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Parametric Quantification of Low GWP Refrigerant for Thermosyphon Driven Solar Water Heating System

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Abstract

Modern lifestyle, industrialization and economy thrive on energy which is getting expensive overtime. Thermosyphon based systems are getting attraction for their promising heat transfer efficiency and zero energy utilization. Refrigerants having ozone depletion potential (ODP) and high Global Warming Potential (GWP) have been banned or under time bared permission under Montreal (1987) and Kyoto (1997) protocols. We have devolved a Refrigerant Parametric Quantification (RPQ) method for the choice of optimal refrigerant for density driven solar water heaters. A set of 29 refrigerants are simulated Using REFPROP under various temperature and pressure conditions. The optimal parameters of thermosyphon system are identified from governing equations, international environment safety protocols and open literature. The proposed RPQ method shows most appropriate refrigerant for given temperature range. In second part, the proposed system is simulated in TRNSYS using forced circulation method. In the end, a glass evacuated tube collector is developed and tested on the principal of thermosyphon, employing the best refrigerant emerged from simulation study.

Keywords: *Thermosyphon, natural refrigerants, thermodynamics, supercritical fluid*.

1. Introduction

Natural convection based systems are getting attraction for their promising heat transfer properties, better thermal performance and zero energy utilization. Thermosyphon is green energy device widely used in water and space heating, Third Generation (3G) telecommunication equipment cooling [1], power generation aerospace applications [2], heat recovery from sewage gases, food storage units [3], auto industry, nuclear reactors cooling and many more [4]. Nuclear power plants require continuous cooling with operation stability and safety. An earthquake in March 2011 crippled six nuclear reactors and storage pools in Fukushima due to failure of active heat removal system. Indeed, thermosyphon driven system have emerged as sustainable heat transfer device which could substantially be integrated for cooling of 439 active nuclear reactors across the world as well as for next generation nuclear plants [5]. Thermosyphon devices are characterized as single and two phase system by virtue of their operation. Many researchers theoretically and experimentally showed that thermal performance and overall efficiency of Two Phase Closed Thermosyphon (TPCT) system is better than conventional single phase thermosyphon solar water heaters in terms of fouling, scaling, freezing, corrosion, life and overall performance [6-9]. B.R. Chen and his co-researchers experimentally demonstrated the thermal performance of TPCT system using flat-plate solar collector. They used alcohol as mediating fluid with 40% filling ratio and demonstrated an efficiency of 63%. [9]. Redpath et al showed that evacuated glass tube solar collector have better thermal performance over flat plate [10-11]. Many researchers used NH3, H2O, CO2, R-11, R-22, R134a, R-407 C, R-410A, water-ethylene glycol as mediating fluid in TPCT devices. These refrigerants have been examined in different working environment and experimental setups [9; 12; 13-19].

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2. Next Generation Refrigerant for TPCT

Historically, refrigerants are being used in cooling and heating devices for last two centuries. Oliver Evans (1805) presented the idea of refrigeration cycleusing ether as refrigerant, and Jacob Perkin successfully implemented the proposed refrigeration machine in 1834 under British patent no. 6662 [20]. Natural refrigerants NH3, CO2, SO2, H20, Hydrocarbons (HCs) remained widely used mediating fluids from 1830-1930s. These refrigerants were toxic (NH3, SO2), flammable (HCs) and exhibit high working pressure (CO2) due to which accidents were common [21-22]. Thomas Midgley and Albert Henen [23] invented first synthetic refrigerant Chlorofluorocarbon (CFC) in 1929which was commercialized in 1932 by Dupont de Nemours [24-25]. Natural refrigerants were replaced by synthetic ones by virtue of their better thermal performance, reactivity and safety [26]. Prof. Curtzen, Molina and Rowland found that anthropogenic CFCs and HCFS cause decomposition of stratosphere ozone in presence of sun light. CFCs have long atmospheric life and destroy the stratospheric ozone layer which shields earth from harmful ultraviolet sun radiations. Taking Ozone depletion as view point, the Montreal Protocol (1987) banned the production and use of CFCs and HCFCs after 1995 on account of their high ODP and long atmospheric life. Kyoto Protocol (1997) recommended complete phase out of HCFC by 2020 for developing countries and 2030 for developed countries [27-28].This yielded start of third generation of refrigerants (1990-2010s) with focus on zero ODP and low GWP. The past, present and future outlook of refrigerant is shown in Fig. 1.

