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Highlights

- Design and Construction
- Insulator Mixers and MWCNTS
- Machine for the Productivity
- Design & Analysis of Multipurpose

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Design and Analysis of Multipurpose Machine for the Productivity of Sheet Metal Process

By Alie Wube Dametew
Wollo University KombolchaInstitute of Technology

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Keywords: multi-purpose, design and analysis, sheet metal forming, productivity, economic analysis.

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Keywords: multi-purpose, design and analysis, sheet metal forming, productivity, economic analysis.

I. Introduction and Background

Manufacturing is the process of converting raw material into semi or finished goods. There are many manufacturing process are found in production process. Depending on the manufacturing process industries use numerous machines to convert raw materials in to products. Since to increase the productivity and profit of the industries, manufacturing machines/equipment should be properly designed and manufactured. However the methodology of rapid machine design attempts to shorten design-to-manufacture time of production equipment by using advanced engineering tools such as Computer Aided Design systems (CAD), mathematical and Finite Element Analysis (FEA) during the conceptual design phase. Since identifying and to apply the best design concepts, overall development time can be shortened. However, in this paper the new approach to conceptual design can be applied at any phase during the concept generation, whether it is the design as a whole or a component in particular. As components are already part of the machine assembly, changes in their design are automatically updated in the whole skeleton. However, currently, number of sheet metal machines have been designed and fabricate to manufacture different sheet metal products. Main while most machines are performs single operations [P. S. Thakare 20012] and limited production performances. Those machines have their own disadvantages, till sheet metal machines have limited functions which could cause the productivity, the efficiency, versatility and competitiveness the sectors. however the main challenges that are seen on single purpose machines were higher machine cost, reduce sheet metal process productivity, poor production time and non-versatile. As the result multipurpose sheet metal machine need to be designed and fabricated in order to reduce machine cost and increase sheet metal process productivity and improve competitiveness of the sectors.

Since, the main objective of the study is to design analysis and fabricate multipurpose sheet metal machine so as to improve the productivity and competitiveness of sheet metal manufacturing sectors. The design and analysis process were includes select proper materials for production of multipurpose sheet metal machine, mathematical design, analysis and fabrication of multipurpose sheet metal machine, and productivity and economic analysis were considered.

a) Research Methodology

The study was conduct through literature review of research articles, books, magazines, manuals and electronic sources which are discuss related to design, manufacturing and mechanics of sheet metal production process. The analysis was done with mathematical methods and the analysis consider attempts to analyzed, investigate optimum capacity multipurpose sheet metal machine, so as to improve the productivity and comparativeness of sheet metal production sectors. The study attempts to analyses the deflection, material types, the effects of stresses, wear-resistance, ultimate strength of the machine, thermal

Author: School of Mechanical Engineering Wollo University Kombolcha Institute of Technology. e-mail: wubealie@gmail.com

effect natural frequencies under subjected loads using mathematical analysis were done. Since, the design, manufacturing productivity, economic analysis and evaluation of the project was done using the selected empirical analysis. in addition detail and assembly drawing of the machine is done using Solid work or Catiya Software). Finally the conclusion and recommendation of the study was done.

II. Literature Reviews

a) Introduction

Sheet metal fabrication plays an important role in the metal manufacturing world (Cloutier, 2000). Since, sheet metal process is metal formed into thin and flat pieces. It is one of the fundamental forms used in metalworking, and can be cut and bend into a variety of different shapes. Thicknesses can vary significantly, although extremely thin pieces of sheet metal would be considered to be foil or leaf, and pieces thicker than ¼ inch or a centimeter can be considered plate. [Xiaoyun Liao, 2007]. There are many special purposes machines used in this industry to make products. However, the proper selection of the machines depends upon the type of the work under-taken by the particular industry. Since, there are many examples of sheet metal work, which can be seen in our everyday lives. Although, the metals generally used for sheet metal work include black iron sheet, copper sheet, tin plate, aluminum plate, stainless sheet and brass sheet. on the other hand sheet metal is used and applicable in the production of materials ranging from tools, to hinges, automobiles, airplane wings, medical tables, roof for building etc. In terms of process type sheet metal fabrication ranges from deep drawing, stamping, forming, and hydro forming, to high-energy-rate forming (HERF) to create desired shapes (Cloutier, 2000). However, sheet metal is usually produced in sheet thickness less than 6mm by reducing the thickness of a long work piece by compressive forces applied through a set of rolls. This process is known as rolling [Xiaoyun Liao, 2007]. Due to the versatility, sheet metal process is a metal forming process which is spread throughout the world [Michael Lindgren 2009]. Since sheet metal forming is one of the most important semi finished and finished products used in the steel industry, and sheet metal forming technology is therefore an important engineering discipline within the area of mechanical engineering. Consequently, currently, sheet metal forming products have numerous applications, for example in manufacturing industries, buildings, airplanes and the automotive sector as well as infrastructure and domestic appliances, [Anas Muzamil 2009]. Since sheet metal process is a highly productive process and its use increases every year, [P. Groche 2006]. Although, the main feature of sheet metal is its ability to be formed and shaped by a variety of processes. Thus, each process does something different to the metal giving it a different shape or size [Xiaoyun Liao,2007]. The following are the common type of sheet metal process that performed using different type of sheet metal machines.

b) Shape Rolling

Shape rolling of sheet metal is the bending continually of the piece along a linear axis. This causes alteration of the original form of the sheet as it passes through a pathway of series of rollers. Such work tool as shape rolling machine is found to be very useful in manufacturing processes for used parts in various industries like inner and outer panels and stiffeners in automotive and agricultural industries, small metal workshops to roll round and conical profiles for stoves, cylinders (flue pipe, water pipes), basic machine elements with curved surfaces, buckets, bins, gear box cover, mud guards, drinkers and feeders for poultry, feed mixers etc. The machine rolls sheet metal up to 1.5 mm thick and 1m wide and rolls complete cylinders down to 75 mm diameter with 55 mm diameter rollers. Since considering high cost of tools and products in sheet rolling processes, detection and controlling factors for producing precise product are important. In most processes, geometry and configuration of rolling components could be obtained from the geometry of product at the end of loading. Therefore elastic recovery (known as spring back) formed part of the unloading process, and it is the most important factor in deviation of final products from desired geometry.

Also spring back is influenced by a combination of various process parameters such as tool shape and dimension, contact friction condition, material properties, thickness, were investigated and considered [Alie Wube Dametew & Tafesse Gebresenbet, 2016]. Since in this study the design and evaluation of a shape rolling machine with simple crank mechanism, higher roller diameter were analysis.

Cutting/Shearing: Sheet metal can be done in various ways from hand tools called tin snips up to very large powered shears. With the advances in technology, sheet metal cutting has turned to computers for precise cutting and shearing.

Punching Process: Punching is the process of using a machine to press a shape through a sheet of metal and into a die cutter to create that shape in the metal. These machines are single purpose type use, manual type, hydraulic, pneumatic, or electrical power to press the shape with enough force to cut the metal. [K. Abdel-Malek,2008]. Since punching can be better understood as pressing the material against a die with a huge force, this force pushes the material into the die shape and sheers of excess material. [K. Abdel-Malek, 2008]. Improved and effective punching machine is designed in these projects.

Bending Process: In engineering mechanics, bending (also known as flexure) characterizes the behavior of a slender structural element subjected to a lateral load. Since, a structural element subjected to bending is known as a beam. Although, closet rod sagging under the weight of clothes on clothes hangers is an example of a beam experiencing bending. [K. Abdel-Malek,2008].However, bending produces reactive forces inside a beam as the beam attempts to accommodate the flexural load; in the case of the beam, the material at the top of the beam is being compressed while the material at the bottom is being stretched [Xiaoyun Liao, 2007] .

Type of sheet bending

❖ *Edge bending* is also known as flanging in which one edge of the sheet is bent to 90 while the other end is restrained by the material itself and by the force of blank-holder and pad. The flange length can be easily changed and the bend angle can be controlled by the stroke position of the punch[Alie Wube Dametew & Tafesse Gebresenbet,2016].

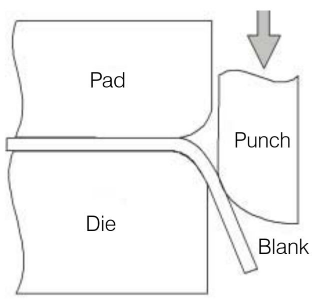

Figure 1: Edge bending

❖ *V-Bending* In V-bending, the clearance between punch and die is constant (equal to the thickness of sheet blank). It is used widely. The thickness of the sheet ranges from approximately 0.5 mm to 25 mm [Alie Wube Dametew & Tafesse Gebresenbet,2016].

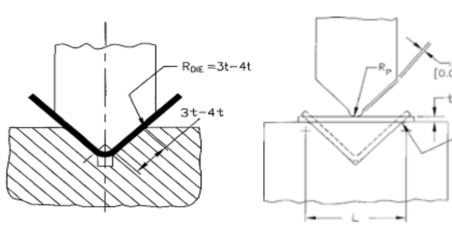

Figure 2: V – Bending

❖ *U - Bending* U-die bending is performed when two parallel bending axes are produced in the same operation. A backing pad is used to force the sheet contacting with the punch bottom. It requires about 30% of the bending force for the pad to press the sheet contacting the punch.

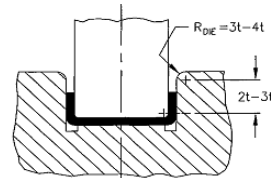

Figure 3: U – Bending

❖ *Coining* is a bending process in which the punch and the work piece bottom on the die and compressive stress is applied to the bending region to increase the amount of plastic deformation. This reduces the amount of spring-back. The inner radius of the work piece should be up to 0.75 of the material thickness. [Ivana Suchy, 2006 Hand book of die design]

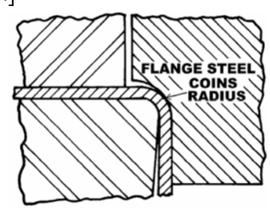

Figure 4: Coining bend

In this study comparative study of single and multipurpose sheet metal machine has been done. From the results, it is cleared that, productivity of three in one sheet metal machine is higher than that of single purpose. The result also shows that total cost for manufacturing metal sheets using three in one machine is 10.43% lower than single purpose machine. Although, the status of sheet Metal forming industry in Ethiopia is not going as it is expected, because of high cost of machine, problem of skill man power, [MOST,2013]. But ,in this time of globalization, the production of sheet forming process use single purpose machines which requires more money, time and labor for atomization of the process. All the machines for using in sheet metal forming process still a single purpose which results low productivity. In order to increase the productivity of sheet metal process, versatility of sheet forming machine is important issue for engineers. However this project deals with the design, analysis and fabrication of five in one sheet metal machine (multipurpose sheet metal machine) is done. Five in one sheet metal machines are perform number of operations within a single machine that is rolling, bending, cutting/ shearing, punching beading and circular shear operations are done within a single Machine.

III. Design and Analysis

a) Rolling Design considerations

i. *General design principles*

Following basic rolling operation on a sheet metal, components can be rolled to give it a definite shape. Bending of parts depends upon material properties at the location of the roll bend. To achieve bending, the work material must be subjected to two major forces; frictional force which causes a no-slip action when metal and roller came in contact and a bending force acting against the forward speed and the torque applied to move the material (Figure).

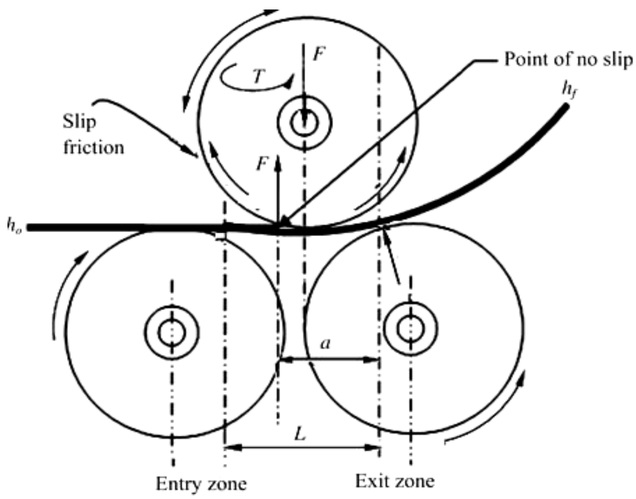

Figure 5: Shape rolling mechanism

where, a = distance from exit zone to the no-slip point (assume $a = L/2$);

F = force applied to rollers; T = torque applied to rollers; L = roll gap; r = radius of rollers; μ = frictional force 0.4 Nm^{-1}; ho, hf = thickness of the sheet before and after time t.

At least two rollers were involved in flat rolling depending on the thickness and properties of material while three or multiple roller system is required in shape rolling. A work material under bending load is subjected to some form of residual stress and deformation as it bends. Since, materials at the outer bend radius undergo tensile plastic deformation while the material at the inner bend radius undergoes compressive plastic deformation. However, at least two rollers were involved in flat rolling depending on the thickness and properties of material while three or multiple roller system is required in shape rolling. A work material under bending load is subjected to some form of residual stress and deformation as it bends. Materials at the outer bend radius undergo tensile plastic deformation while the material at the inner bend radius undergoes compressive plastic deformation.

$$Lb = \theta(r + \kappa T) \tag{1}$$

where, Lb = bend allowance; θ = bend angle; r = bend radius to neutral axis; k = constant for material, for $r<2T$; $k=0.33$; for $r>2T$, $k=0.5$, T=thickness of material.

