Studies on a Solar Assisted, CO$_2$ Based Trigeneration System for Milk Processing: Performance Comparison between Throttle Valve and Ejector Expansion Valve

Vutukuru Ravindra and M Ramgopal

Abstract—**In this study, a transcritical CO$_2$ based solar assisted trigeneration system for a dairy farm is analyzed. Performance comparison is made between an ejector based system (C1) and a conventional throttle valve based system (C2). A mathematical model of the system operating under steady state conditions is developed in Engineering Equation Solver (EES). The results are presented based on the consideration that the evaporator load for chilling of milk exactly matches with heating load in the process heat exchanger for pasteurization of milk. A parabolic trough collector is assumed to be used to heat the working fluid and an auxiliary heater is used to supplement the solar heat. The operating parameters are adjusted in such a manner that net power produced is always positive. The effects of turbine inlet temperature, ambient temperature, turbine inlet pressure and process heat exchanger pressure on overall COP, cooling COP and power cycle efficiency are analyzed. It is observed that compressor power input required in C1 configuration is about 45% lower than that required in C2 configuration. Increase in turbine inlet temperature marginally affects the power cycle efficiency for both the configurations. Cooling COP reduces significantly with increase in ambient temperature for both the configurations. This study provides a basis for the feasibility of trigeneration systems in dairy application where simultaneous heating and cooling are required in addition to small amount of electricity for parasitic loads such as lights, fans etc.**

Index Terms—**CO$_2$, ejector, trigeneration, dairy application, solar energy.**

I. INTRODUCTION

Combined cooling, heating and power (CCHP) systems or trigeneration systems that run on renewable energy sources such as solar, biomass etc. find many commercial and industrial applications. In the past, several systems based on a variety of system configurations have been suggested to provide all these three outputs simultaneously. Almost all the systems suggested can be viewed as integration of power and refrigeration cycles using different working fluids for power cycle and refrigeration cycle. Though these cycles offer many advantages, they are complex due to the number of sub-systems and components involved. The system complexity, and possibly the initial cost of the system itself can be reduced considerably if a single cycle based on a single working fluid can be devised that can provide cooling,

heating and power simultaneously using solar or other renewable heat sources.

Wu & Wang [1] have provided a detailed review about CCHP technology by covering various aspects of the CCHP systems. Basic definitions related to CCHP along with present status and developments have been summarized. In addition, diverse CCHP configurations have been discussed along with a status report in US, Europe, Asia & Pacific and the Rest of the world. Li *et al.* [2] have carried out a sensitivity analysis of energy demands and their influence on performance of CCHP system. Numerical studies were carried out based on energy demands of a typical hotel and a hospital. Wang *et al.* [3] have proposed a new CCHP system driven by solar energy. The effects of hour angle and slope angle of the aperture plane for the solar collectors on the system performance are examined. Water is used as storage liquid and R123 is used as a working fluid. A detailed explanation of developing the solar subsystem has been provided along with necessary equations for both the solar collector and the thermal storage tank.

Ravindra and Ramgopal [4] have performed an energy analysis on two novel solar CCHP configurations. One is with an expansion turbine in the refrigeration loop and other with a conventional throttle valve. It is found that use of an expansion turbine in place of a throttle valve improves the performance under all conditions Fumo and Chamra [5] have studied the potential of solar thermal energy for CCHP systems. They have focused on the use of solar collectors to reduce Primary Energy Consumption (PEC) and emission of CO$_2$ in office buildings. Five different locations are considered to prepare a comparative study of the effect of solar collectors on system performance.

Use of ejector as an expansion device in place of the throttling valve in the refrigeration loop is a promising alternative to reduce the throttling losses in the refrigeration loop. Because of its simple structure, ease of manufacturing, no moving parts, low cost and low maintenance requirements, the use of two-phase ejectors has become an important cycle modification recently. Ejector reduces the compressor work by raising the suction pressure [6].

In this paper, a transcritical CO$_2$ solar assisted trigeneration system for a dairy farm is analyzed. Performance comparison is made between an ejector based system (C1) and a conventional throttle valve based system (C2). A steady state, mathematical model is developed in Engineering Equation Solver (EES) and results are obtained by varying important operating parameters.