According to fourth assessment report of Intergovernmental Panel on Climate Change (IPCC), the presence of anthropogenic greenhouse gases in atmosphere is scientifically considered as major cause of increased air temperature which in turns elevated melting glaciers and thus raised ocean level. Kyoto Protocol (1997) and F-Gas laws (2006 and amend.-2014) quest for new refrigerants having zero ODP, low GWP, high efficiency with lower atmospheric life time [30].The European F-gas law implemented the use of low GWP refrigerants (≥150) in future Mobile Air-conditioning (MAC) devices from January 2015[31-32].

Scientists are in search of set of an adequate refrigerant for thermosyphon and heat-pump devices which is environmental friendly and have higher efficiency. In this regard many researchers have reviewed the impact of environment benign refrigerant for utilization in next generation heat transfer applications [22; 24; 34-36]. Saleh et al. compared the performance of HCFC-22 with low GWP synthetic and natural refrigerants employing physical BACKBONE equation [37]. Several refrigerant optimization studies based on data mining techniques for thermo physical properties [38], neural networks [39] as well as hybrid formulae for thermodynamic properties [40], cost based methods [41], and comparison of operating performance [42].

The choice of optimal refrigerant in thermosyphon driven solar water heating system is evaluated in present study. We selected ASHREA envisaged natural refrigerants [43], some of their low GWP blends and synthetic refrigerants [44].The refrigerants are chosen with zero ODP, low GWP (150) in accordance with European standard and very low GWP in accordance with UNEP Technical Options Committee for Refrigeration, Air-conditioning, and Heat Pumps. The selected refrigerants are shown with their technical characteristic in Table 1.

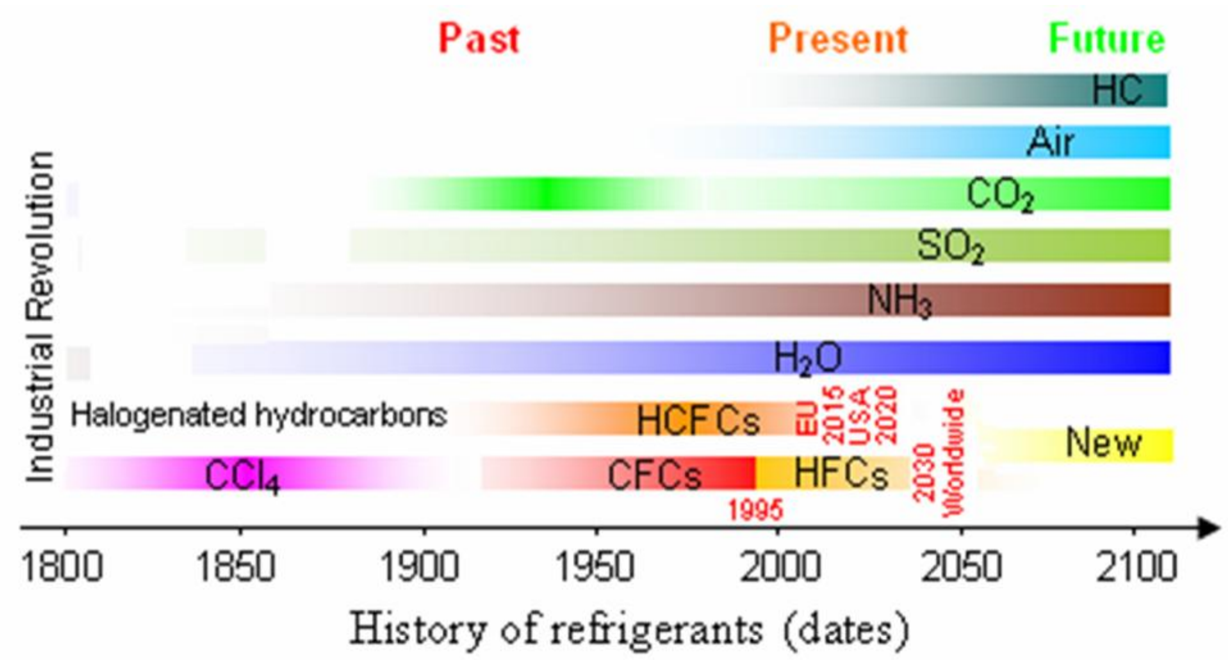

Fig. 1 Brief Histogram of Synthetic and Natural Refrigerants [29]

Table 1. List of Selected Refrigerants [44-46]