The strain on the outermost fibers of the bend is evaluated by Equation (2) given by Jack (2003):

$$\varepsilon = \frac{1}{\frac{2r}{T}+1}$$

Maximum bending force is calculated by Equation (3) given below (Jack, 2003):

$\frac{\sigma yield * LT2}{W} = \frac{\sigma UTS * LT2}{W}$, where, P = maximum bending load; k = constant for particular die from 0.3 to 0.7; σ_{yield} = yield stress for material; σ_{UTS} = ultimate tensile stress for the material; L = length of bend (along bend axis); w = distance between reaction supports When the rollers are in contact with the load, there is a frictional force existing, and an applied force, F and a slip between rollers and the load, which is not constant over the entire surface area of contact (Wagoner and Li,2007). An assumption of no reduction in size of material thickness during rolling makes, the thickness uniform i.e. and for small diameter bending, $a = L$. For large diameter bending $L>a$. Thus the maximum force is given by Hugh (2003) as , $\mu 2 * r = hf - ho = Maximum\ draft$ where, μ = frictional force 0.4 Nm-1; ho, hf = thickness of the sheet before and after time t. Analytical solutions of bending process have been presented by several researchers (Dongjuan *et al.,* 2007; Kim *et al.,* 2007; Wagoner and Li, 2007); however, for inverse analysis of spring back in free bending process, a state of plain strain and negligible shear deformation is assumed.

Since, considering the two strain components; the elastic strain (εe) and plastic strain (εp), the total axial strain (εx) can be written as, $\varepsilon x = \varepsilon e + \varepsilon p = \frac{(1-v2)\sigma x}{E} + \varepsilon p$

where, εx = total axial strain;εe = elastic strain; εp = plastic strain; E = Young's modulus, V=Poisson's ratio. Required bending moment (M) can be calculated as, $M = \int_A \sigma xydA$, where, A = area of shaft; σx = axial stress; y = radial arm in mm. Axial strain (εx) can be obtained as

$$\varepsilon_x = \frac{2}{t}\int_{y_c}^{t/2} \varepsilon_p\, dy + \frac{24y}{t^3}\int_{y_c}^{t/2} \varepsilon_p\, ydy + \frac{12M(1-V^2)y}{Ebt^3}$$

Assume axial stress (σx = I):can be obtained as

$$\sigma_x = \frac{2E}{(1-V^2)t}\int_{y_c}^{t/2} \varepsilon_p\, dy + \frac{24y}{t^3}\int_{y_c}^{t/2} \varepsilon_p\, ydy + \frac{12My}{bt^3} - \frac{E\varepsilon_p}{(1-V^2)}$$

Where, b and t are width and thickness of the sheet respectively; εe = elastic strain; ε_p = plastic strain; E = Young's modulus and V= Poisson's ratio. Bend radius after spring back can be written as

$$\rho' = \frac{1}{\frac{1}{\rho} - \frac{12M(1-V^2)}{bt^3E}}$$

Where, ρ and ρ' are bending radius before and after spring back respectively. Knowing the thickness and width of sheet plate and considering material's behavior, analysis of V-bending for various bending angle and radius become possible. The power required to roll the material is given by

$$P = force \times velocity = (Lwy_{ave}) \times (2\pi rn)$$

where, P = power in watts required to roll the sheet and n = speed in rmin-1. The spring back effect in bending is compensated by the following Equation (11) (Jack, 2003).

$$\frac{\sigma_{before}}{\sigma_{after}} = 4\frac{(r_{before}\ \sigma_{yield})^{3}}{ET} - 3\frac{r_{before}\ \sigma_{yield}}{ET} + 1$$

where, $\sigma_{before} = \sigma_{after} = 1$ for flat sheet; € = Bending strain.

b) Results and discussion of roller design

The result table below indicates the maximum bend radius obtainable as an approximate radius of roller for a typical material length of 500-630 mm is 2.5 mm. The largest measured diameter of complete cylinder the machine can handle (roll) is 184 mm using the given diameter of rollers. At reduced roller aperture of 2.5 mm, the material folded over and the radius of cylinder reduced. This result showed a remarkable improvement over Rob's report of 75 mm diameter.

Table 1: Design results of aperture, roll gap and the bend radius

Diameter of the Roller in mm	Material Length in mm	Roller aperture/before	Roller aperture/after bending in mm	Roll gap in mm	Bending radius in mm	Product shape
76	500-630	10mm	8	2	123	Semi-circle
		8mm	5.5	2.5	100	Full circle
		7.0mm	2.5	3	55	Double folded circle

The maximum bend radius when the aperture is closed is approximately the circumferential distance round the roller. The spring back effect on the rolled plate is noticeable as the rolling aperture decreases the bend radius considerably decrease. This effect can be increased by stretching out the material as it rolls or increasing the aperture width. Alternatively, variable roller diameters could be used when handling materials of varying length. Other dimensions can be obtained by varying the width of aperture until the bend radius approaches a straight line depending on the length of work material. The maximum width of material that can be handled with significant bend radius is,123 mm and the length is 500-630 mm.

Besides, the maximum width (machine capacity) of strip of material the machine could handle is 1,050 mm for tinplates and mild steel with thickness not exceeding 2 mm. Although, when the machine was used to roll mild steel plate of 2.5 mm thickness, the bearing blocks showed evidence of possible failure. The wooden bearing blocks could not support the bending stress exerted by the materials with higher thickness than 2 mm and as a result the machine usage can only be restricted to light gauge metal work and thus find use in tinsmith and welding workshops. Slip friction in rolling was eliminated between the rollers and the load by keeping the contact surfaces smooth and free of lubricants and dirt. Since, the average number of operators required to operate the rollers at a giver operation is shown in Table below with their average weights. Material thicker beyond 2 mm sheet thickness requires two operators while 3 mm thickness material could not be conveniently roll due to the required bending force and the strength of the material used in construction. The bearing capacity of wood cannot support such material thickness. The analysis result shows that, the average percentage acceptance of the machine by the artisan is 70.59% (n=24) indicating that the technology is acceptable. Fifteen percent of the welders and 40% of tinsmiths were well acquainted with the functions of the machine. A total of 73.53% (n=26) of the respondents are not acquainted (technology awareness) with the shape rolling technology while 26.47% (n=9) have a fair knowledge of the use of such machine in metal rolling.

Table 2: Table of performance tests Result

Material Type	Material Thickness in mm	No, operations	AV.Wet operator kg	Max width of work in mm	Length of the material
Aluminum sheet	3	1	65	1050	1200.00
	1.5	1	65	1050	1200.00
Galvanized sheet	1	1	71	525	800.00
	2	2	69.8	525	800.00
Milled steel sheet metal	1	1	65	1050	1200.00
	2	2	72	500	600.00
	3	2	72	300	400.00

c) Design of pipe Bending Machine

Companies considering the purchase of tube bending equipment are faced with an extensive set of alternatives. It is important that a potential purchaser research his needs and the various equipment available. The manufacturer who gives special attention to details will gain a significant edge over his competition. In this discussion we will investigate some of the more popular options available for tube bending, their benefits, limitations, cost and applications. Although, the design of pipe bending machine has undergone many changes, development and improvements over a period of time. Pipe bending requires mechanical force which acts on the pipe either directly or indirectly. This was done manually with the operator providing the effort required for bending the pipe. The major setback was the energy, time and effort expended in accomplishing the task. This means that the quality of bend would depend on the strength and skill of the operator. Though relatively cheaper, manual pipe bending falls short of dimensional accuracy and uniformity. Many versions of pipe bending machine have been developed aimed at eliminating human effort (www.paramount-roll.com). In one arrangement, the mechanical force required for bending is provided by a hydraulic ram powered by combustible fuel in an internal combustion engine, or by electricity. In this case, the hydraulic pump which pumps hydraulic into the ram is powered by an electric motor. By early 80's, the development of mechanized pipe bending machine came into existence In general, the bending process uses mechanical force to push the pipe against a die: this way, the pipe is forced to get conformed to the shape of the die. In many cases, the end of the pipe is rolled and rotated around the die, while the pipe itself is firmly held in place (Fig.)

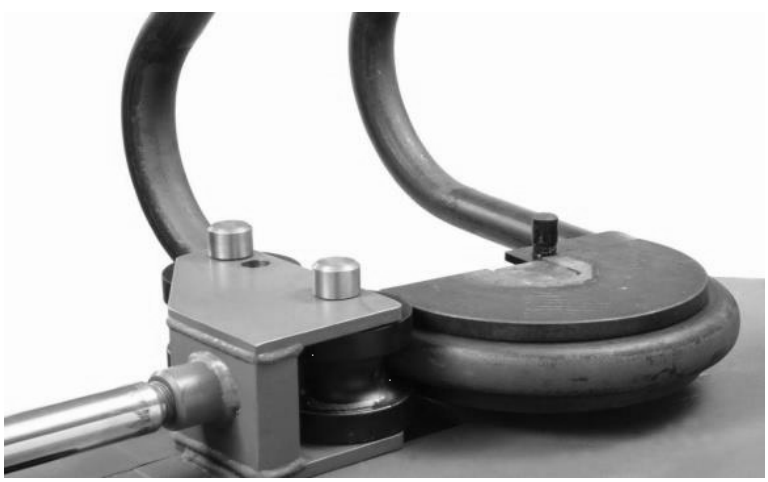

Figure 6: Pipe Bending Process

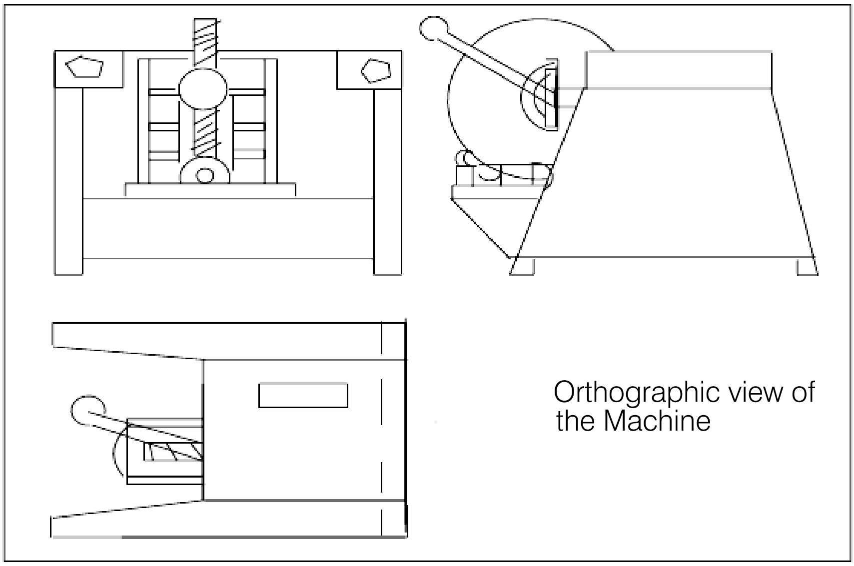

Figure 7: Orthographic View of the Machine

i. *Effort Required to Bend the Pipe*

The machine is considered as a lever with effort arm inclined, the load, W is provided by the pipe resistance to bend. The effort arm, x has both horizontal and vertical components, with the vertical component representing the active force. Since horizontal component, Fh = F sin θ and Vertical component, Fv = F cos θ Taking moment at the support reaction, R; , Fx cos θ = Wy; F = Wy / x cos θ; where F = Effort required to bend the pipe, Let θ = 30o and x = 5y (i.e. depending on the length of the effort arm), F = 23.3 x 103. y / 5y cos 30 = 5.5 kN; where W = average bending force of pipe, 23.3 x 103 N

ii. *Power Requirement*

A gradual application of effort will bend the pipe quite smoothly. This means that very small velocity will be required. An available motor capacity standard is therefore selected and reduced to appropriate speed output.

Choosing a motor of 1.5 kW; Power (P) = Force (F) x Velocity (V) ; Thus, V = P / F = 1500 / 5500 = 0.273 m/s

d) Speed Reduction (Spur Gear Design)

i. *Minimum number of teeth on the pinion*

Tp = 2Aw / G√ 1 + 1/G (1/G + 2) sin2θ – 1 (Shigley and Mischke, 1989), Where G = Gear ration / Velocity ratio; and θ = pressure angle, 20 , Aw = Fraction by which the standard addendum is multiplied, 1m for θ =20

Tp = 2 x 1 / 2 √ 1 + ½ (1/2 + 2) sin220 -1 = 14 2, Thus, we choose Tp = 18 from standard table (Shigley and Mischke, 1989), Number of teeth on the gear, Tg = 2Tp = 2 x 18 = 36

But centre distance between the gears, L = Dg/2 + Dp/2, Where Dg = Diameter of gear, and Dp = Diameter of pinion Dg / Dp = 2; Dg = 2Dp; L = 2Dp / 2 + Dp / 2 = 3/2 (Dp) = 1.5 Dp 64 = 1.5Dp; Dp = 43, Dp = m Tp ; where m is the module m = Dp / Tp = 43/18 = 2.4; Use standard value, m = 2.5

Pitch Circle Diameter of gear, Dg = 2Dp = 2 x 43 = 86 mm

Face Width of the Pinion and the Gear: _Pitch line velocity, V =πDpNp / 60 = π x 0.043 x 1410 / 60 = 3.17 m/s

For medium load shock condition and between 8∼10 hours of service per day (Khurmi and Gupta, 2004); Service Factor, Cs = 1.54 and 2.369 for non-enclosed gears.