Fig. 1. (a) Schematic diagram of C1 configuration (b) T-s diagram of C1 configuration.

Fig 2. (a) Schematic diagram of C2 configuration (b) T-s diagram for C2 configuration.

II. SYSTEM DESCRIPTION

The trigeneration system (Fig. 1 and Fig. 2) consists of a compressor, recovery heat exchanger, parabolic trough collector (PTC), turbine-generator, process gas cooler for milk pasteurization, ambient heat rejection unit, an ejector/expansion valve, separator, evaporator and throttle valve. Subcritical CO_2 is compressed in the compressor where its temperature and pressure rise. An isobaric heat addition process is followed firstly in a recovery heat

exchanger and then in a parabolic solar collector. An auxiliary heater is used in the downstream of the solar collector to supplement solar heat, if required. The high temperature, high pressure CO_2 expands in a turbine-generator to produce electricity. The CO_2 exhaust from the turbine is cooled in the recovery heat exchanger by transferring heat to the incoming CO_2 in the cold stream. Then it is cooled further in the process heat exchanger where the heat rejected is utilised for milk pasteurization process. Finally it is cooled further in the ambient heat rejection unit by rejecting heat to the ambient air. Depending upon the configuration (C1 or C2), the CO_2 from the ambient heat rejection unit flows into the evaporator either through the ejector or through the expansion valve. In the evaporator, CO_2 vaporizes by cooling milk and is then compressed in the compressor to complete the cycle. From the milk side, the chilled milk to be pasteurized flows initially through the milk heat exchanger, where it extracts heat from the hot pasteurized milk coming from the gas cooler. It is then pasteurized in the gas cooler recovery heat exchanger, cooled in the milk heat exchanger and is chilled back to the initial temperature in the evaporator.

In C1 configuration as shown in Fig. 1a, CO_2 coming from the ambient heat rejection unit acts as the primary flow in the ejector where it entrains the secondary vapour from the evaporator. The primary and secondary flows get mixed in a constant area mixing chamber to attain a stable pressure. Then the primary-secondary mixture enters the diffuser where its pressure is increased at the expense of reduction in fluid velocity. The pressurized CO_2 from the ejector, which is in 2-phase, enters the separator where it is separated into saturated liquid and saturated vapour streams. The saturated vapour stream (primary fluid) flows back into the compressor. The saturated liquid (secondary fluid) enters the evaporator through a throttle valve. In the evaporator, the low pressure CO_2 absorbs heat from milk, gets vaporized and is entrained back into the ejector to complete the cycle. Figure 1b shows the complete process on T-s diagram for configuration C1.

In C2 configuration as shown in Fig. 2a, CO_2 coming from the ambient heat rejection unit enters an expansion valve and expands, thereby reducing its pressure and temperature to subcritical conditions. The CO_2 flowing through the evaporator absorbs the heat and gets vaporized to produce the cooling effect for chilling of milk. Fig. 2b shows the complete process on T-s diagram for configuration C2.

III. THERMODYNAMIC MODEL

A thermodynamic model of the trigeneration system is developed based on the mass and the energy conservations. In order to simplify the model, following assumptions are made.

- The system operates in steady state.
- The pressure drop in the process heat exchanger, ambient heat rejection unit, solar collector, separator, evaporator and the connection tubes are neglected.
- Milk to be processed enters the dairy plant at 4^0C and leaves the dairy plant after pasteurization and chilling at 4^oC.

- The results presented are calculated for total refrigerant flow rate (Primary+Secondary in case of C1) of 1kg/s.
- Secondary nozzle pressure drop in C1 configuration is assumed as 30 kPa [7].
- Only the process heat exchanger is utilised for pasteurization of milk
- The net power output from the system is always positive.
- The results presented here are based on the consideration that the evaporator load for chilling of milk exactly matches with heating load in the process heat exchanger for pasteurization of milk.
- For C1 configuration, the motive nozzle, mixing chamber and diffuser efficiencies are taken as 0.8, 0.8 and 0.75 respectively.