	Refrigerants	Chemical composition	Molecular Weight	Critical Point (Tc) (îC)	Critical Point Pc (Bar)	Critical Point ° (Kg/m^3)	T_{NBP} (îC)	ODP	$GWP_{100\ Years}$	Life (Years)	ASHREA -34 Safety Class	Corrosiveness
Natural	Methane (R-50)	H3C-CH3	16.04	82.5	45.9	162.6	-161.4	0	25	12	A3	N
Natural	Ethane (R-170)	CH2C=CH2	30.07	32.1	48.7	206.1	-88.58	0	5.5	12	A3	N
Natural	Propane (R-290)	CH3-CH2-CH3	44.1	96.7	42.5	220.4	-42.11	0	3.3	12	A3	N
Natural	Butane (R-600)	CH3-CH2-CH2-CH3	58.12	151.9	37.9	228.0	-0.49	0	4.0	12	A3	N
Natural	Isobutene(R-600a)	CH3-CH-CH3-CH3	58.12	134.6	36.2	225.5	-11.74	0	3	12	A3	N
Natural	Ethylene (R-1150)	H2C=CH2	28.05	9.2	50.4	214.2	-103.7	0	3.7	12	A3	N
Natural	Propylene (R-1270)	CH2=CH-CH3	42.08	91.0	45.5	229.6	-47.61	0	1.8	12	A3	N
Natural	Water (R-718)	H-O-H	18.01	373.9	220.6	322.0	99.97	0	0	0.026	A1	Y
Natural	Ammonia (R-717)	N-H3	17.03	132.2	113.3	225.0	-33.32	0	0	0.019	B2L	Y
Natural	Carbon dioxide(R-744)	O=C=O	44.01	31.1	73.7	467.6	-78.46	0	1	1	A1	N
Natural	Air (R-729)	Nitrogen+ Oxygen + Argon	28.97	140.6	37.8	342.8	-194.2	0	0	∞	A1	N
Blends of Natural	R-432A	R-1270/R-E170 (80/20)	44.82	97.3	47.6	240.1	-46.4	0	< 3	9.6	A3	N
Blends of Natural	R-433A	R-1270/R-290 (30/70)	43.47	94.4	43.4	222.2	-44.5	0	< 3	12	A3	N
Blends of Natural	R-433B	R-1270/R-290(5/95)	43.99	96.3	42.6	220.6	-42.6	0	< 3	12	A3	N
Blends of Natural	R-433C	R-1270/R-290(25/75)	43.59	94.7	43.3	221.8	-44.2	0	< 3	12	A3	N
Blends of Natural	R-436A	R-290/R-600a (56/44)	49.33	115.9	42.7	220.3	-26.1	0	< 3	12	A3	N
Blends of Natural	R-510A	R-E170/R-600a (88/12)	47.24	125.6	51.8	268.5	-25.1	0	< 3	1.45	A3	N
Syntheti	R-1234yf	CF3CF=CH2	114.04	97.7	33.8	475.5	-29	0	4	0.03	A2L	N
Syntheti	R-1234ze	Trans, CHF=CHCF3	114.04	109.3	36.3	486	-20	0	6	0.03	A3	N
Syntheti	R-152a	F2HC-CH3	66.05	114.0	45.1	368.0	-24	0	124	1.4	A3	N

3. Physical Model and Governing Equation

The thermosyphon loop consists of Evacuated Glass Tube Solar Collector (EGTSC) with U shaped heat removal tubes. Heat source (EGTSC) is at lower side and heat sink is elevated to the top in the form of water tank as shown in Fig. 2, The heat removal loop has two-dimension geometry, up-rise that drive the heated fluid to condenser where it gives off heat and returns through down-comer using thermosyphon. The present model has ability to handle high pressure filling and its operation sustainability is discussed in our previous work [47-48].

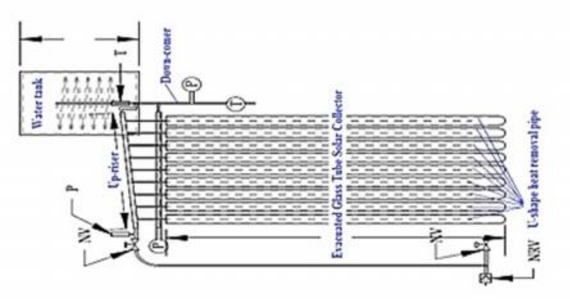

Fig. 2 Schematic of the Physical Model (Thermosyphon Driven Solar Water heater)

Natural convection occurs under buoyancy forces which cause fluid motion due to density difference in presence of gravity. Considering a laminar, steady state, two dimensional free convection system; the buoyancy forces are generated due to density difference caused by the temperature gradient. We know;