Tangential Tooth Load, WT = Cs (P/V) = 2.369 x 1500/3.20 = 1110.5N, Velocity Factor, Cv = 4.5 / 4.5 + V = 4.5 / 4.5 + 3.20 = 0.584 , Since the pinion and the gear are of same material, the pinion is weaker. For 20o involutes teeth;

Lewis Form Factor, Yp = 0.154 – (0.912 / Tp) = 0.154 – (0.912 / 18) = 0.1033 ,Thus, design tangential tooth load; WT = δWp x Cv x b x π x m x Yp, Where δWp is the safe stress of the pinion, 140 MPa and b is the face width of both pinion and gear. , WT = 140 x 0.584 x b xπ x 2.5 x 0.1033; b = 17mm, But minimum face width is taken as (9.54 ∼ 12.5)m ; , Thus, let minimum face width, b = 9.54 x 2.5 = 24 mm

Power Transmitted: P = WT x V = 1110.50 x 3.2 = 355kW

e) Check for Static and Dynamic Loading

Flexible endurance limit for steel, δs = 252 (Khurmi and Gupta, 2004), Static load or endurance strength, Ws = δs x b x π x m x y ,Ws = 252 x 23.8 x π x 2.5 x 0.1033 = 4865.9N , Power that can be transmitted due to static loading is; Ps = 4865.9 x 2.9 = 14.3 kW, Since Ps (14.3 kW) is greater than P (1.5 kW), the design is safe from the standpoint of static loading. Also Dynamic Load, WD = WT + [2/V (bc + WT) / 2/V √bc + WT.But from table (Khurmi and Gupta, 2004), C = 228, and tooth error, e = 0.02 , WD = 1110.5 + [2/3.2 (23.8 x 228 + 1110.5) / (2/3.2√23.8 x 228 + 1110.5) = 3918.29N, Power that can be transmitted from this dynamic load, PD = WD x V = 3918.29 x 3.2 = 12.5 kW , Since PD (12.5kW) is greater than P (1.5kW), the design is safe from the standpoint of dynamic loading.

✓ *Design of Pinion Shaft of the pipe bender*

Load acting between the tooth surface; WN = WT / cos θ = 1110.5 / cos 20 = 1181.8N, Weight of pinion, Wp = 0.00118 x Tp x bm2 = 0.00118 x 18 x 23.8 x 2.52 = 3.16 N.

f) Resultant load acting on the pinion during bending

WR =√WN2 + WP2 + 2WNWPCos θ = √1181.82 + 3.162 + 2(1181.8 x 3.16 x Cos 20) = 1183.29 N

Bending Moment due to this resultant load;
MB = WR x Dp/2 = 1183.29 x 43/2 = 25440.74 N-mm

✓ *Twisting Moment on pinion;*

MT = WT x Dp /2 = 1110.5 x 43/2 = 23875.8N-mm
Equivalent Moment, ME =√MB2 + MT2 = √25440.742 + 23875.82 = 34889.6 N-mm
But equivalent twisting moment is given by;, TE = (π / 16) x 40 x Dp3 ; Dp3 = 34889.6 x 16 /π x 40 = 4440.5 ; DP = 16.5 mm , This shows that with Dp = 43 mm, the design is quite safe.
Diameter of pinion hub = 1.8 Dp = 1.8 x 43 = 77 mm
Length of hub = 1.25 Dp = 1.25 x 43 = 54 mm , Minimum web thickness = 1.8m = 1.8 x 2.5 = 4.5 mm (use web thickness = 10 mm).

✓ *Design of Gear Shaft*

Normal load acting on the gear, WN = 1181.8N
Weight of gear, Wg = 0.00118 Tgbm2 = 0.00118 x 36 x 23.75 x 2 52 = 6.31 N
Resultant load acting on the gear; WR =√WN2 + Wg2 + 2WNWg Cos θ
WR = √1181.82 + 6.312 + 2(1181.8 x 6.31 x Cos 20 = 1189.73N
Bending moment due to resultant load, MB = 1189.73 x 86/2 = 51158.39 N-mm
Twisting moment, MT = 1110.5 x 86/2 = 47751.5 N-mm
Equivalent moment, ME = √MB2 + MT2 = √ 51158.392 + 47751.52 = 69981.3 N-mm
But equivalent twisting moment, TE = (π/16) x 40 x Dg3 = (π/16) x 40 x Dg3
Dg3 = 69981.3 x 16 / π x 40 = 8906.7; Dg (minimum value) = 21mm
This shows that with Dg = 86 mm, the design is quite safe.
Diameter of gear hub = 1.8 Dg = 1.8 x 86 = 154.8 mm
Length of gear hub = 1.25 Dg = 1.25 x 86 = 107.5 mm
Minimum web thickness = 1.8m = 1.8 x 2.5 = 4.5 mm (use web thickness = 12 mm).

✓ Design of Worm Gear

The output of the gear is transmitted to the worm (Fig.3.2), such that Ng = Nw = 705 rev/min

Motor torque x speed of motor = Torque on gear shaft x speed of gear

10.16 x 147.65 = Ts x 73.83; Ts = 10.16 x 147.65 / 73.83 = 20.37 N-mm

Hence, transmitted power, P = TsV = Ts x 2 xπ x N/60 = 20.37 x 2 x π x 705 / 60 = 1.5 kW.

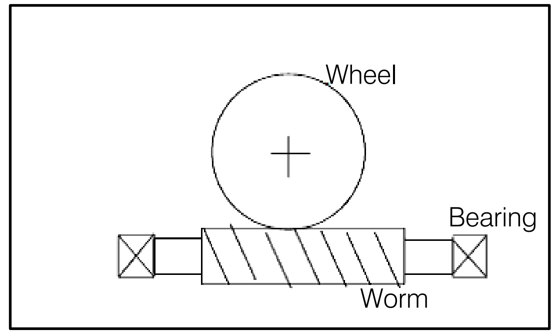

Figure 8: Worm Gear

- ✓ The minimum value of X/LN will correspond to;
- ✓ Cot³ Φ = V.R = 20 Where; X = Lead = 200 mm, LN = Normal load, and V.R. = Velocity ratio
- ✓ Cot Φ = 2.71; Φ = 20.22o
- ✓ X / LN = ½ (1/sin Φ + V.R/cos Φ) = ½ [(1/sin 20.22) + (20/cos 20.22)] = 3.85
- ✓ LN = 200 / 3.85 = 51.92 N
- ✓ Axial load, LA = LN / cos Φ = 51.92 / cos 20.22 = 55.33 N
- ✓ For a V.R. of 20, the number of starts or threads on the worm, n = Tw = 2 (Allens et al, 1980)
- ✓ Thus, axial pitch of the thread on the worm;
- ✓ Pa = LA/2 = 55.33 / 2 = 27.67
- ✓ Module, m = Pa/π = 27.67/π = 8.8 (Take standard module = 8).
- ✓ Thus, axial pitch of the threads on the worm, PA = πm = π x 8 = 25.14
- ✓ Axial lead of the threads on the worm,
- ✓ La = Pa n = 25.14 x 2 = 50.28 mm
- ✓ Normal lead of the threads on the worm, Ln = La cos Φ = 50.28 x cos 20.22 = 47.2 mm.
- ✓ But centre distance = LN/2π (1/sin Φ + 1/cos Φ) = 47.2/2π (24.21) = 181.73 mm
- ✓ Let DW = Pitch circle diameter of the worm; then
- ✓ Tan Φ = L / πDW
- ✓ DW = La /π tan Φ = 50.28 /π tan 20.22 = 43.44 mm
- ✓ Since V.R. is 20 and the worm has double threads; Number of teeth on the worm gear Tg = 20 x 2 =40
- ✓ Face length of the worm (i.e. length of the threaded portion);
- ✓ LW = PA (4.5 + 0.02TW) = 25.14 (4.5 + 0.02 x 2) = 114.09 mm (this is normally increased by 25 ~ 30 for the feed mark) (Khurmi and Gupta, 2004);
- ✓ Thus LW = 144 mm

- ✓ Depth of tooth, h = 0.686PA = 0.686 x 25.14 = 17.24 mm
- ✓ Addendum, a = 0.313PA = 0.313 x 25.14 = 7.86 mm
- ✓ Outside diameter of worm,
- ✓ DOW = DW + 2a = 43.44 + 2 x 7.86 = 59.42 mm

g) Circle pitch diameter of worm gear

Dg = m Tg = 8 x 40 = 320 mm

Outside diameter of worm gear,

Dog = Dg + 1.0135PA = 320 + 1.0135 x 25.14 = 345.5 mm

Face width,

b = 2.38 PA + 6.5 mm = 2.38 x 25.14 + 6.5 = 66.31 mm

✓ *Design of Worm Shaft*

Torque acting on the worm gear shaft,

Tg = P x 60 / 2πNg

Considering 30% overload; Tg = 1.3 x 1500 x 60 / 2 xπ x 35.25 = 528.26 N-m

Torque acting on the worm shaft,

TW = Tg / V.R. x η where η = efficiency of worm gear

But η = Tan Φ / Tan (Φ + α) where α = angle of friction

Rubbing velocity, Vr =πDWNW / cos Φ = π x 0.0434 x 705 / cos 20.22 = 102.53m/mm.

Coefficient of friction, μ = 0.275 / Vr 0.25 = 0.275 / (102.53) 0.25 = 0.086

Thus, angle of friction, α = tan -1 μ = tan -1 0.086 = 4.92

Efficiency of worm gear, η = tan 20.22 / tan (20.22 + 4.92) = 0.78%

Thus, torque acting on the worm shaft, TW = 528.26 x 103 / 20 x 0.78 = 33.86 x 103 N-m

Tangential load on the worm, WT = Axial load acting on the worm gear

WT = 2 TW / DW = 2 x 33.86 x 103 / 43.44 = 1559.06 N

❖ *Axial load acting on the worm,*

WA = Tangential load on the worm gear

WA = 2Tg / Dg = 2 x 528.26 / 320 = 3301.63 N

Radial or separating force on the worm gear,

WR = WA tan Φ = 3301.65 x tan 20.22 = 1216.07 N

If distance between worm shaft bearing and worm gear, X = 400mm,

Then, bending moment due to WR in the vertical plane = WRX/4 = 1216.07 x 40 / 4 = 12160.7N-m

Bending moment due to axial force in the vertical plane = 3301.63 x 320/4 = 264130.4 N-m

Total Bending Moment in the vertical plane, MV = 12160.7 + 264130.4 = 276291.1 N-m.

Bending Moment in the horizontal plane, MH = WT x 40/4 = 1559.06 x 40/4 = 15590.6 N-m.

h) Resultant Bending Moment acting on the worm shaft, during bending process are

- ✓ MR = $\sqrt{}$MH^2 + MV^2
- ✓ MR = $\sqrt{}$276291.12 + 15590.62 = 276730.63 N-m
- ✓ Equivalent twisting moment on the worm shaft,
- ✓ MET = $\sqrt{}$ TW2 + MR2
- ✓ MET = $\sqrt{}$ (33.86 x 103)2 + (276730.63)2 = 278794.44 N-m
- ✓ But MET = π/16 x τ x DW2
- ✓ where DW = diameter of worm shaft; τ = allowable shear stress, 40 N/mm2
- ✓ DW = 32.86 mm (Use DW = 36mm).

Since, from the analysis we observed that, the pipe bending can be achieved in both ways (either in upward or downward direction) up to 4.25 mm pipe thickness could be bent manually depending on the operator's physical strength. During the bending process, the slight deviation from intended angle of bend was occurred due to spring back action of the pipe, which obviously reduces with decrease in angle of bend. However, wrinkles and bulging of the pipe during bending were noticed in pipes of lesser thickness. in this machine with a semi-circular bell, angles between 80^{o} to 180^{o} were obtainable, below which the bent pipe was observed to follow the bell's contour to give a U or C shape. Since, to obtain a lower bend angle, a different bell specially made for 45^{o} is used. For pipes of lesser thickness, a mandrel should be introduced in order to prevent collapse. The provision for manual bending makes it possible to use the machine in rural areas were there may be no power supply.

i) Bending Design

Bending is a metal forming process in which a force is applied to a piece of sheet metal causing bending of it to an angle and forming the desired shape (Manar, 2013). While, the operation is typically performed on a machine called a press brake which can be manually or automatically operated. Though, to bend sheet metal, a bottom tool (die) is mounted on a lower, stationary beam (bed) and a top tool (punch) is mounted on a moving upper beam (ram). Since the design and analysis of this study is considered this points properly, to achieve optimum bending products. However the bending design consists and considers the f parameters given in the table below.

Table 3: Initial Conditions for Design

Maximum bending length	1800 mm
Maximum bending thickness	2mm
Tensile strength of sheet metal Mild steel	248 MPa
Clearance between folding beam and clamping beam	2 mm
Maximum folding angle	1050
Frame material	Structural Steel
Folding beam and clamping beam material	Machine Steel

j) Maximum Folding Force

The force required to perform folding depends on the strength, thickness, and length of the sheet metal (Groover, 2010). However, the maximum folding force can be estimated by means of the following equation: $F = \frac{Kbf\,(TS)wt2}{D}$,where, *Kbf* = 0.33,*TS* = 248 MPa, *w* = 1 800 mm, *t* = 2 mm; and *D* = 2 mm. Therefore; $F = 0.33 * 248 * 1800 * 2 * \frac{2}{2} = 294.6KN$

k) Clamping Beam Design

The clamping beam exerts a force that holds down the sheet metal onto the folding bed. The hold down force when performing folding operation is 50% the required folding force. Since it is applied across two ends of the machine. Therefore the clamping force is given by: Clamping force = 0.5 x folding force, Clamping force = 0.5 x 294.6 kN, Clamping force = 147.3 kN.

The clamping beam is designed such that it is welded onto side plates that are connected to a clamping mechanism. The clamping mechanisms are located on both sides of the clamping beam, but the clamping knob is only located on one end. The adjusting screws on the clamping mechanism of bending machine must resist the clamping force they are exposed to. Since, the load is shared equally on either side of the clamping mechanism, therefore is equal to half the clamping force that is 73.65 kN. Allowable stress levels to 75% of proof strength are to be used in the clamping mechanism bolts. The selected material for the clamping mechanism according to the Society of Automotive Engineers (SAE), is grade 4 with no head marking and proof strength of 65 ksi. Then the allowable stress is also given as σ_a =0.75*proof strength, Since σ_a=0.75*65000psi=<u>48759psi</u>

In addition, the force applied on each side of the clamping mechanism is 73.65 kN = 16.55 klb

Therefore the required tensile area to which the force should act is:

$$At = \frac{Load}{\sigma a} = \frac{16550\,lb}{\frac{48750\,lb}{in2}}$$, Since, A_t+=0.339in^2. Thus, tensile stress area of 0.339 in2 requires a diameter of 7/8 inches, which is equivalent to 22.22 mm. Hence the diameter of the clamping mechanism column should be 22.22 mm with a course thread of 9 threads per inch.

l) Design of the Folding Beam

Figure shows the front view for the folding beam.