Each component in the system is treated as an independent control volume. The conservation of mass and energy principle for each component can be expressed as:

$$\Delta_{out}^{in}\left(\sum_i m_i\right) = 0 \tag{1}$$

$$\Delta_{out}^{in}\left(\sum_i m_i . h_i\right) + \Delta_{out}^{in}\left(\sum_j Q_j\right) + \Delta_{out}^{in}\left(\sum_k W_k\right) = 0 \tag{2}$$

The isentropic efficiency of the compressor is calculated from the following empirical correlation developed by Robinson & Groll [8]

$$\eta_c = 0.815 + 0.022\left(\frac{P_{high}}{P_{low}}\right) - 0.0041\left[\left(\frac{P_{high}}{P_{low}}\right)^2\right] + 0.0001\left[\left(\frac{P_{high}}{P_{low}}\right)^3\right] \tag{3}$$

For C1 configuration, the mathematical model of the ejector is given below.

Motive nozzle: By using the definition of motive nozzle isentropic efficiency; the enthalpy at the nozzle exit is given by

$$\eta_m = \frac{h_{motive,out} - h_{motive,in}}{h_{motive,out,is} - h_{motive,in}} \tag{4}$$

where, $h_{motive,out,is} = h\left(s_{m,in}, P_{m,out}\right)$

The velocity at the exit of the nozzle can be found by:

$$h_{motive,in} = h_{motive,out} + \frac{u^2_{motive,out}}{2} \tag{5}$$

Suction chamber: Similar to those for the motive nozzle, equations governing the suction chamber are presented.

$$\eta_s = \frac{h_{suction,out} - h_{suction,in}}{h_{suction,out,is} - h_{suction,in}} \tag{6}$$

where $h_{suction,out,is} = h\left(s_{suction,in}, P_{suction,out}\right)$

The velocity at the exit of the suction nozzle can be found by:

$$h_{suction,in} = h_{suction,out} + \frac{u^2_{suction,out}}{2} \tag{7}$$

Another assumption taken for this study is $P_{motive,out} = P_{suction,out} = P_{mix}$

Mixing Chamber: Applying the conservation of momentum principle in a constant mixing chamber, the speed at the exit of the mixing chamber can be calculated as:

$$u_{mix,out} = \left(\frac{1}{1+k}\right) * u_{motive,out} + \left(\frac{k}{1+k}\right) * u_{suction,out} \tag{8}$$

By applying conservation of energy principle, enthalpy at the exit of the mixing chamber can be found by:

$$h_{mix,out} = \left(\frac{1}{1+k}\right) * \left(h_{motive,out} + \left(k * h_{suction,out}\right)\right) - \left(\frac{u^2_{mix,out}}{2}\right) \tag{9}$$

Diffuser: The enthalpy at the exit of diffuser can be found as:

$$h_{diffuser,out} = \frac{h_{motive,in} + \left(k * h_{suction,in}\right)}{1+k} \tag{10}$$

$$\eta_{diffuser} = \frac{h_{diffuser,out,is} - h_{mix,out}}{h_{diffuser,out} - h_{mix,out}} \tag{11}$$

Pressure at the outlet of ejector is found as:

$$P_{diffuser,out} = P\left(s_{mix,out}, h_{diffuser,out,is}\right) \tag{12}$$

To maintain cycle continuity, quality of the stream leaving the ejector has to obey the equation:

$$x_{diffuser,out} = \frac{1}{1+k} \tag{13}$$

In the above equation, k is the mass entrainment ratio, which is defined as the ratio of the secondary mass flow rate over the primary mass flow rate.

The overall performance of the CCHP is indicated in terms of cooling COP of the transcritical refrigeration cycle (COP_c), efficiency of the Brayton cycle (η_{power}) and COP of overall cycle (*COP*). These are defined as:

$$COP_c = \frac{Q_{evap}}{W_{comp}} \tag{14}$$

$$\eta_{power} = \frac{W_{turb}}{Q_{solar}} \tag{15}$$

$$COP = \frac{W_{net} + Q_{gc1} + Q_{evap}}{Q_{solar}} \tag{16}$$

where the net power output from the system is given by:

$$W_{net} = W_{turb} - W_{comp} \tag{17}$$

Q_{gc1} is the amount of heat which is transferred to the milk for the pasteurization in process heat exchanger. Q_{evap} is the

refrigeration capacity of the evaporator and Q_{solar} is the solar heat input.