Continuity equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

For momentum equation

$$\rho\left[u\frac{\partial u}{\partial x}+v\,\frac{\partial u}{\partial y}\right]=\ -\rho g-\ \frac{\partial p}{\partial x}+\mu\frac{\partial^2 u}{\partial y^2} \tag{2}$$

Energy equation

$$\rho C_p\left[u\frac{\partial T}{\partial x}+v\,\frac{\partial T}{\partial y}\right]=\ k\,\frac{\partial^2 u}{\partial y^2} \tag{3}$$

At boundary layer edge; $u = 0$; $y \rightarrow \infty$

$\frac{\partial p}{\partial x}=-\rho_\infty g$; ρ_∞ is fluid density outside the boundary layer whilst $\frac{\partial p}{\partial x}\approx 0$

$$-\rho g-\ \frac{\partial p}{\partial x}=\left(\rho_\infty-\ \rho\right)g \tag{4}$$

The volume expansion coefficientβ may be related to density difference

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_P = -\frac{1}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_P \; 1/_K; \quad \beta \approx \frac{1}{\rho}\frac{\Delta P}{\Delta T}$$

$$\Delta\rho = -\rho\beta\Delta T$$

$$(\rho_\infty - \rho) = \beta\rho\,(T_\infty - T) \qquad (5)$$

Substituting the value from equation (4) into equation (5)

$$-\rho g - \frac{\partial p}{\partial x} = \beta\rho(T_\infty - T)g.$$

Momentum equation for boundary layer becomes

$$u\frac{\partial u}{\partial x} + v\,\frac{\partial u}{\partial y} = \beta g\rho(T_\infty - T) + v\,\frac{\partial^2 u}{\partial y^2} \qquad (6)$$

Let's us define following dimensionless quantities

$$X = \frac{x}{L} \qquad Y = \frac{y}{L} \qquad U = \frac{u}{u_0} \qquad V = \frac{v}{u_0} \qquad \theta = \frac{T - T_\infty}{T_w - T_\infty}$$

The governing equations becomes

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \qquad (7)$$

$$U\frac{\partial U}{\partial X} + V\frac{\partial V}{\partial Y} = \frac{G_r}{{R_e}^2}\theta + \frac{1}{R_e}\frac{\partial^2 u}{\partial y^2} \qquad (8)$$

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{1}{R_e P_r}\frac{\partial^2 u}{\partial y^2} \qquad (9)$$

Equations (7) to (9) explain the free convection. The flow regime in thermosyphon systems is characterized by Gr as

$$G_r = \frac{\text{Buoyancy forces}}{\text{Viscous forces}} = \frac{g\beta\Delta TV}{v^2} = \frac{g\beta(T_h - T_c)L^3}{v^2} \qquad (10)$$

Where Ra can be expressed in terms of Gr and Pr

$$R_a = G_r P_r = \frac{g\beta(T_h - T_c)L^3}{v^2} P_r \qquad (11)$$

Then Nu is written as

$$N_u = \frac{hL_c}{k} = C(G_r P_r)^n = C{R_a}^n$$

Nu for vertical plate is represented by

$$N_u = \left(0.825 + \frac{0.387 {R_a}^{\frac{1}{6}}}{\left[1 + \left(0.492/_{P_r}\right)^{9/_{16}}\right]^{8/_{27}}}\right)^2$$

; For an inclined plate at angleθ; g is replaced by g cosθ

The heat transfer rate in natural convection is given by

$$h_x = \frac{q_w}{T_w - T_\infty}$$

$$\dot{Q}_{convection} = hA(T_h - T_c) \quad \text{Watt}$$

$$\dot{Q}_{convection} = \frac{N_u \,.\, k.\, A(T_h - T_c)}{L_c}$$

For turbulent flow, convective flow of super-critical fluid, the Ga may be defined under general Normalization Group (RNG) κ-ε by Lin Chen et. al. is[5]

$$G_r = \frac{\rho^2 g\beta QHL^3}{\mu^2 AC_p} \qquad (12)$$

Where 'Q' is input solar heat, 'H' is height of thermosyphon loop having diameter 'D' and area 'A'.