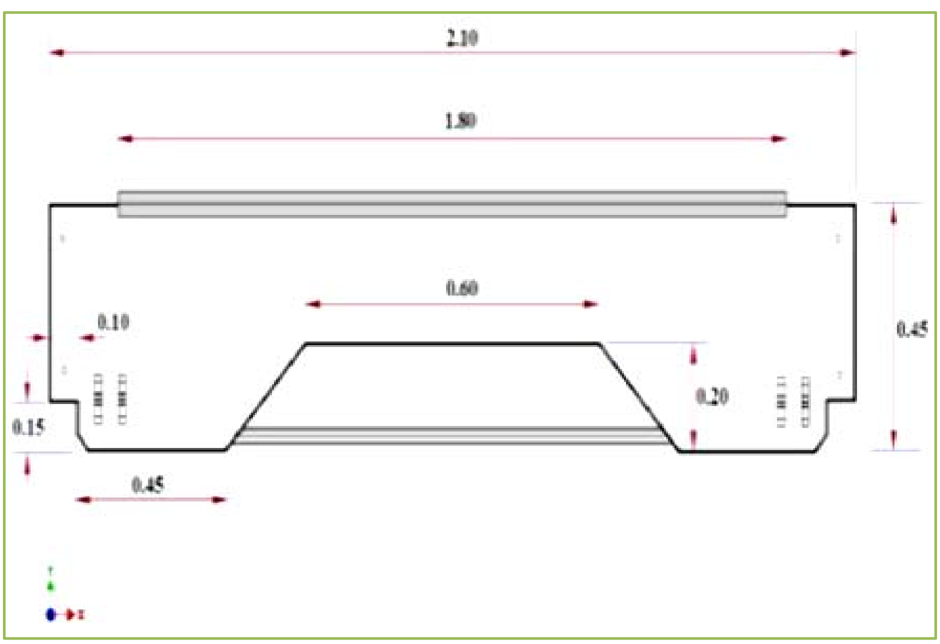

Figure 9: Folding Beam Front View

The weight of the folding beam is calculated using the formula below. Weight of folding beam = surface area x density x thickness Surface area, A, of folding beam: Surface area = 2.1 (0.45), $-0.5(0.2)(1+0.6)-2(0.15)$

Therefore *surface area*=0.755 m^2, Using machine steel of density 77 kN/m^3, if the thickness of the folding beam is therefore, the weight of the folding beam is: *W = Surface area* x *density* x *thickness*

W = 0.755 m2 x 77 kN/m^3 x *t*, ∴*W* =58.135 *t kN*, The beam is supported on its two ends and other force (including its weight) acting on the beam is the maximum required folding force of 294.6 kN acting uniformly across the length of the beam. Figure below also represents the loading induced on the beam during bending process.

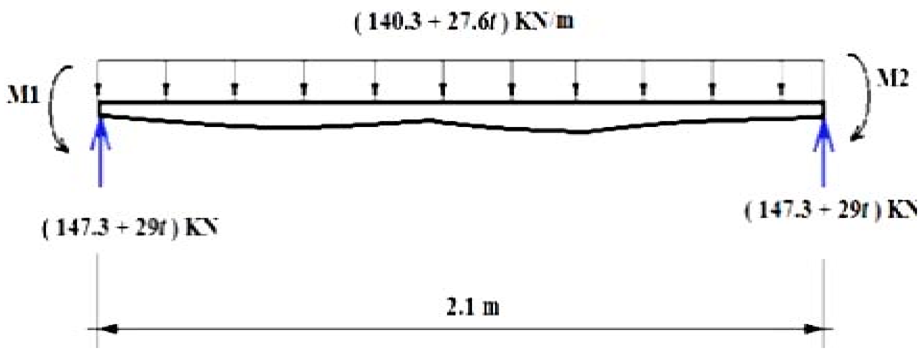

Figure 10: Loading on the Folding Beam

Total force acting on the beam = (294.6 +58.135t) kN.

$$Force\ acting\ per\ unit\ length = \frac{Total\ force\ action\ on\ the beam}{Length\ of\ the\ beam}$$

$$Reaction\ at\ each\ support = \frac{Total\ force\ action\ on\ the beam}{2}$$

$$Reation\ at\ each\ support = \frac{(294.6+58.135t)kN}{2},$$

Therefore reaction at each support =(147.3 +29 t)kN.

Taking moments and resolving forces at determined points along the folding beam and factoring a safety factor of n = 3, and an allowable stress of 350 Mpa, t,is found to have the following value; t = 0.015 or t = -0.015. Therefore the thickness of the folding beam is 15 mm.

IV. Productivity Analysis

Increase in productivity is the key factor for prosperity at all levels. It is the relationship between the result obtained and the factors employed to achieve the result. Productivity is the relationship between outputs to input. It is an indication of an enterprise capability. In case of the defined machine, the output highly depends upon the working skill of the employed persons. If he is having long experience of working over the machine then definitely his rate of making pipes would be higher than the rate of person who is new to machine. As per the definition of productivity, we have a simple relation for it and it is given by

$$Force\ acting\ per\ unit\ length = \frac{(294.6+58.135)kN}{2.1}$$

Thus, the force acting per unit length (140.3 +27.6 t)kN /m Reaction force at the beam supports also calculated as,

$$\text{Productivity} = \frac{\text{Output}}{\text{input}}$$

To directly compare the productivity in terms of capital required, we take the ratio of output in terms pipe Productivity=[(M anf. cost of material RS/ pieces x No. of material smanf, /day)][(Expenditure/material x No. of mant/day]

a) *Productivity of Manually Operated Machine*

Since time required for manufacturing the one pipe is 40 minutes, total number of pipes manufactured in a day is 12. Using above equation, Productivity of Manually operated machine is calculated as

$Productivity = 217 * \frac{12}{58.38*12} = 3.71$------For Manually operated machine

b) *Productivity of Power Operated Machine*

Since time required for manufacturing the one pipe is 10 minutes, total number of pipes manufactured in a day is 48. Thus, Productivity of Power operated machine is calculated as

$productivity = 217 * \frac{48}{26.07*48} = 83.46$-------------

For Power operated machine. Since our multipurpose machine is power operated , then it is productive. in a single operation , the productivity is double for manual operated machine, but our machine multipurpose, within this single machine performed more than four type of sheet forming process. Since the productivity is four times a single power operated machine. thus this multipurpose power operated sheet metal machine is efficient, effective and more extremely product type of machines.

V. Conclusion and Recommendation

a) Conclusion

In this study the design and analysis of multipurpose sheet metal machine is done using mathematical methods. During the analysis and design process different parameters were considered for improving the productivity and performance of sheet metal forming process. As well comparative study of manually operated multipurpose machine and power operated sheet bending machine has been done. the multipurpose machines are very efficient in rolling, bending, grooving and beading of metallic components. The machine is cost effective based on the materials of production and simplicity of the design of component parts. Operational mode meets the level of technical knowhow of the artisans. Also, productivity of both the machines has been calculated. From the results, it is cleared that, productivity of power operated sheet bending machine is higher than manual type. As a result a multipurpose power operated machine is much, much efficient, effective and productive on compared to manual operated machine. Since the design and analysis of this study was meet the objectives properly. Since, sheet metal manufacturing industries improve their financial performance and productivity by using these type of multipurpose machines rather than single purpose machine.

b) Recommendation

This multi type machines have numerous advantages including, optimum machine cost, high production capacity and productive, efficient and effective and contributes the comparativeness of the sectors. Since, sheet metal manufactures could be use such type of machines to improve their performance and productivity. Further recommend that, researchers further study to add the function of such type of machines, hence this used as a base for the researchers.

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Design and Construction of a Portable Charger by using Solar Cap

By Md. Rakib Hasan, Md. Sabbir Hossain & Kazi Pavel Rahman
Khulna University of Engineering & Technology

Abstract- Sun is a source of renewable energy called solar energy. Solar energy is a basic need of living plants and human being on the earth. By the use of solar energy there is no pollution and no waste. There are many fields of using solar energy. It can be used directly in a variety of thermal applications like heating of water or air, charging batteries, drying, distillation, cooking etc.

Bangladesh is an under developing country. It is a country of lot of problems. Energy crisis is one of the important problems. To overcome this problem solar energy may be used as an alternative. It is not possible to solve the giant problem over a night but it can be decreased. Solar energy is one kind of renewable energy. Everyday a lot of power is used for charging purpose like mobile, camera, light etc.

Keywords: solar energy, energy, renewable energy, solar cap, reservoir.

GJRE-A Classification: FOR Code: 290501

DESIGNANDCONSTRUCTIONOFAPORTABLECHARGERBYUSINGSOLARCAP

Strictly as per the compliance and regulations of:

Design and Construction of a Portable Charger by using Solar Cap

Md. Rakib Hasan [α], Md. Sabbir Hossain [σ] & Kazi Pavel Rahman [ρ]

Abstract- Sun is a source of renewable energy called solar energy. Solar energy is a basic need of living plants and human being on the earth. By the use of solar energy there is no pollution and no waste. There are many fields of using solar energy. It can be used directly in a variety of thermal applications like heating of water or air, charging batteries, drying, distillation, cooking etc.

Bangladesh is an under developing country. It is a country of lot of problems. Energy crisis is one of the important problems. To overcome this problem solar energy may be used as an alternative. It is not possible to solve the giant problem over a night but it can be decreased. Solar energy is one kind of renewable energy. Everyday a lot of power is used for charging purpose like mobile, camera, light etc. Those devices can be easily charged by using solar charger. In this project a solar cap is designed and constructed for charging mobile phone, camera etc. which is nothing but a solar panel based charging system. Here a solar panel is placed on a cap. An USB port is attached with the panel. A cable is connected with the solar panel and the device that will be charged. At day time the device can be easily charged by using this solar cap. If the device is fully charged then the extra charge can be stored in a reservoir. So, by using the charger the devices can be charged day and night.

Keywords: solar energy, energy, renewable energy, solar cap, reservoir.

I. Introduction

Energy crisis is one of the basic problem in developing country like Bangladesh. One step to overcome this problem may be the use of solar energy as an alternative. A huge amount solar energy is available in the environment that can be utilized and also could be stored to use any suitable time.

Solar energy, radiant light and heat from the sun, is harnessed using a range of ever-evolving technologies such as solar heating, solar photovoltaic, solar thermal electricity, solar architecture and artificial photosynthesis. Solar technologies are broadly characterized as either passive solar or active solar depending on the way they capture, convert and distribute solar energy. Active solar techniques include the use of photovoltaic panels and solar thermal collectors to harness the energy. Passive solar techniques include orienting a building to the sun, selecting materials with favorable thermal mass or light dispersing properties, and designing spaces that naturally circulated with air.

Author α σ ρ: Department of Mechanical Engineering, khulna University of Engineering & Technology, Khulna, Bangladesh.
e-mail: rakibme2k9@gmail.com

Sun is responsible for most of accessible energy resources. Solar energy can be used both directly and indirectly. It can be used directly in a variety of thermal applications like charging of batteries, heating water or air, drying, distillation, cooking etc. The heated fluids can in turn be used for applications like power generation. A second way in which solar energy can be directly through the photovoltaic effect in which it is converted to electrical energy. Indirectly, the sun causes winds to blow, plants to grow, rain to fall and temperature differences to occur from the surface to the bottom of oceans. Useful energy can be obtained for commercial and non-commercial purposes through all these renewable sources. Solar portable charger is one type of chargers which can be carried any place at any time. In addition, a good portable solar charger should be straightforward and easy to use. In this type of portable charger, solar panel is placed on the cap which is put on the head. When sun strikes on the solar panel photons release from it. Then electron starts flow though the cable which is connected with solar panel. A PCB board is also connected with solar panel. Solder the positive output wire of the voltage regulator to the USB's positive. Similarly, connect the negative output of regulator to the negative of USB. The USB port must be fixed properly to the PCB board. A reservoir is used which store charge and supply charge to the battery when require.

Solar portable charger is very effective for everyday use. It is suitable for use in rural area where electricity is not available or load shading frequently occur. Travelers and advantageous people can also use this type portable charger.

II. Design

a) *Assumptions for Design*

- Solar panel should be capable of supplying 5-6 volts.
- The cap should be easily carried.
- Reservoir should store charge properly and supply it when require.

b) *Design of Components*

Different components needed for this system are designed. When designing above assumptions are taken under consideration.

- Solar panel

There were various types of solar panel. In this construction TYN355-366 type solar panel was used. The capacity of this type of solar panel was 5V (volt) and 5W (watt).It could able to supply 800-1000 mA current which was required for charging a battery (mobile, camera etc.). It was required to place the solar panel perfectly on the head. So that the dimension is selected Length= 16 cm, Width= 12 cm

- Reservoir

The length of the reservoir was 10.3 cm and width is 2.8 cm. Reservoir had charged storing capacity. It was able to supply charge to battery when required.