Assuming that the entire heat is transferred to the milk for pasteurization process in the process heat exchanger, the milk handling capacity is calculated as:

$$m_{milk} = \frac{Q_{gc1}}{Cp_{milk} * \Delta T_{milk}} \tag{18}$$

where Cp_{milk} is the specific heat of milk and ΔT_{milk} is the temperature difference of milk across the process heat exchanger which depends on the effectiveness of the milk heat exchanger (milk HX). For our study, we have assumed the milk HX effectiveness as 0.9.

IV. RESULTS & DISCUSSION

The effect of turbine inlet temperature, ambient temperature, turbine inlet pressure and process heat exchanger pressure on overall cycle COP, power cycle efficiency and cooling COP are presented below. The values of the operating parameters considered for both the configurations are listed below.

TABLE I: LIST OF PARAMETERS

Parameters	Default Values/Ranges
Turbine isentropic efficiency	0.85
Recovery heat exchanger effectiveness	0.9
Ambient temperature	35^0C / 30^0C-45^0C
Turbine inlet temperature	500^0C / 400^0C-800^0C
Evaporator temperature	2^0C
Cycle maximum pressure (Phigh)	20000 kPa / 16000 kPa-25000 kPa
Process heat exchanger pressure (Pmed)	9000 kPa / 8000 kPa-11000 kPa
Compressor suction pressure (Plow)	4210 kPa at 2^0C
Specific heat of milk	3.93 kJ/kg K

A. Effect of Turbine Inlet Temperature

Fig 3 shows the variation of turbine inlet temperature on overall COP, net electrical power output, power cycle efficiency and solar heat input for both the configurations. Increase in turbine inlet temperature results in a rise in solar heat input resulting in an increase in the turbine work output. Compressor power and refrigeration capacity remain unaffected. As the turbine power increases, the turbine exhaust heat also increases resulting in a rise in total heat output (heating output for milk pasteurization + heat rejected to the ambient). However, the heating output for milk pasteurization remains unaffected thereby keeping the milk handling capacity constant. Moreover, the COP decreases with increase in turbine inlet temperature. So in context of dairy application, an increase in turbine inlet temperature is advisable only when there is a need for generating more electricity to meet the parasitic loads. Power cycle efficiency shows a marginal increase with rise in turbine inlet temperature for both the configurations.
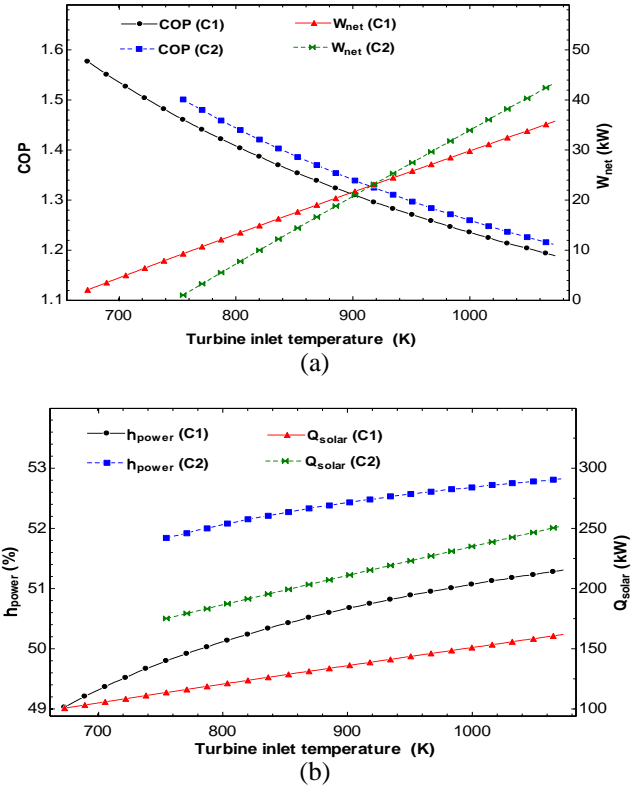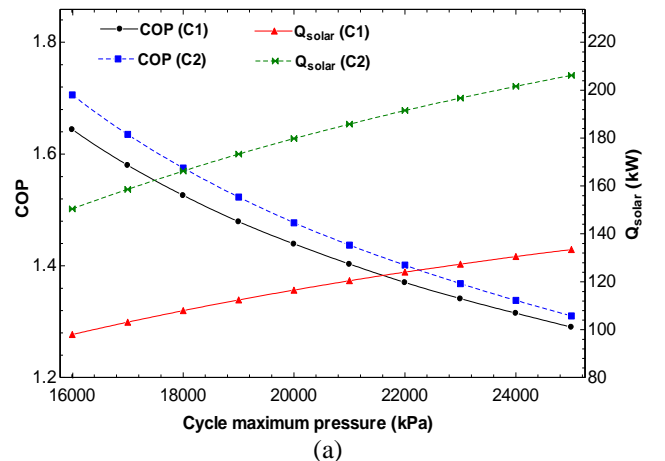