4. Qualitative Parametric Quantification

The governing equation shows the thermodynamic and heat transport parameters that have significant impacton the performance of thermosyphon driven EGTSWH.A parametric quality factor in terms of thermal conductivity, density difference, kinematic viscosity, critical pressure, volumetric expansivity, specific heat, Prandtl number, GWP, corrosiveness, toxicity and flammability is optimized for suitable refrigerant in density driven EGTSWH system, likewise qualitative standardization technique developed in our preceding work[47].Under this user can normalize the parameters from 1-0 by dividing the parametric value with highest good in the list. In case, the lowest parametric value is desirable, the user can normalize it by dividing with actual parametric quantity. For instance, the density difference of CO2 (R-744) is 406.88 kg/m3 in the range of -20 to 30 °C compared to 161.3 kg/m3 of R-1234yf. Considering the table 2, the values can be normalized to highest density difference resulting 1(R-744) and 0.37 (R-1234yf). After calculating normalized value, the refrigerant parametric quantification (RPQ) may be obtained by

$$RPQ(\%) = \frac{\rho_a + k_a + v_o - P_{r\,a} + \beta_a + Pc_o + C_{P\,a} + GWP_{CO_2} + FT_a + C_a}{\rho_a + k_m + v_a + P_{r\,m} + \beta_m + P_{C\,a} + C_{P\,m} GWP_a + F_m + C_m} \times 100 \qquad (1.13)$$

Subscript 'a' refers to actual value, 'm' to possible maximum value, 'o' to optimal value undertaken for consideration.

Table 2. Convection Driven Heat Transfer Refrigerant Parameters, Range and Optimum Values

Parameter	Units	Optimal value / Normalized parameter
ρ	Kg/m^3	Normalized to highest density difference.
[illegible]	mW/m-k	Normalized to highest average thermal conductivity.
ν	cm^s/s	Normalized to lowest average kinematic viscosity.
[illegible]	---	Normalized to highest average Prandtl number.
β	1/K	normalized to highest average volumetric expansivity
Pc	bar	36.29 (critical pressure of Iso-Butane (R-600a) /critical pressure of specific natural refrigerant.
C_P	KJ/Kg-k	Normalized to highest average specific heat.
GWP_{100}	No.	Normalized to CO_2 =1.
C[illegible]	---	1(Non-corrosive), 0.5 (corrosive with copper or Steel), 0 (Corrosive with copper & steel).
FT	ASHREA34 safety standard	[Toxicity (0.5) + Flammability (0.5)] A1=1, A2L=0.87, A2=0.75, A3=0.5 B1=0.5, B2L=0.37, B2=0.25, B3=0.

The proposed system shown in fig. 2 is simulated for three sets of working temperature -20 to 30 °C (Space heating) [49], 30 to 70 °C (domestic water heating) [13] and 70 to 120 °C (commercial heating)[15] for all selected refrigerants at supercritical pressure. The choice of supercritical pressure is adopted by many researchers [15; 50] in thermosyphon driven water heaters and circulation loop owing to dramatic variation in density causing strong convective flows and excellent heat transfer performance in supercritical region, as compared to base conditions [51-52].

5. Results

Thermodynamic and heat transfer parameters are calculated using REFPROP [45] whilst environmental and economic parameters are taken from open literature [53]. Fig. 2 represents the optimization results: space heating, domestic water heating and commercial heating application. R-744 (CO2), R-1150 (ethylene) and R-170 (Ethane) shows higher quality factor, respectively, among all refrigerants in space heating range. In domestic water heating applications, R-744 (CO2) again emerged superior whilst R-170 (ethane), R-1234yf, R-718 (water) are notable in context of overall performance. For industrial heating applications, the newly developed synthetic refrigerants (R-1234yf, R-1234ze) and blend of natural refrigerants (R-433A) proved superior and favorable in performance, respectively. The comparison of physical, thermodynamic, heat transfer and environmental properties indicate R-744 (CO2) reveals better performance in the range of -20 to 70 °C. The newly developed synthetic refrigerants (R-1234yf, R-1234ze) have potential utilization in commercial heating system based on thermosyphon. The results above are also supported by simulation and experimental reports in literature e.g., by [5; 15; 31; 35-36;54].