- Charging system

A PCB board was connected with solar panel. The length of the PCB board was 3 cm and width=2cm.USB port and LED lamp attached with PCB board. Finally a cable was connected between battery cell and USB port. The length of the cable was used according to required.

c) Description of the designed system

There are two types of charging method a) Direct charging b) Charging by use reservoir. In 'direct charging' method, one end of cable is connected with USB port of solar panel and other end of the cable is connected with battery cell. In 'charging by reservoir' method, only different from the previous method is that it uses a reservoir which primarily store charge. The storage charge from the reservoir then supply to the battery cell by cable.

d) Circuit diagram

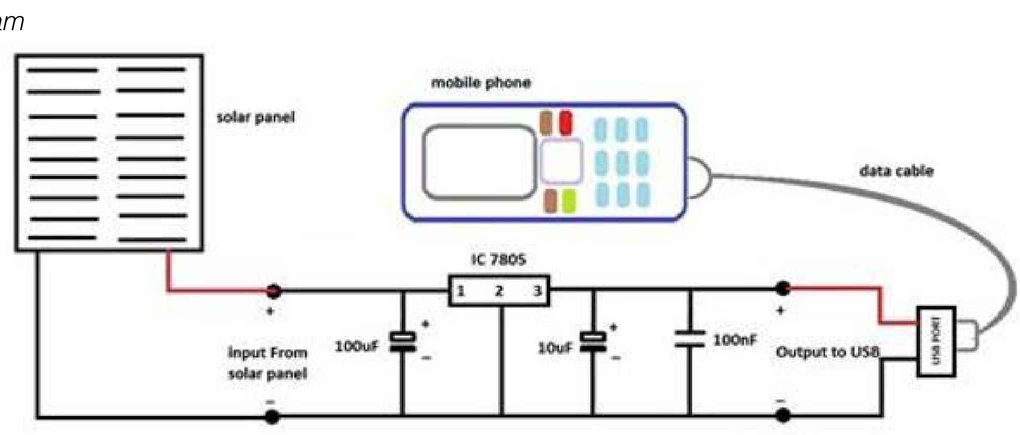

Figure 2.1: Circuit diagram (mobile phone connects with solar panel)

Fig. shows that, a solar panel is connected with IC 7805. This integrated circuit(IC) is divided with 3 section and amount of current flow is 100μF. Now, IC is connected with the output of USB. A data cable is connected with USB port. Finally current flows though the data cable and charges the battery of mobile, camera etc.

III. Construction

Components Required:

a) The required components are

i. Solar Panel
ii. Cap
iii. PCB board
iv. USB port
v. Reservoir
vi. Wire
vii. LED lamp
viii. Battery (mobile, camera, torch etc.)

b) Description of Construction

- ☐ A PCB board is connected with solar panel. Solar panel consists of
- ☐ Semiconductor material silicon (Si).There are two types of silicon i) p-type silicon ii) n-type silicon. This two types of silicon produce pnp junction or npn junction.

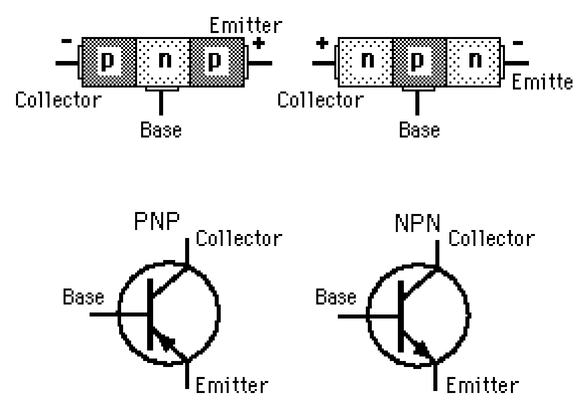

Figure 3.1: PNP junction and NPN junction

- Solder the positive output wire of the voltage regulator to the USB's positive. Similarly, connect the negative output of regulator to the negative of USB. The USB port must be fixed properly to the PCB. Next, connect the solar panel to the input of the voltage regulator (positive of solar panel to positive input of voltage regulator and negative of solar panel to negative input).
- The regulator circuit consists of the following components.

1) IC7805
2) 100uF
3) 10uF
4) 4) 100nF

- USB port and LED lamp is attached with PCB board. When sun light strikes on the panel photon release and electron start to flow.
- A cable is connected with USB port. Once everything is connected, measure the output voltage in open sun light. It should be around 5V. Now, connect batteries of mobile phone or camera and it starts charging.

Figure 3.2: Direct charging of battery of a mobile phone

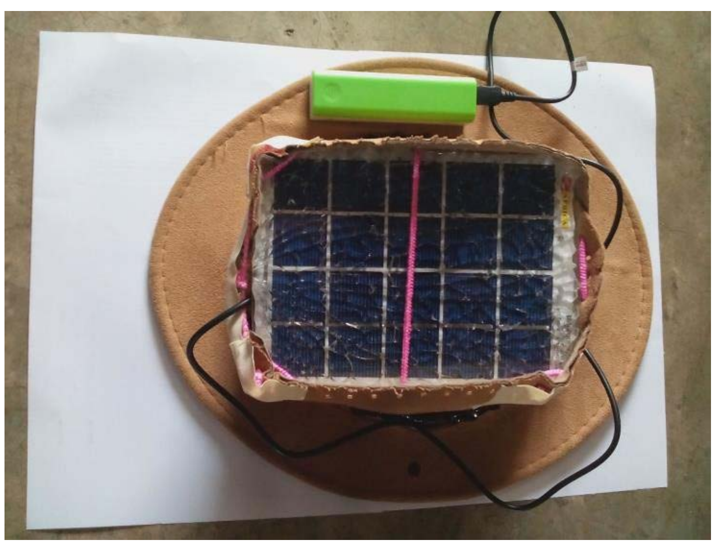

Figure 3.3: Charging the battery of mobile from reservoir Test Procedure

IV. Test Procedure

Batteries were charged both at stationary and moving conditions. In both the cases it was observed to charge the battery successfully. The performance of portable charger depends on solar intensity that was also observed while charging in sunny and cloudy sky. Storing of charge in the reservoir was checked by charging battery at night successfully.

V. Result and Discussion

Test results of battery charger was found satisfactory. It took almost same amount of time to be fully charged from main. Storing of charge and also charging from the reservoir were checked and found satisfactory.

Performance of storing of charge and charging of battery were found satisfactory and both were found satisfactory and both were delayed in cloudy sky was also observed.

VI. Conclusion

A portable solar charger by using a solar cap has been designed and constructed successfully. Battery has been charged directly by the dc voltage produced by a solar panel through a USB port. Performance of the devices has been tested and following results are obtained.

1. Batteries can be charged both stationary and moving condition.
2. Charging time takes almost same amount of time to be fully charged from main.
3. Performance of portable charger depends on solar intensity
4. Charge can be stored in the reservoir.

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Two Different Viewpoints about using Aerosol-Carbon Nanofluid in Corrugated Solar Collectors: Thermal-Hydraulic Performance and Heating Performance

By Soroush Sadripour, Mohammad Adibi & Ghanbar Ali Sheikhzadeh
University of Kashan

Abstract- In this study the effects of corrugated absorber plate and using aerosol-carbon black nanofluid on heat transfer and turbulent flow in solar collectors with double application and air heating collectors, were numerically investigated. The two-dimensional continuity, momentum and energy equation were solved by finite volume and SIMPLE algorithm. In the present investigation all the simulations were done for two different angles of tilt of collector according to horizon, that these angles were the optimum ones for the period of six months setting. As a result the corrugated absorber plate was inspected in the case of triangle, rectangle and sinuous with the wave length of 1mm and wave amplitude of 3 mm in turbulent flow regime and Reynolds number between 2500 to 4000.

Keywords: solar collector, corrugated absorber plate, turbulent flow, performance evaluation criteria, heat performance coefficient, nanofluid.

GJRE-A Classification: FOR Code: 290501p

TWODIFFERENTVIEWPOINTSABOUTUSINGAEROSOLCARBONNANOFLUIDINCORRUGATEDSOLARCOLLECTORSTHERMALHYDRAULICPERFORMANCEANDHEATINGPERFORMANCE

Strictly as per the compliance and regulations of:

Two Different Viewpoints about using Aerosol - Carbon Nanofluid in Corrugated Solar Collectors: Thermal-Hydraulic Performance and Heating Performance

Soroush Sadripour [α], Mohammad Adibi [σ] & Ghanbar Ali Sheikhzadeh [ρ]

Abstract- In this study the effects of corrugated absorber plate and using aerosol-carbon black nanofluid on heat transfer and turbulent flow in solar collectors with double application and air heating collectors, were numerically investigated. The two-dimensional continuity, momentum and energy equation were solved by finite volume and SIMPLE algorithm. In the present investigation all the simulations were done for two different angles of tilt of collector according to horizon, that these angles were the optimum ones for the period of six months setting. As a result the corrugated absorber plate was inspected in the case of triangle, rectangle and sinuous with the wave length of 1mm and wave amplitude of 3 mm in turbulent flow regime and Reynolds number between 2500 to 4000. Choosing the proper geometry was carried out based on the best performance evaluation criteria (PEC), for collectors with dual usage and increasing the air temperature from collector inlet to outlet for air heating collector. The results revealed that using corrugated absorber plate has a considerable influence on flow field and heat transfer. For all times of the year the highest PEC was obtained for corrugated Sinusoidal model, however the highest temperature increase from inlet to outlet was obtained for rectangular corrugated model. Also it was understood that in the case of using air as a base fluid, whether for the case of temperature increment from inlet to outlet or the highest PEC,the optimum Reynolds is 2500. For each of the corrugated absorber plate with sinusoidal and rectangular models, the carbon black nanoparticles were added to air base fluid in volume fractions of 0.1% to1%. The results indicated that in sinusoidal model the nanoparticles volume fractions increase leads to heat performance coefficient increase and the best heat performance conditions were attained in volume fraction of 1% and Reynolds number of 4000 for both six months period. In rectangular corrugated model using nanofluid and Reynolds number increase do not worth and lead to outlet temperature decrease. Therefore for this model using air and Reynolds number of 2500 is recommended.

Keywords: solar collector, corrugated absorber plate, turbulent flow, performance evaluation criteria, heat performance coefficient, nanofluid.

Author α σ ρ: Department of Mechanical Engineering, University of Kashan, Kashan, Iran. e-mail: soroushsadripour@hotmail.com

I. Introduction

The analyses of the international energy organization show that the world energy demand between 2008 and 2035 increases by 35%. According to the limited fossil energy fuels and the side effects of using them on environmental cycle the probe for finding renewable energy in order to deal with this increasing energy demand is necessary. According to international energy agency predictions, more than 13% of this increasing energy demand will be provided by renewable energy [1]. The solar energy is considered as the cleanest, the cheapest and the most accessible energy in the world. The flat plate solar collector comparing with other collector types, has simple design and low costs of construction and in addition to direct solar radiation absorption they can also absorb the emissive radiation [2]. The hot water and air have a wide range of application in industry, agriculture, animal husbandry, and household chores. Therefore it is possible to use a collector that can heat water and air at the same time. The present study concentrates on solar collectors with dual usage and solar hot air collectors. Sofar lots of numerical and empirical studies related to solar collectors have been conducted. The results of these studies demonstrate that the overall performance of collector is related to many factors including the distance between absorber plate and glass coverand pipe diameter [3,4], windvelocity [5], solar radiation [6], collector material [7], flow rate [8], and channel depth [9].

There are numerous ways to enhance the solar collector efficiency. One way is to use the methods for absorbing more solar radiation. This method is done by setting the collector angle of tilt and put the collector in the optimized angle of tilt. Khorasanizadeh and Meschi[2] specified the optimized angle of tilt in the case of monthly, seasonally, six months and annual for solar collector in kashan. They suggested of 9° and 51°angle of tilt for setting in the first six months and the second six months, respectively.

Nomenclature		Greek symbols	
C_k	thermal conductivity ratio	α	thermal diffusivity, (m^2/s)
C_μ	turbulence model constant	α	wavy amplitude, (mm)
C_P	specific heat capacity, (J/kg.K)	β	collector slope
D_h	hydraulic diameter, (m)	Γ	molecular thermal diffusivity
f	friction factor	Γ_t	turbulent thermal diffusivity
G_k	generation rate of TKE	ε	Turbulent dissipation rate, (m^2/s^3)
h	heat transfer coefficient, (W/m^2.K)	η	heat performance coefficient
H	height of the collector channel, (mm)	ϑ	kinematic viscosity, (m^2/s)
$\overline{H}$	average of heat flux in 6 months	λ	wavelength, (cm)
k	thermal conductivity, (W/m.K)	μ	dynamic viscosity, (Ns/m^2)
k	turbulent kinetic energy, (J/kg)	ρ	density, (kg/m^3)
L	collector length, (m)	ϕ	nanoparticles volume fraction
Nu	Nusselt number		
P	pressure, (Pa)	*Subscript*	
Pr	Prandtl number	av	Average
ΔP	pressure drop, (Pa)	bf	base fluid
PEC	performance evaluation criteria	c	Local
Q	heat flux	i, j	components
r	sinusoidal geometry function	in	Inlet
Re	Reynolds number	nf	Nanofluid
t	average of sunny hours in day	np	nanoparticle
T	temperature , (K)	t	Turbulent
u	flow velocity component, (m/s)	w	Wall
u'	fluctuated velocity, (m/s)	0	smooth surface

The second way is to apply some changes in solar collector geometry in order to reach the highest thermal performance. Getting the heat exchanger jagged and grooved on the interior side is one of the methods for breaking the laminar sub layer and creating the local wall turbulence(due to repetitive flow separation and adhesion between successive grooves). This method decreases the thermal resistance and increases the heat transfer considerably.

Some numerical and experimental researches on the flow fluid and heat transfer inside the corrugated channels have been carried out by some researchers. Comini et al. [10] studied numerically the flow and heat transfer characteristics in three-dimensional wavy channels. They found that the Nusselt number as well as friction factor increases with decreasing aspect ratios.

Grant Mills et al [11] conducted a numerical study on heat transfer enhancement and thermal–hydraulic performance in laminar flows through asymmetric wavy wall channels. The results are crucial for designing compact heat exchangers that are capable of having high performance in the laminar regime. Mohamed et al. [12] presented laminar forced convection in the entrance region of a wavy channel. They solved numerically the governing equations using the finite volume method. The effects of Reynolds number, Prandtl number and the amplitude of the corrugation on the flow and thermal fields were introduced by them. It was realized that the shear stresses and Nusselt numbers increase as the Reynolds number increases. Rostami et al [13]investigate doptimization of conjugated heat transfer in wavy walls in micro channels. Numerical results reveal that the Nusselt number in wavy microchannels is more than that for flat walls micro channels. Also unlike flat walls microchannels there is an optimum geometry for wavy walls micro channels, which has the maximum Nusselt number. Duan and Muzychka [14] inspected the influences of axial corrugated surface roughness on fully developed laminar flow in micro-tubes analytically. The Stokes equation was solved to predict friction factor and pressure drop in corrugated rough micro-tubes for continuum flow and slip flow. It was observed that there was a significant increase in pressure drop due to roughness.