Fig. 3. Effect of turbine inlet temperature on (a) Overall COP & Net electrical power output and (b) Power cycle efficiency & Solar heat input.

B. Effect of Cycle Maximum Pressure

The COP for both the configurations decreases with increase in cycle maximum pressure as shown in Fig 4. The refrigeration effect is independent of the cycle maximum pressure and remains unchanged. Compressor power and turbine power output (electrical power) increase due to increase in pressure ratio across the components. Increase in cycle maximum pressure raises solar heat required to raise the temperature of CO_2 due to increase in enthalpy difference across the solar collector. Increase in both the turbine work output and solar heat supplied results in a marginal rise in power cycle efficiency as the cycle maximum pressure is increased. Similar to the case of increase in turbine inlet temperature the heating output for milk pasteurization remains unaffected thereby keeping the milk handling capacity constant.
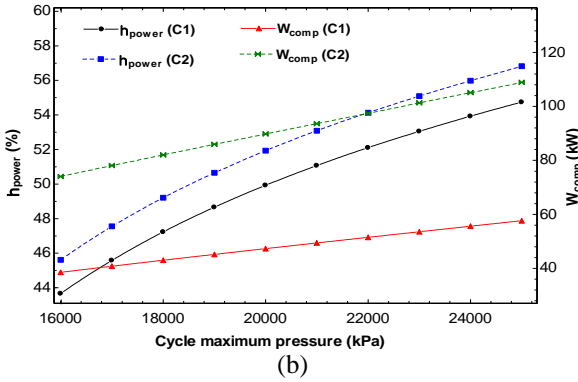
Fig. 4. Effect of cycle maximum pressure on (a) overall COP & solar heat input and (b) power cycle efficiency & compressor power input.
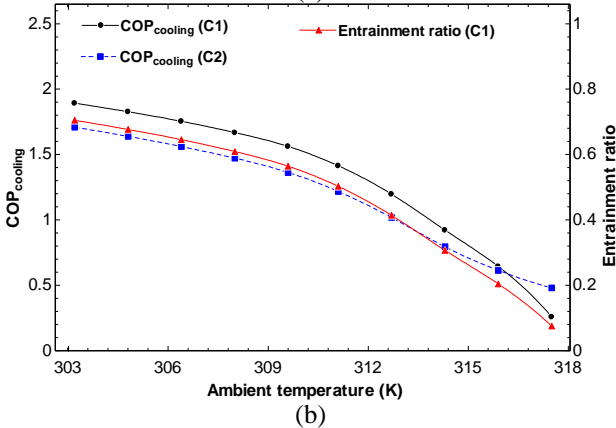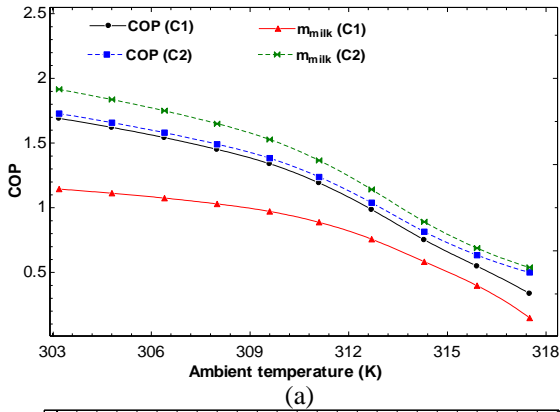

(a)

(b)

Fig. 5. Effect of ambient temperature on (a) Overall COP & milk handling capacity and (b) Cooling COP & entrainment ratio.