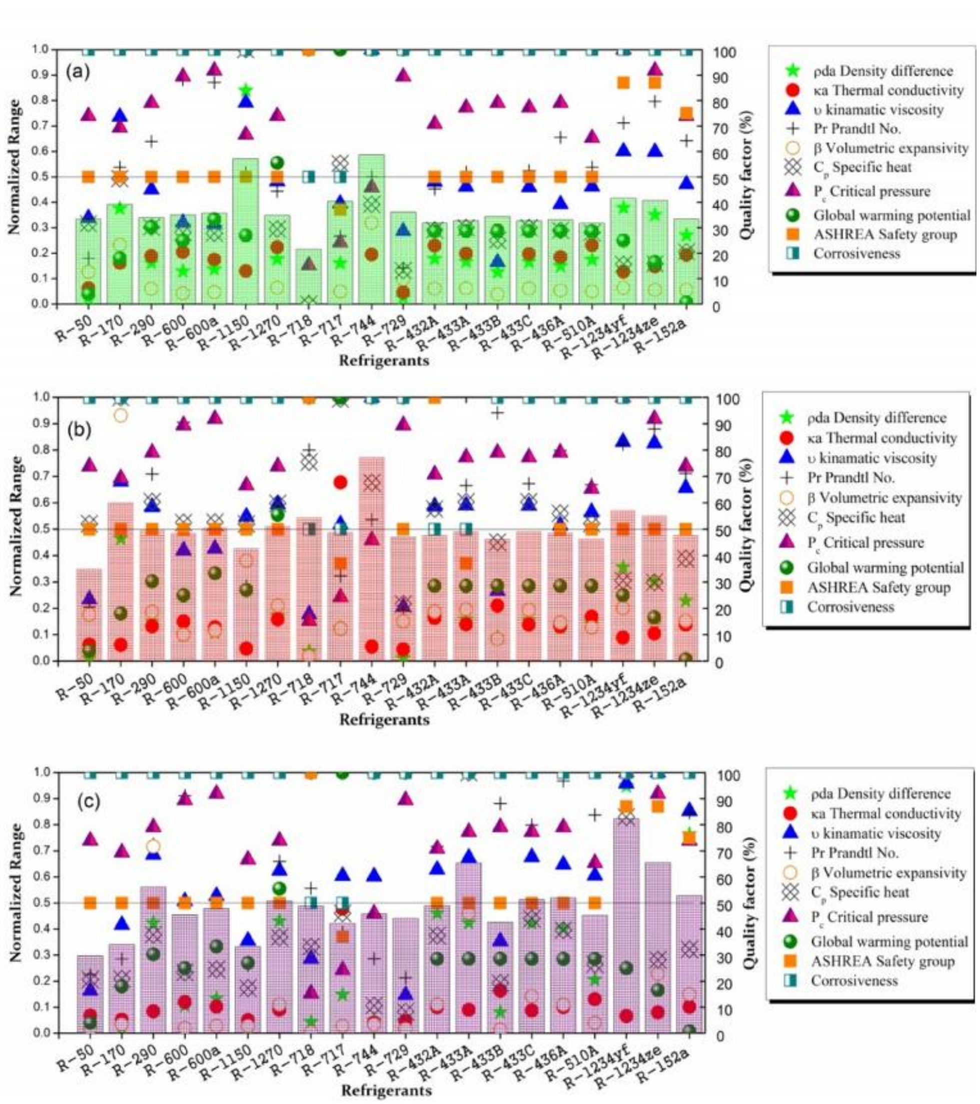

Fig. 2 Refrigerant Parameters Normalized By Optimal Values Defined in Table 2: (a) Space Heating; (b) Domestic Water Heating; (c) Commercial Heating Applications

6. TRNSYS Simulation Study

TRNSYS is an interactive transient simulation study software package used to model and simulate the energy systems like Solar PV, thermal, wind energy, engines, batteries and much more. The software is provided with a flexible graphical based simulation modules having flexibility to change the parameters and their corresponding transient effects can be visualized in graphical forms.

Many authors used this comprehensive tool to simulate the solar water heating system and found it up to the mark. Y. Chen [55] and co researcher performed a dynamic simulation study on evacuated glass tube solar water heating system using CO_2 as mediating fluid for combining heat and power production for solar insolation area. Safa et al. measured the performance of ground source heat pump using TRNSYS and validated by field trials [56].

Abdunnabi et al. validated the results of forced circulated solar water heating system using TRNSYS. In present work, an evacuated glass tube solar collector with water tank at top is modeled and simulated in TRNSYS solar water heating module for a weather pattern of Lahore (31.5497° N, 74.3436° E). The present system is shown in Fig.3 is similar to our earlier work [57] with addition of a pump for forced circulation of mediating fluid (CO_2 in this case). The detail of simulation components and parameters of TRNSYS simulation is given in Table 3.

Table 3. TRNSYS Simulation parameters.