The third method is to increase the heat transfer between fluid and solar absorbing plate. One common and suggested way is to add the nanoparticles to the base fluid used in collector.

Khoshvaght-Aliabadi [15] analyzed heat transfer and flow characteristics of the sinusoidal-corrugated channels with Al_2O_3-water nanofluid. The effects of different geometrical parameters were calculated at the nanoparticle volume fraction below 4%. The channel height and amplitude indicate the highest influences on Nusselt number and friction factor values. Mohammad et al [16] numerically analyzed the heat transfer and water flow characteristics in a wavy micro channel heat sink (MCHS) and also with different wave amplitude by using the finite volume method. They found out that the heat transfer coefficient, wall shear stress, pressure drop, and friction factor increase by increasing the wave amplitude through the channel. Heidary and Kermani [17] numerically studied the effect of forced convective heat transfer of Cu-water nanofluid on heat transfer field and flow field in channels with sinusoidal walls. They noticed that by using the nanofluid and horizontalwavy walls at the same time, the heat transfer increases by 50%. Jena and Mahapatra [18] in their numerical modeling investigated the radiative and natural convective heat transfer of aerosol-carbon black nanofluid, in the presence of magnetic field for a two-dimensional chamber. Their results demonstrated that by increasing the volume fraction of carbon black nanoparticles in the base fluid of air, the overall heat transfer inside the chamber will enhance.

Different nanoparticles with various base fluids, higher volume fractions and smaller nanoparticles are found to increase heat transfer [19]. Although in this study the new nanoparticle called carbon black has been used with volume fraction between 0 to 1%. It is inferred from the previous investigations that no numerical studies has been carried out on aerosol-carbon black nanofluid with turbulent flow inside the solar collector with corrugated walls.

In lots of aforementioned numerical investigations the effects of corrugated plates on enhancing heat transfer in channels have been done and the comparison between smooth and sinusoidal corrugated channels has been made. However for solar collectors, the accurate comparison between linear corrugated models(rectangular and triangular) and nonlinear models (sinusoidal) has not been made in the case of Nusselt number, pressure drop, friction factor, PEC, and the difference between inlet and outlet temperature, and mainly the considered collectors were solar flat plate collectors.

In current investigation the precise comparison for scrutinizing the thermal-hydraulic characteristics of the turbulent air flow and aerosol-carbon black nanofluid inside collectors with rectangular, triangular,and sinusoidal corrugated absorbing plate has been made by numerical procedure. Among absorbing plates with different corrugations, the optimized configurations have been chosen based on the higher PEC and difference between inlet and outlet temperature. Then the effects of aerosol-carbon black nanofluid on the thermal-hydraulic characteristics and temperature increase from inlet to outlet are analyzed for these optimized configurations.

II. Numerical Modeling

a) *Physical model*

The schematic diagram of the two-dimensional collector with entrance height of 2H=20mm is shown in Fig. 1.The problem geometry includes two-dimensional corrugated walls with 18 waves along the length of the test-section. The heat transfer and flow field are studied for sinusoidal, triangular and rectangular corrugations. The wave length of the walls in all three models is $\lambda = 1$ cm and the wave amplitude is $\alpha = 3$ mm. The collector length equals to $L = 2$ m. For the left section of the channel the velocity inlet boundary condition in Reynolds number between 2500 and 4000 is considered and for the outlet section of the channel pressure outlet boundary condition is assumed. The absorber plate is in the constant and uniform temperature. This temperature has been obtained by the empirical measurements for the installed collectors in Kashan with 51° longitude and 30 minutes in east direction and 34° latitude and 5 minute in north direction located in Iran for the first and second period of the year. The glass cover has the constant heat flux boundary condition that the inlet heat flux value for the first and second period of the year is obtained from[2]. The flow inside the channel is considered steady and turbulent. The ambient temperature for the both periods of the year is different too and these data are collected from Iran's weather forecast organization for Kashan [20].

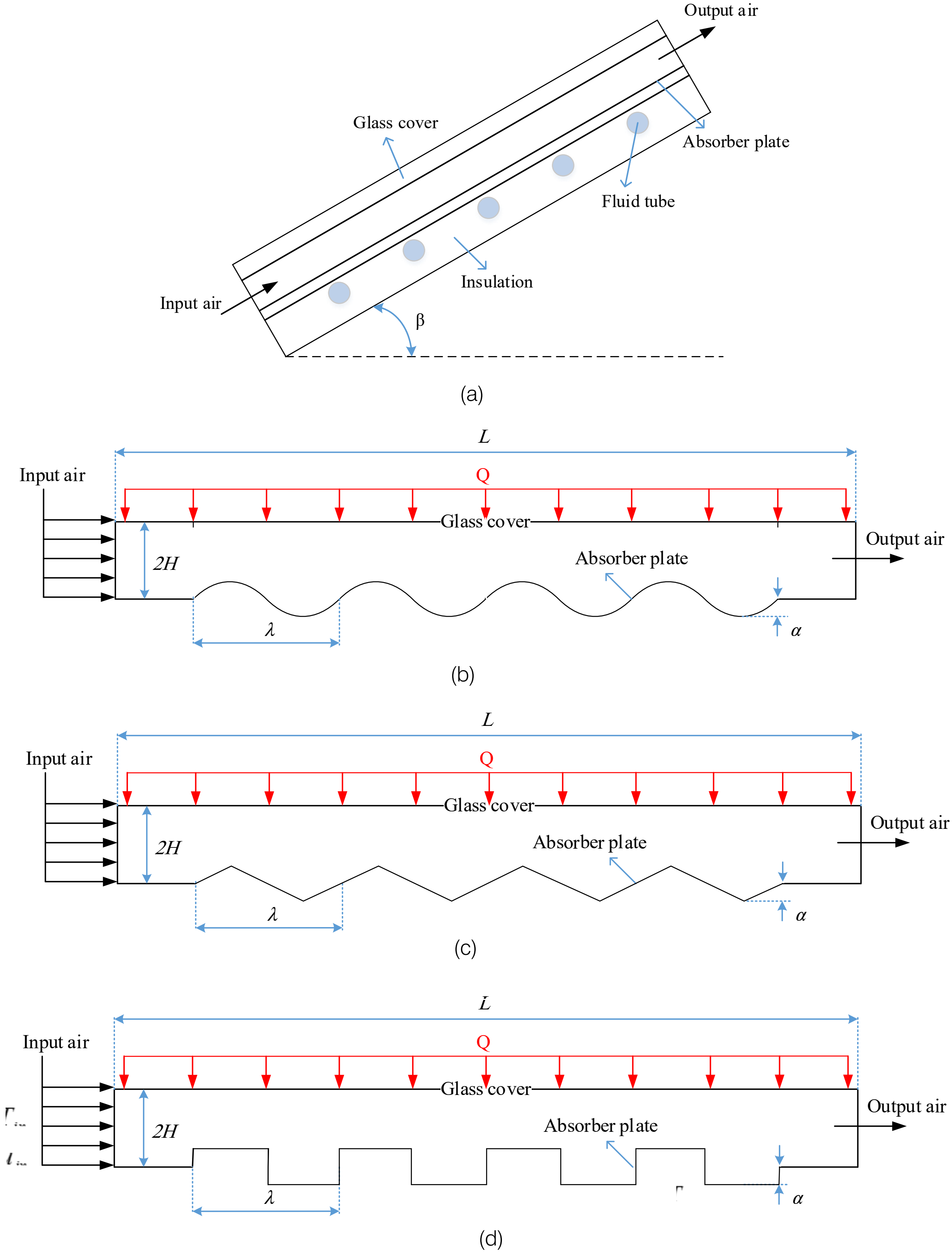


Figure 1: (a) The schematic diagram of the two dimensional collectorwith double application

Two dimensional solar collector models with:(b) sinusoidalcorrugation,(c) triangular corrugation, and(d) rectangular corrugation. (The actual geometry consists of 18 corrugations but for clear presentation only 4 waves have been shown.)

b) Governing equations

In this section the governing equations related to heat transfer and flow field are presented [21]. The considerations are as follows:

1. Steady state and two-dimensional.
2. Incompressible flow.
3. The flow properties are independent of the temperature.

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

where ρ is the density and u_i is the axial velocity. Momentum equation:

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}(-\rho\overline{u_i' u_j'}) \tag{2}$$

In the above equation μandu'arefluid viscosity andfluctuated velocity, respectively. The term of$\rho\overline{u_i' u_j'}$indicates the Reynolds stress.

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left[(\Gamma + \Gamma_t)\frac{\partial T}{\partial x_j}\right] \tag{3}$$

whereΓ and Γ_t are the molecular thermal diffusivity and turbulent thermal diffusivity, respectively and are defined as follow:

$$R = \frac{\mu}{Pr} \quad and \quad \Gamma_t = \frac{\mu_t}{Pr_t} \tag{4}$$

In order to model the turbulence it is necessary to model the Reynolds stress in Eq. (3). The standard k-ε model has been used for turbulence modeling. One common way to connect the Reynolds stress to average velocity gradients is Bossinisque assumption.

$$(-\rho\overline{u_i' u_j'}) = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{5}$$

The turbulent viscosity is derived from Eq. (6):

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{6}$$

The kinetic energy conservation equation and the loss rate are measured from Eq. (7) and (8):

$$\frac{\partial}{\partial x_i}[\rho k u_i] = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k - \rho\varepsilon \tag{7}$$

$$\frac{\partial}{\partial x_i}[\rho \varepsilon u_i] = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}G_k + C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{8}$$

where G_k is the turbulent kinetic energy generation rate and$\rho\varepsilon$is the loss rate and is defined by:

$$G_k = -\rho\overline{u_i' u_j'}\frac{\partial u_j}{\partial x_i} \tag{9}$$

The boundary values for turbulence adjacent to the wall are specified by enhanced wall treatment. Thecoefficients$C_\mu = 0.09$, $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$ and $Pr_t = 0.85$are chosen as empirical coefficients in turbulence transport equation [22]. The Nusselt number, Reynolds number, friction factor,performance evaluation criteria and heat performance coefficientare non-dimensional parameters that are calculated from the below equations [17, 23]:

$$Nu_{av} = \frac{h_f D_h}{k_f} \tag{10}$$

In the above equation h and k are conducting heat transfer coefficient and convective heat transfer coefficient respectively. In the present work the Nusselt number is measured on the absorber plate.

$$Re = \frac{\rho_f u_{in} D_h}{\mu_f} \tag{11}$$

In this equation u_{in} is the average velocity of the fluid in the collector inlet. The hydraulic diameter is also defined as follows:

$$D_h = 2H + \alpha \tag{12}$$

$$f = \frac{2}{\left(\frac{L}{D_h}\right)}\frac{\Delta P}{\rho_{nf} u_{in}{}^2} \tag{13}$$

where $\Delta P$is the pressure difference between collector inlet and outlet.

$$\Delta P = P_{av,inlet} - P_{av,outlet} \tag{14}$$

Where$P_{av,inlet}$ and $P_{av,outlet}$ are average pressure in inlet and outlet, respectively.

In order to compare the effect of Nusselt number change to pressure drop with corrugated absorber plate usage toward smooth absorber plate theperformance evaluation criteria(PEC) is calculated by the below equation [24]:

$$PEC = \left(\frac{Nu}{Nu_0}\right) \cdot \left(\frac{f}{f_0}\right)^{-1/3} \quad (15)$$

In the above equationNu and Nu_0 are average Nusselt number in corrugated collector and average Nusselt number in collector with smooth absorber plate, respectively. On the other hand f and f_0 are friction factor inside the corrugated collector and collector with smooth absorber plate respectively.

Eq. 16 is used to compare the effect of using the nanofluid on average Nusselt number and pressure drop toward base fluid usage [25]:

$$\eta = \left(\frac{Nu_{nf}}{Nu_f}\right) \cdot \left(\frac{f_{nf}}{f_f}\right)^{-1/3} \quad (16)$$

where Nu_{nf} and f_{nf} are average Nusselt numberand friction factor in collector with nanofluid, respectively and Nu_f and f_f are average Nusselt number and friction

The temperature difference from inlet to outlet is computed by:

$$\Delta T = T_{av,outlet} - T_{av,inlet} \quad (17)$$

where $T_{av,outlet}$ and $T_{av,inlet}$ are average temperature in inlet and outlet.

Thermal diffusion coefficient, kinematic viscosity and prantdl number for fluid and nanofluid are calculated from the below equations:

$$\alpha = \frac{k}{\rho\, C_p} \quad (18)$$

$$\vartheta = \frac{\mu}{\rho} \quad (19)$$

$$Pr = \frac{\vartheta}{\alpha} \quad (20)$$

The local Nusselt number in the isothermal wall is measured by [18]:

$$Nu_c = -\left(\frac{k_{nf}}{k_f}\right)\frac{\partial T}{\partial y} \quad (21)$$

In the above equation $\frac{\partial T}{\partial y}$ is the temperature gradient in thermal boundary layer.

c) Model validation

i. *Grid independence test*

The grid independence test was done for collector with air fluid. According to Fig. 2, four different grids with 143476, 145327, 149771 and 151825 nodes are considered for the smooth absorber plate model. By comparing the four cases the grid with 149771 nodes is chosen as an acceptable grid.

