use if constant heat flux conditions are applied. In contrast, correlations for the peripherally averaged Nusselt number for constant wall temperature boundary conditions are useful. The secondary flow increases friction losses and heat transfer rates. In addition, the secondary flow decreases entrance lengths and reduces the difference between laminar and turbulent heat transfer rates, relative to the straight tube cases considered previously in this chapter. Pressure drops and heat transfer rates exhibit little dependence on the coil pitch [12].

$$Nu = \left[\left(3.66 + \frac{4.33}{a}\right)^3 + 1.158 * \left(Re * \frac{\left(\frac{D}{C}\right)^{0.5}}{b}\right)^{1.5}\right]^{0.333} \times \left(\frac{\mu}{\mu_s}\right)^{0.14}$$

$$a = \left[1 + \frac{927\left(\frac{C}{D}\right)}{\mathrm{Pr} \times Re^2}\right]$$

$$b = 1 + 0.477/Pr$$

V. Results

Various graphs were plotted for the scroll expander using idealised situations.

The first graph shows the variation of efficiency vs. P_ex curve and Work output vs. P_ex curve for our scroll expander when it is running at constant RPM and P_in. This curve is very similar to the idealised characteristics of an expander. The performance decreases as the exit pressure increases. The best performance is obtained at an idealised pressure of P_crit when the under-expansion and over expansion losses are both negligible. The P_crit pressure is obtained by the isentropic expansion of the working fluid from P_in to P_out. The efficiency of this graph was plotted by dividing the work obtained with the maximum work that can be obtained in idealised conditions.

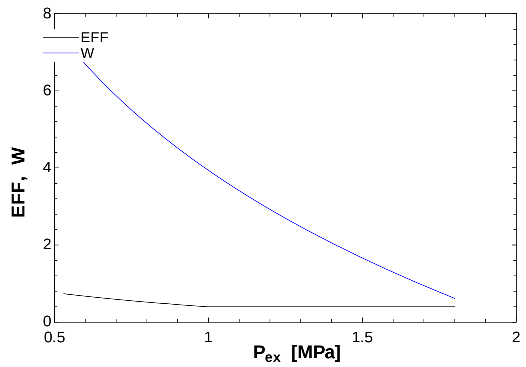

Figure 7. Variation of effectiveness (EFF) and work output vs. P_ex

This graph is also similar to the first one but the input variables were changed and the P_ex was varied in more drastic manner.

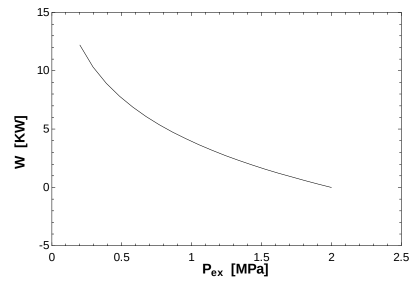

Figure 8. Variation of work output for more severe variation

This graph is very important graph it shows the variations of efficiency for our expander working at constant parameters except the P_in. The first segment shows the variation when they are under expansion losses till when a peak reaches where the fluid gets expanded from P_in to P_cit and the best possible efficiency is obtained under specific working conditions. After the peak there are over expansion losses and the efficiency decreases drastically. These are the graphs for idealised situations where the input variables were assumed and the effects of variations were of outmost important then the magnitude of those variations. These graphs are very similar to the ones produced by many authors using testing on actual expander. Yiji Lu [16] was having similar results for increase in P_in. His efficiency was also increasing but he might have not considered the optimum operating position. There was a gradual change observed when experiments were done by using Carbon dioxide as working fluid [10].

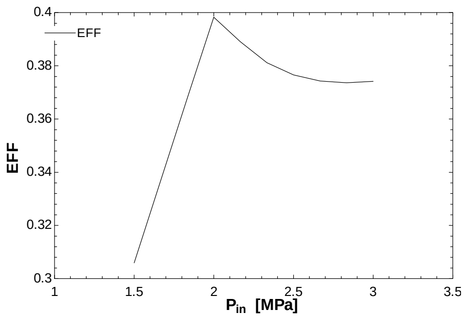

Figure 9. Variation of effectiveness vs inlet pressure showing optimum operating condition

This Graph shows the variations in work obtained vs. the pressure ratio S. More is the pressure ratio more is the out obtained. But it is not the case for the efficiency which decreases due to many exergy losses. This curve starts from the zero when the P_in is equal to P_out and there is zero work developed. And it gradually increases till the P_in increase to twice the value of P_ex. Xinjing Zhang was having similar results for variation in work output vs the pressure ratio [14].