Simulation module / Sub Routine	Description
Evacuated glass tube solar collector (type 71)	No. of collector: 1 Collector area: 1.475 m2 Collector Angle: 45° $\eta = ao - a1\frac{\Delta T}{I_T} - a2\frac{(\Delta T)^2}{I_T}$
Weather data (type 15)	Location: Lahore (31.5497° N, 74.3436° E) Day Selected: 1 October (11:00 to 2:30) An external file PK-Lahore 416400 is called from built-in weather data.
Heat storage tank (type 534 coiled)	This subroutine models a fluid-filled, constant volume storage tank with immersed heat exchangers. Storage tank. Fluid in tank: Water (R-718) Fluid in Heat exchange: CO2 (R-718) Tank volume: 0.5 m3; Tank height: 1.5 m
Pump	Maximum power: 240KJ/hr Flow rate: variable
Working Fluid	CO2 (R-744).
Unit Conversion Routine (Type 57)	Use to convert the out to user specified units
Plotter 65b	Use to plot output results using user specified.

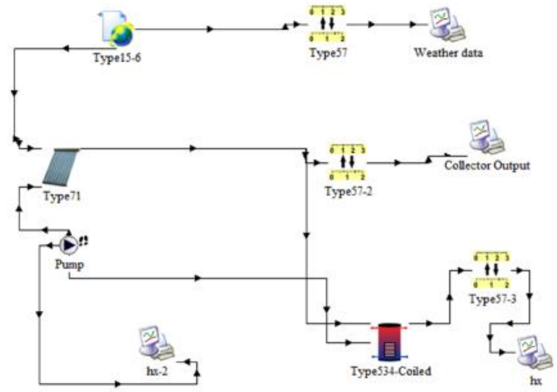

Fig. 3 Schematic diagram of simulation in TRNSYS

7. TRNSYS Simulation Results

The proposed experimental system is simulated for a typical day (1 October 2014) based on weather data of Lahore generated by METEONORM. The results are shown in Fig. 4.

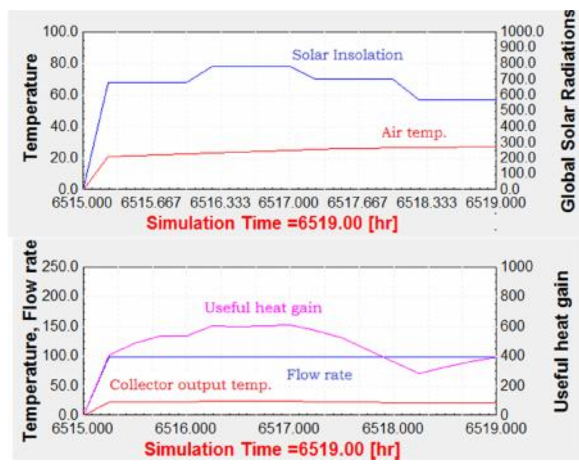

Fig. 4 Simulation results of solar water heating system

The liquid pump feed CO2 at a pressure of 58 bar to solar collector with a constant flow rate of 100 kg/hr while consuming maximum power at rate of 240KJ/hr. The resultant collector outlet temperature is 25°C, useful heat gain of 600W with 0.78 heat recovery factor during the experiment time. With 100% increase in flow rate the collector outlet temperature is decreased with slight increase in useful gain. When the flow rate is decreased to 10kg/hr, a tremendous increase in collector outlet temperature is observed with reduced heat recovery efficiency (0.70).

We fabricated and developed a gravity driven evacuated glass tube solar water heater using CO2 as mediating fluid. The designed system consists of borosilicate glass evacuated tubes of dimension (1.8m ×0.058 m × 0.047 m). A set of 9 such tubes is mounted on a purpose built aluminum stand at an inclination of 45° from ground level fixed at a 33.6518° N, 73.1566° E. Heat collected inside the collector is removed through U-shaped copper tubes of outer diameter 0.00636m and wall thickness 0.00176m. The U-shaped heat removal tubes are further connected to hot and cold header (manifolds). The hot header is connected to up-riser connecting water tank which in turns coupled to down comer and ways back to cold header completing the loop as shown in Fig. 5.

Supercritical stage of CO_2 (31.1 °C, 73.3 bar) is succeeded by filling the system at 68-bar pressure, while temperature gradient is gained inside the solar collector. A special designed semi-circle aluminum fin (thermal diode) is placed inside the evacuated glass tube to collect the heat and deliver to refrigerant carrying copper tubes. Special arrangement in manifolds discussed in our previous work inside the evacuated tubes makes it possible to stop reverse thermosyphon [48].