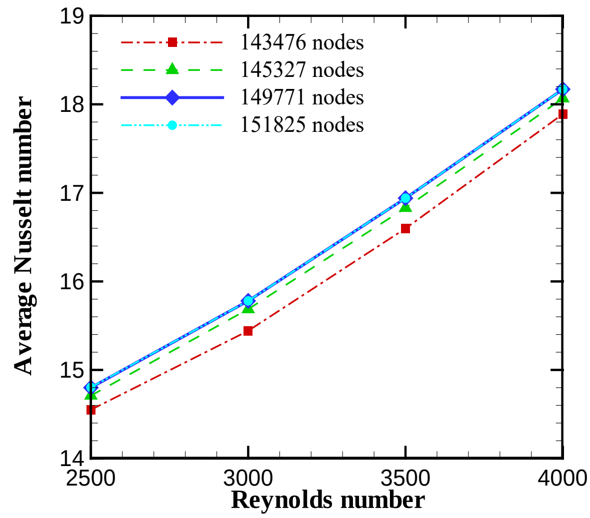
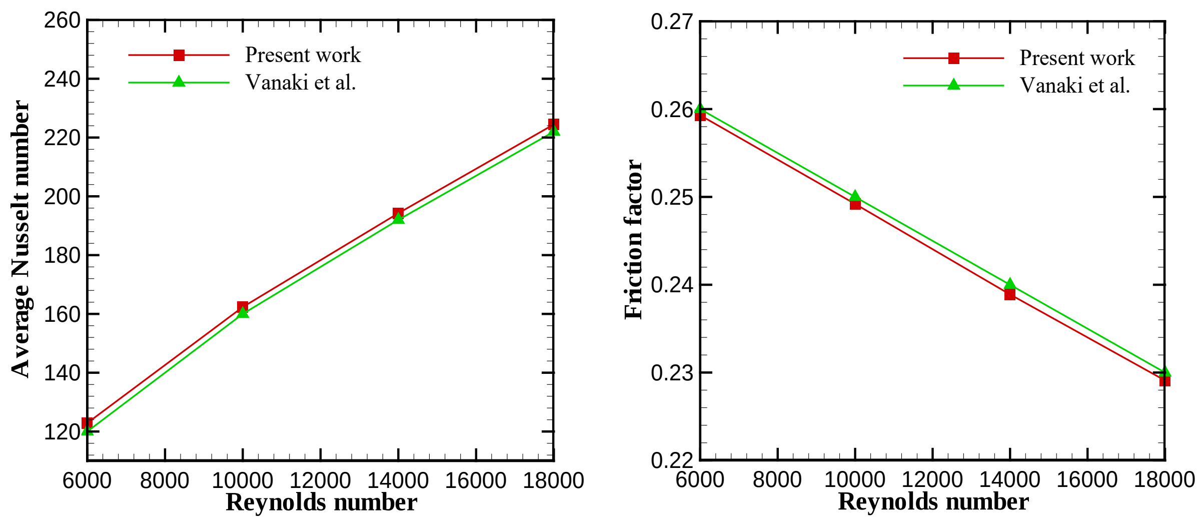


Figure 2: Average Nusselt number variation diagram according to Reynolds number for different grid sizes in smooth absorber plate

ii. *Validation*

The computer software validation was done based on the geometry and boundary condition of [24]. In this study the heat transfer performance and water flow forced convective heat transfer in a heated channel with corrugated wall is investigated. The average Nusselt number and friction factor are compared in Fig. 3. It is clear that there is a good conformity between the results.



(a) (b)

Figure 3: Current research validation for (a) average Nusselt number,(b)friction coefficient, in comparison with [28] in a sinusoidal corrugated channel with phase shift of180° and amplitude of1 mm

d) Thermophysical properties of nanofluid

The nanofluid density and specific heat transfer in a reference temperature T_{in} is shown with ρ_{nf} and $(C_p)_{nf}$ respectively [19]:

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_{np} \tag{22}$$

$$(\rho C_p)_{nf} = (1-\phi)(\rho C_p)_f + \phi(\rho C_p)_s \tag{23}$$

The carbon black nanoparticles in nanofluid mixture are considered spherical and the thermal conductivity is computed by [18]:

$$\frac{k_{nf}}{k_f} = \frac{k_s + 2k_f - 2\phi(k_f - k_s)}{k_s + 2k_f + \phi(k_f - k_s)} \tag{24}$$

The nanofluid viscosity is calculated by [26]:

$$\mu_{nf} = \frac{\mu_f}{(1-\phi)^{2.5}} \tag{25}$$

The thermo physical properties of carbon black nanoparticles and air are listed in Table 1.

Table 1: The thermophysical properties of carbon black nanoparticles and air at T=300 K [18]

Thermophysical properties	Air	Carbon black
$\rho(Kg/m^3)$	1.225	2000
$C_p(J/Kg\cdot K)$	1006.43	710
$k\ (W/m\cdot K)$	0.0242	2000
$\mu(Ns/m^2)$	0.000017894	-

e) Numerical procedure

A steady numerical simulation from the flow field was considered through the two-dimensional corrugated channel tosolv and investigates the flow and heat transfer model. The control volume method and SIMPLE algorithmwere applied to solve the equations. The turbulent standard k-εmodel was used with enhanced wall treatment. The numerical calculation was carried out by solving the governing equations with boundary conditions and with finite volume method. The diffusion term in momentum and energy equations and for convective term werediscretisized by the second order backward difference. The error value was10^{-5} in order all parameters converge.

f) Boundary conditions and environmental properties

The average daily temperature of Kashan for the first and second period of the year is 297 K and 282 K, respectively. These data are collected from the average temperature for the first and second period of the year based on the Iran's weather forecast organization [24]. The ambient pressure for Kashan is 88588 Pa [24] and the gravity acceleration is 9.806. The heat flux on the glass cover is computed by:

$$Q = \frac{\bar{H}}{3600\ t} \tag{26}$$

In this equation$\bar{H}$is the six months average for daily flux on horizontal surface and t is the average sunny hours during the day. The glass transmission coefficient is 0.88 and absorption coefficient for aluminum with dark cover is 0.95 [27]. Because this work is based on the six months setting of the flat plate collector, the average sunny hours, average daily temperature, and monthly average of daily flux from sun on the horizontal surface must be calculated in Kashan for the first and second six months period. Therefore with using equation (26) the received flux amount by glass cover is obtained for the first and second six months period. In addition the empirical calculation of absorber plate temperature during the year for Kashan shows that the average temperature of absorber plate for this city is approximately constant for a specified period of time. The results of these measurements are presented in Table 2.

Table 2: The average sunny hours, average daily temperature, monthly average daily flux received from sun on the horizontal surface for Kashan, the received flux by glass cover and absorber plate temperature during the first and second six months of the year

Period of time	Average sunny hours during the day	Average daily temperature	Monthly average daily heat flux on horizon surface$\overline{H}$	Received heat flux by glass cover	Absorber plate temperature
	(hr)	(K)	(MJ/m^2.day)	(W/m^2)	(K)
Spring and Summer	10.25	297	25.35	687	355
Fall and Winter	7.6	282	13.87	507	345

III. Result and Discussion

In this section the effects of using the corrugated absorber plate for different Reynolds number on flow and heat transfer field areinspected.Also the effects of the carbon black nanoparticle are studied. Whether the optimized thermal-hydraulic performance is of high interest or outlet temperature increase, the appropriate shape of corrugations is chosen. adopting the appropriate corrugation shape for optimized thermal-hydraulic performance is based on equation(15) and for the highest air temperature in outlet is based on equation(17). Then for the optimized corrugation shape the effect of using nanofluid on flow and heat transfer is investigated.

a) *The effect of corrugation shape of the absorber plate on thermal-hydraulic characteristics*

In this section the effect of using corrugated absorber plate with different shapes on the flow and heat transfer is analyzed for the first and second six months period. In Figures4aand 5a, the diagram of average Nusselt number change according to Reynolds number for the first and second six months period is shown, respectively. It is observed that by increasing the Reynolds number,the average Nusselt number increases too. In fact higher Reynolds numbers indicate higher velocities that lead to turbulent flow and therefore enhance the heat transfer. The results reveal that the average Nusselt numbers for corrugated absorber plate are always higher than the smooth plate. This is due to more turbulence and consequently thinner boundary layer in corrugated channels that causes higher temperature gradients. The triangular corrugated channels in the first and second six months of the year have the highest average Nusselt number in all Reynolds number and also they can intensify the heat transfer to 27%and 25% in comparison with smooth channels in Reynolds numbers equal to 2500 for the first and second six months of the year, respectively. Furthermore comparing to smooth channels, the corrugated absorber plate with sinusoidal shape can increase the heat transfer to 25% and 23% in Reynolds number of 2500, for the first and second six months of the year, respectively.

Figures 4d and 5b show PEC according to Reynolds number. The PEC values have decreasing and similar behavior for all models in the first and second six months of the year within investigated Reynolds number and for the mentioned collectors PEC decreases by increasing the Reynolds number.As it is seen in figure 4although Nusselt number increases by increasing the Reynolds number, the pressure drop is also increasing. However increasing the Nusselt number cannot conquer the growth of pressure drop and finally it leads to PEC decrease by increasing Reynolds number. Therefore for the maximum value of the PEC it is possible to determine the optimized Reynolds number for each absorbershape. The optimized Reynolds number for all models is 2500. The collector with sinusoidal absorber plate has the best PEC among all configurations that is 1.08 and 1.06 for first and second six months of the year, respectively.

Figures 4c and 5c show the diagram of temperature increase variation from inlet to outlet in the range of mentioned Reynolds number for the first and second six months of the year. It is observed that the temperature increase in corrugated collectors is more than the smooth collectors in all corrugated models during both periods. Among these collectors the rectangular corrugated collectors during the first and second six months have more temperature increase from inlet to outlet and after that the sinusoidal and triangular corrugated collectors are ranked, respectively.

The highest value of temperature increase in the low Reynolds range for the rectangular corrugated was about 63 K for the first six months and 59 K for the second six months these values for the sinusoidal and triangular models were 59 K and 55 K, respectively while these values were 53 K for the first and 48 K for the second six months of the year in smooth collectors.

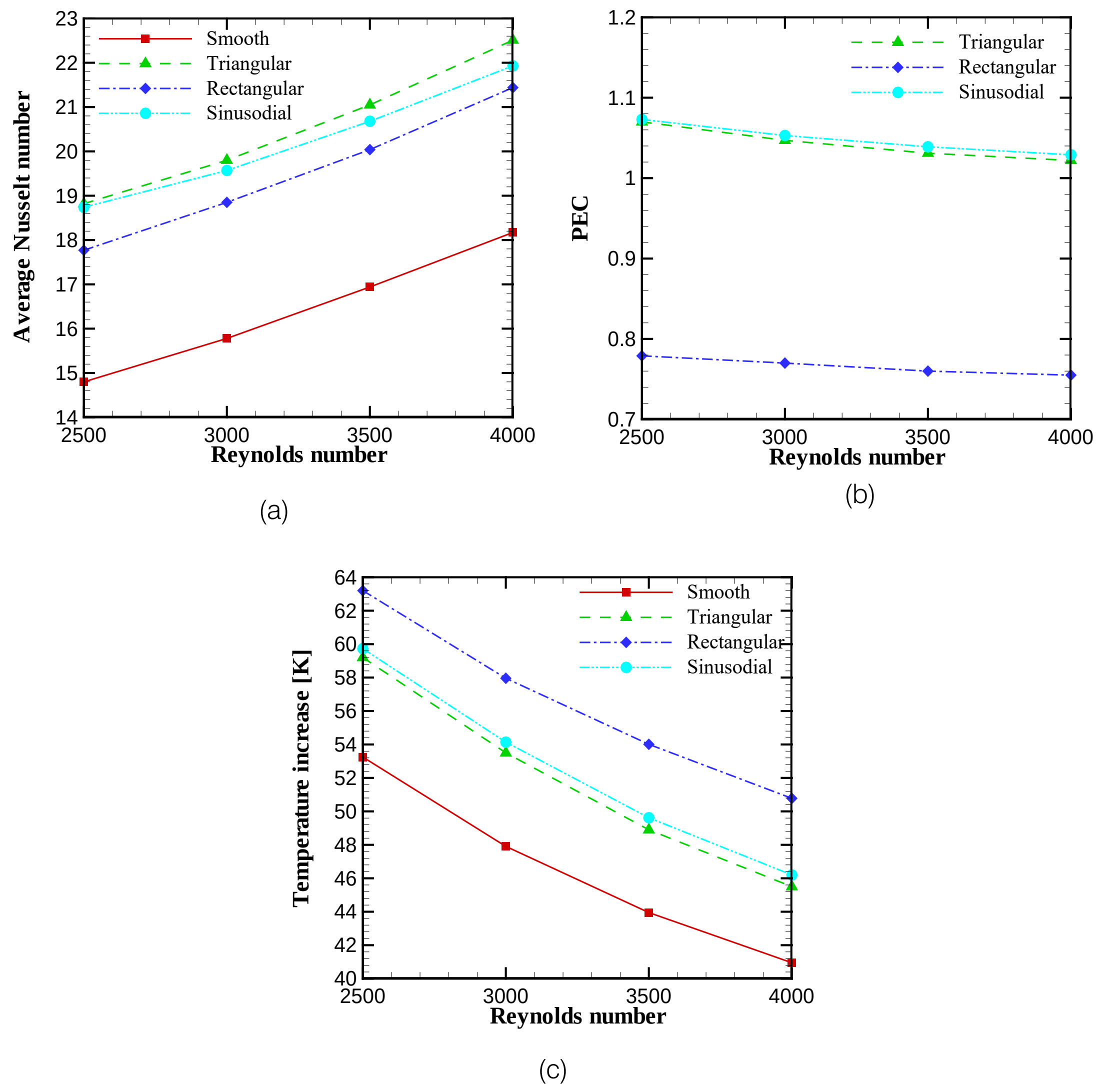


Figure 4: Variation diagram of (a) average Nusselt number, (b) PEC, (c) air temperature increment from collector inlet to outlet, according to different Reynolds number for corrugated and smooth absorber plates during the first six months