A. Effect of Ambient Temperature

Fig. 5 shows the effect of ambient temperature on the overall COP, milk handling capacity, cooling COP for both the configurations. In case of C1 configuration the effect of ambient temperature on entrainment ratio is also shown in Fig 5(b). As the temperature increases, the COP for both the configurations decreases. The turbine power output, compressor power input and solar heat output remain unaffected. In C1 configuration, the entrainment ratio decreases with increase in ambient temperature. As a result there is a decrease in secondary mass flow rate flowing through the evaporator. The vapour fraction of CO_2 at evaporator inlet increases resulting in a reduction in refrigeration effect in both the configurations. This results in a decrease of cooling COP with increase in ambient temperature. Specific heat variation near the critical zone results in a significant reduction in refrigeration output and

heating user output around the critical temperature region of supercritical CO_2. Heating user output reduces due to decrease in enthalpy difference across the gas cooler. Under similar operating conditions increase in ambient temperature results in a decrease in milk handling capacity for both the configurations.
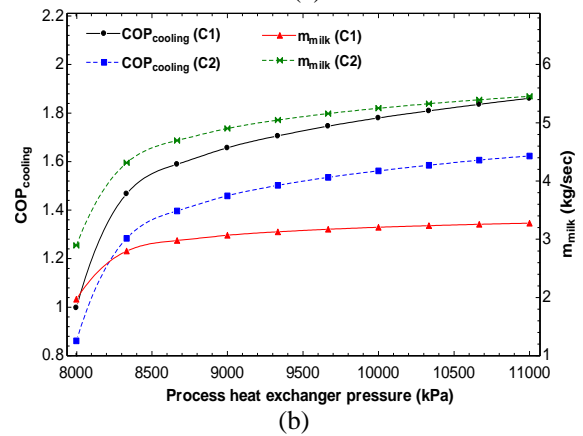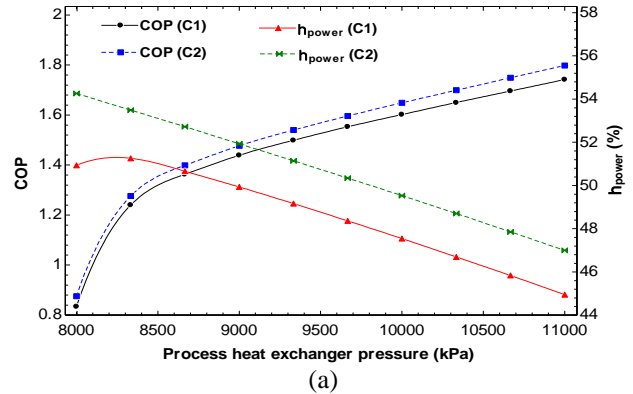

(a)

(b)

Fig. 6. Effect of process heat exchanger pressure on (a) overall COP & power cycle efficiency and (b) milk handling capacity & cooling COP

B. Effect of Process Heat Exchanger Pressure

Fig. 6 shows the effect of process heat exchanger pressure on overall COP, cooling COP, power cycle efficiency and milk handling capacity for both the configurations. As the pressure increases the COP also increases for both the configurations. The vapour fraction of CO_2 at evaporator inlet gets reduced resulting in an increase in the refrigeration effect and thereby increase the cooling COP. The entrainment ratio increases with rise in process heat exchanger pressure. Similarly an increase in pressure increases the heating output due to rise in enthalpy difference across the gas cooler whereas turbine power output decreases due to reduction in enthalpy difference between the inlet and outlet of the turbine. This also results in higher turbine exhaust temperature, thereby reducing the solar heat requirement. The reduction in turbine power output and solar heat requirement results in a reduction of power cycle efficiency. Under similar operating conditions increase in process heat exchanger pressure results in an increase in milk handling capacity for both the configurations. For the set of default values of turbine inlet temperature, ambient temperature, evaporator temperature, cycle maximum pressure and process heat exchanger pressure mentioned in Table I, the output values obtained in both the configurations are listed below in Table II.