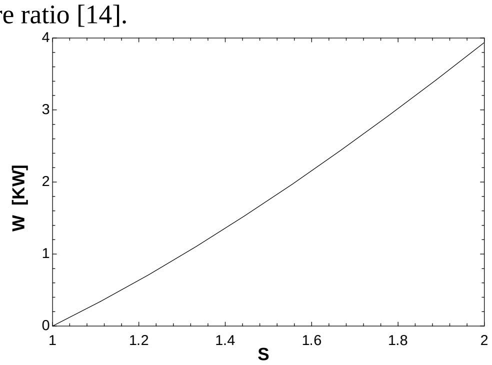

Figure 10. The variation of work output vs. the pressure ratio (S).

This graph was plotted in excel for the three coils of our heat exchanger. Coil 1 is of the smallest diameter and coil 3 is the largest. The smallest diameter coil is having the maximum Nusselt numbers because there are severe secondary flows which are generated inside the tube due to its high low radius curvature. But the area of that tube is also minimum so that the effect of rise in Nusselt number diminishes out. This curve was plotted for Nusselt vs the Reynolds number for constant Prandtl number conditions inside the tube. We can maintain very low Reynolds number that will eventually mimic free convection conditions but when Reynolds number is increased it changes to forced convection which leads to an increase in Nusselt number[11][13].

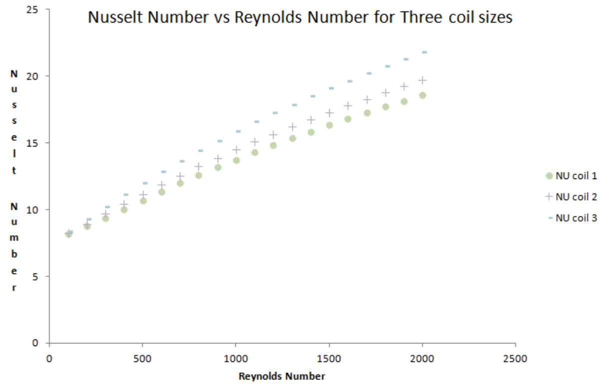

Figure 11. Variation of Nu vs Re for constant Pr for three coils

This graph is a very peculiar characteristic for the heat exchanger. In this graph Nusselt number was varied with Prandtl number for fluid flow and rise in Nusselt number was observed. Coming to some observations when this graph was plotted for various Reynolds numbers a sudden rise in Nusselt number was observed at the very beginning at an Reynolds number of around 100. The inference that can be derived is there is a threshold value of Reynolds number and hence the mass flow rate above only which the effects of secondary flows are important. Hence we should always keep the flow characteristics such that mass flow rate is above the threshold value. Below those values our heat exchanger behaves like a normal shell and tube heat exchanger with only primary flows [12].

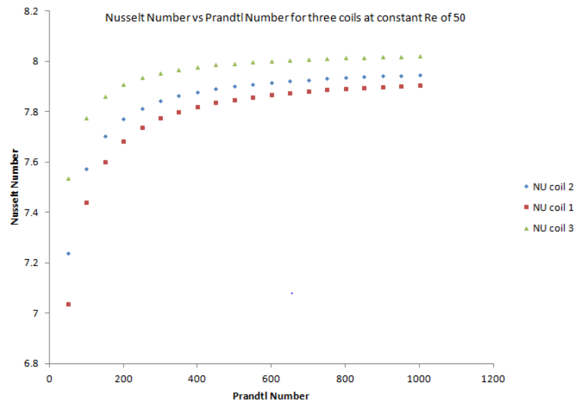

Figure 12. Variation of Nu vs Pr for Re number 50

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