The system was tested on typical day October 1, 2014 and the results are presented in Fig. 6. The CO_2 was filled after evacuation of the solar water heating system at 68 bar and the system was tested from 11:00 to 14:30 hours. Solar insolation and ambient temperature data is taken from metrological department of CIIT, temperature and pressure observation are taken periodically from the system gauges. The useful heat gain and solar collector efficiency are computed from Liangdong's work [58] as

$$Q_u = F'A_c[I_T(\alpha\tau) - U_L(T_f - T_a)]; \quad \eta = \frac{Q_u}{I_{TA_c}} \quad (13)$$

Fig. 5 Picture of Thermosiphon Gravity Driven Solar Water Heater

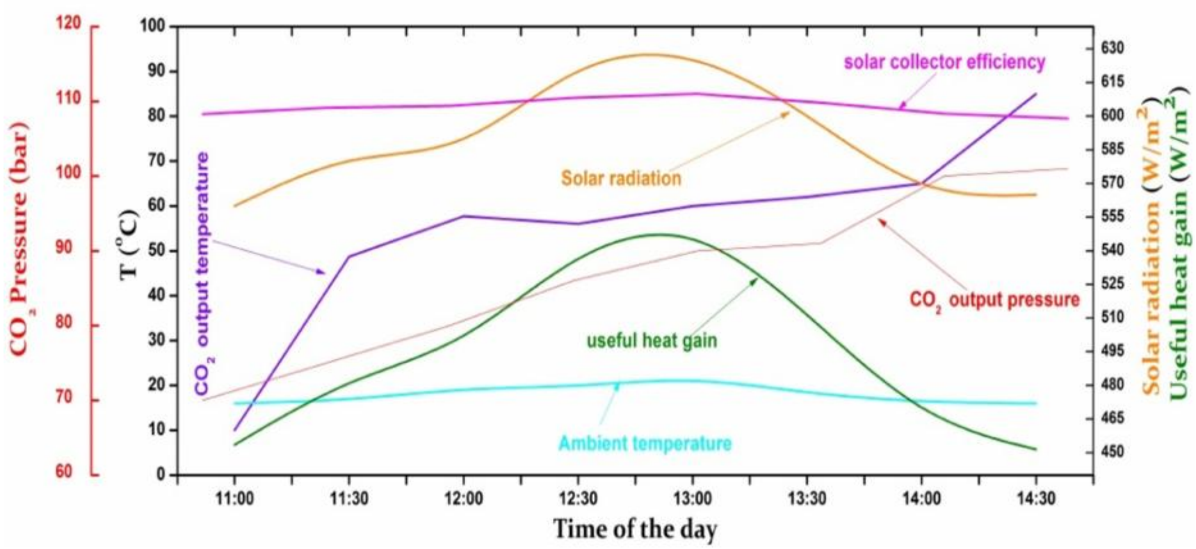

Fig. 6. Experimental results of Solar water heater thermal performance

The result shows that CO_2 refrigerant easily attains 85°C during 16 to 18°C ambient temperatures. The system provides an average 483 W sustained useful heat energy with 82% average collector efficiency. The CO_2 can work well is sub-zero temperature regions and for domestic water heating application.

8. Conclusion

In present study, we employed parametric quantification technique to investigate a set of environmental benign refrigerants (natural and synthetic) for thermosyphon driven solar water heaters. CO_2 showed higher quality factor for the range of -20 to 70 °C whilst R-1234yf emerged superior for commercial water heating application. When CO2 was tested on glass evacuated tube solar collector, it demonstrated excellent heat transfer properties with net heat gain of 483 W and collector efficiency was found to 82% in mild sun shine.

Nomenclature

ρ Density (kg/m3)

Q_u Useful heat gain (W)

ρd Density difference

F' collector efficiency factor

k Thermal conductivity(mW/m-k)

A_c surface area of absorber tube (m^2)

T_{NBP} normal boiling point temperature (°C)

I_T Solar insolation (W/m^2)

T_C Critical temperature (°C)

α absorptivity

Q rate of heat transfer (Watt)

U_L over-all loss coefficient (W/m^2K)

ν^2 Kinematicvelocity (m^2/s)

T_f mean temperature of CO_2(°C)

L3 characteristic length of geometry (meter)

T_a ambient temperature (°C)

g gravitational acceleration (m/s^2)

¦ Efficiency of glass evacuated tube collector

hx Local heat transfer coefficient

Subscripts

Ra Rayleigh number

a actual parametric value

β Coefficient of volume expansion (1/K)

m maximum or highest value

N_u Nusselet number

o optimal/desirable parametric value

G_r Grashof number

P_r Prandtl number

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