TABLE II: OUTPUT VALUES AT DEFAULT PARAMETERS

Configuration	Wturb (kW)	Wcomp (kW)	Qevap (kW)	Qg (Qgc1+Qamb) (kW)	Milk handling capacity (kg/sec)	Qsolar (kW)	COP	Cooling COP	Power efficiency (%)
C1	58.19	47.3	78.38	184	3.068	116.5	1.439	1.657	49.93
C2	93.37	89.79	131	307.2	4.902	179.8	1.477	1.459	51.93

The corresponding pressure lift and entrainment ratio obtained for C1 configuration are 692 kPa and 0.6046 respectively. The milk handling capacity for C1 configuration can be increased by increasing the refrigerant mass flow rate. From the table it can be seen that for each kW of heat supplied by the solar collector, the milk handling capacity of the configuration with ejector (C1) is marginally (about 3%) smaller than the one with throttle valve (C2), however, the electrical power produced per kW of solar heat is much larger (about 360 %).

V. CONCLUSIONS

Based on the thermodynamic analysis of the two trigeneration configurations (i.e ejector based system C1 and throttle valve based system C2), the following conclusions can be drawn.

1. From a dairy application point of view, in a trigeneration system the ambient temperature and process heat exchanger pressure are the dominant factors in determining the milk handling capacity.

2. Higher values of turbine inlet temperatures are required if the need for electrical power is more

3. Under almost similar milk handling capacity conditions, an ejector based system results in lower compressor power and much higher net electrical power output compared to the throttle valve based system.

REFERENCES

[1] D. W. Wu and R. Z. Wang, "Combined cooling, heating and power: A review," *Prog. Energy Combust. Sci.*, vol. 32, pp. 459–495, 2006.
[2] C. Z. Li, Y. M. Shi, and X. H. Huang, "Sensitivity analysis of energy demands on performance of CCHP system," *Energy Convers. Manag.*, vol. 49, no. 12, pp. 3491–3497, 2008.
[3] J. Wang, Y. Dai, L. Gao, and S. Ma, "A new combined cooling, heating and power system driven by solar energy," *Renew. Energy*, vol. 34, no. 12, pp. 2780–2788, 2009.
[4] V. Ravindra and M. Ramgopal, "Thermodynamic analysis of a solar assisted combined cooling, heating and power system with different cooling cycle configurations," *INAE Letters*, vol. 3, issue 2, pp. 107-113, 2018.
[5] N. Fumo, L. M. Chamra, and V. Bortone, "Potential of solar thermal energy for CCHP systems," in *Proc. the ASME 2009 3rd International Conference of Energy Sustainability*, pp. 1–7, 2009.
[6] G. Patel and S. Chaudhary, "Performance comparison of ejector expansion refrigeration cycle with throttled expansion cycle using R-170 as Refrigerant," *International Journal of Scientific and Research Publications*, vol. 4, issue 7, pp. 1-5, 2014.
[7] D. Li and E. A. Groll, "Transcritical CO_2 refrigeration cycle with ejector expansion device," *Int. J. Refrigeration*, vol. 28, pp. 766-773, 2005.
[8] D. M. Robinson and E. A. Groll, "Efficiencies of transcritical CO_2 cycles with and without an expansion turbine," *Int. J. Refrigeration*, vol. 21, pp. 577–589, 1998.

Vutukuru Ravindra received B.Tech. degree in mechanical & energy engineering from VIT University, India in 2011 and M.Tech. degree from NIT Rourkela, India in 2015. At present he is a PhD student in School of Energy Science & Engineering at Indian Institute of Technology, Kharagpur, India. His research interests are transcritical CO_2 based systems, Solar thermal and trigeneration systems. Currently, he is working on solar assisted trigeneration systems using CO_2 as working fluid

M. Ramgopal is a professor in Mechanical Engineering Department at Indian Institute of Technology, Kharagpur. His research areas include refrigeration & air conditioning, CO_2 based systems, Solid sorption systems and Thermal comfort studies.