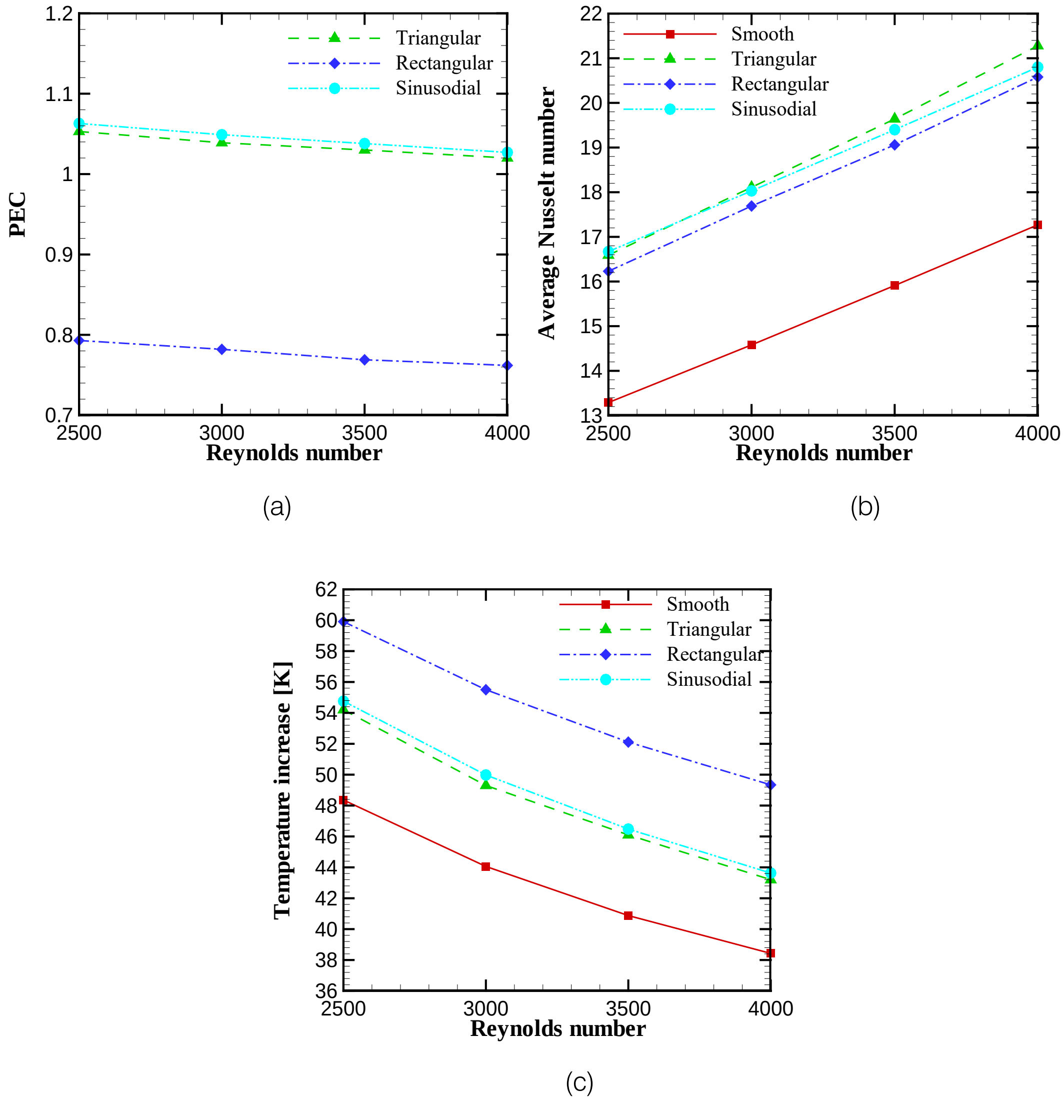


Figure 5: Variation diagram of (a) average Nusselt number, (b) PEC, (c) air temperature increment from collector inlet to outlet, according to different Reynolds number for corrugated and smooth absorber plates during the second six months

The isothermal regions at the first, middle and last corrugation of collector are shown in Fig.6 to analyze the flow and heat transfer field more precisely. As it is demonstrated in the figure the thermal boundary layer growth inside the channel, adjacent to the absorber plate is more in rectangular corrugation and as a result the great part of the fluid is affected by the high temperature of absorber plate and consequently the outlet temperature increases more. On the other hand it is seen that near the glass cover three models have the same condition and in the case of the temperature distribution there is no preferenceto each other.

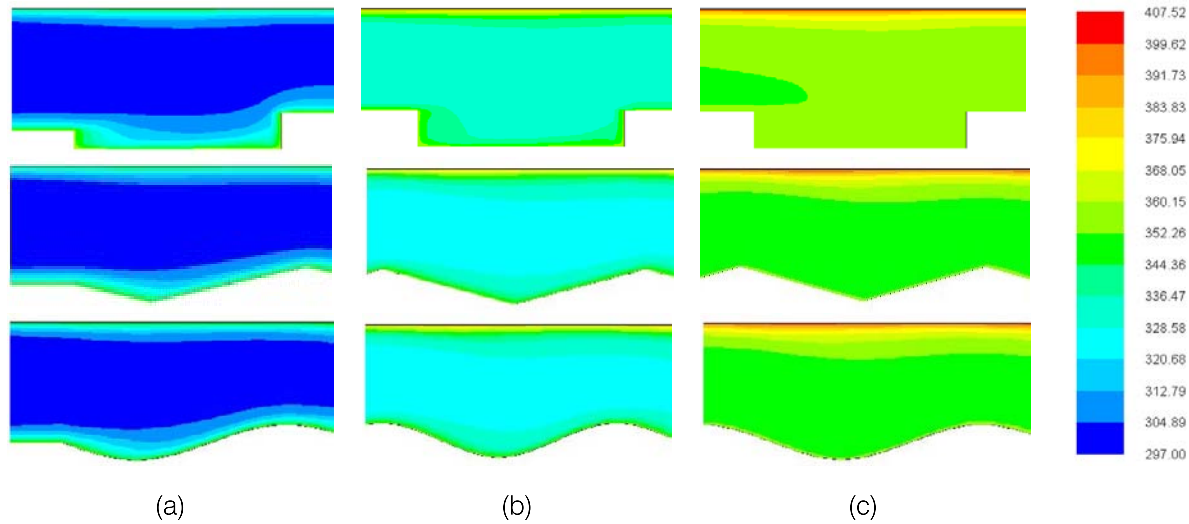


Figure 6: Temperature distribution in the region of (a) the first corrugation,(b)the ninth corrugation,(c)the last corrugation for all three different corrugation of rectangular, triangular and sinusoidal in Reynolds of 2500 during the first six months of the year

Fig.7 shows the velocity vectors in the ninth corrugation of each rectangular, triangular and sinusoidal model.As it is evident in the figure the reverse flow and vortex forms is obviously seen near the absorber plate. In triangular corrugation there was a little reverse flow close to the absorber plate and in sinusoidal model there was no reverse flow. The formed vortexes in rectangular corrugations trap the flow in a part of channel and decrease the velocity to stagnation point this leads to Nusselt number and heat transfer reduction.

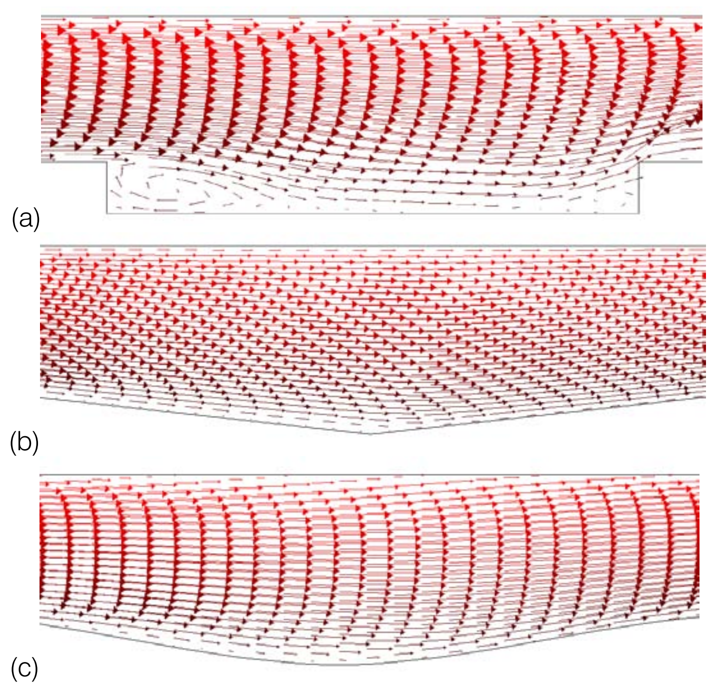


Figure 7: Velocity vectors in the region of the ninth collector corrugation for three different corrugations (a) rectangular, (b) triangular, and (c) sinusoidal, in the Reynolds of 2500 during the first six months of the year

According to the aforementioned discussion the sinusoidal corrugated model is chosen as an optimized model in case of thermal-hydraulic performance due to highest PEC.However in the case of pressure drop it is the best model and overall by considering these two parameters together it was chosen as the most appropriate model for the conditions that heat is needed to be transferred to installed pipes under the absorber plate and also by the optimized performance of the pumping system with the least losses,the air temperature increases, throughout the year. On the other hand for the condition that the solar collector is supposed to be used increasing the air temperature and the pumping system performance and losses are not important, the rectangular corrugated model was adopted because it has the most temperature increase among all models.

b) The effect of using nanofluid in different volume fractions

In this part the effect of using the aerosol-carbon black nanofluid with spherical nanoparticle and different volume fractions on flow and heat transfer field of rectangular and sinusoidal corrugations is investigated. Figures 8 and 9 show the diagram of non-dimensional thermo physical properties change for aerosol- carbon black nanofluid in different volume fractions of carbon black nanoparticles. As can be seen in Fig.8 by increasing volume fraction of carbon black nanoparticle from 0 to 1% the thermal conductivity and dynamic viscosity of nanofluid according to base fluid do not have the intense variation and ultimately each of them increases to 3% and 2.5%, respectively. But as can be seen the nanofluid specific heat variation compared to base fluid has an intense changes and it increases to 28%. In contrast it is observed in Fig.9that the nanofluid density in volume fraction of 1% to 17.5% increases like base fluid density.

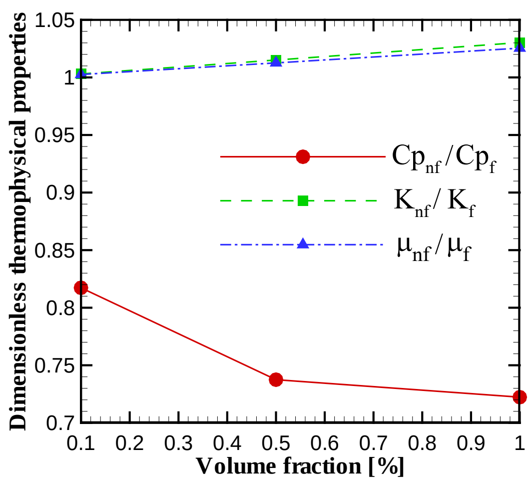


Figure 8: Non-dimensional specific heat, thermal conductivity and dynamic viscosity according to carbon blacknanoparticle volume fraction in nanofluid mixture

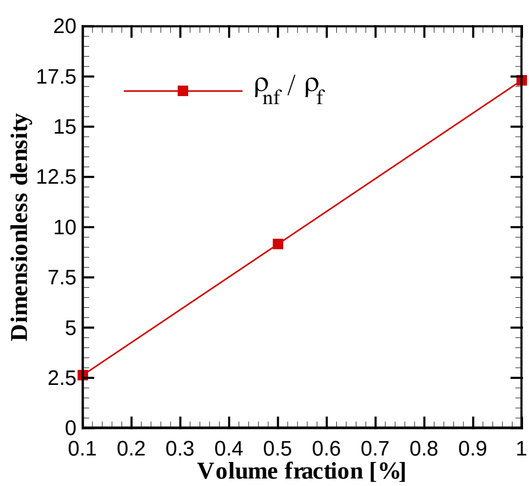


Figure 9: Non-dimensional density variation according to carbon black nanoparticle volume fraction in nanofluid mixture

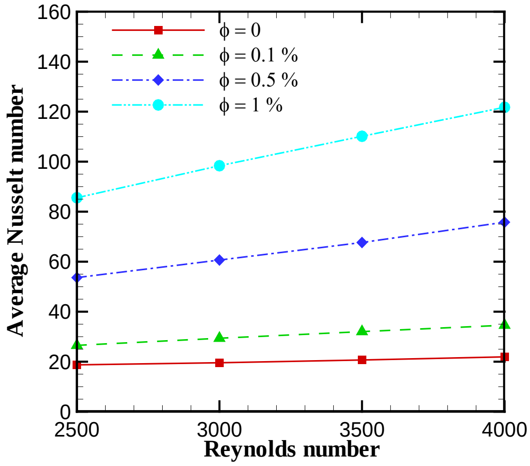
As can be seen in Figures10a and 11a by increasing the volume fraction of carbon black nanoparticles in air, the average Nusselt number increasesas well inside the collector with sinusoidal absorber plate for the first and second six months of the year. As it is observed in the figure for example in Reynolds number of 4000 with volume fraction of 1% the Nusselt number increases by 455% and 483% for the first and second months of the year, respectively. It was perceived that in the Fig.8 nanofluid thermal conductivity increase is not much comparing to the base fluid so it is not possible to attribute this increase in Nusselt number to thermal conductivity however it has an impact on Nusselt number increase certainly. But as it was seen the nanofluid density and specific heat have great changes comparing to base fluid. The calculation with equations of (18) to (20) showed that the Prandtl number is approximately 0.74415 in the case of using the base fluid this demonstrates that the thermal boundary layer thickness is more than velocity boundary layer. By increasing the volume fraction of carbon black nanoparticles the Prandtl number decreased and for instance it reached to 0.53497 in volume fraction of 1%.Besides in the case of using nanofluid the thermal diffusion coefficient and kinetic viscosity decrease were noticed while the kinetic viscosity decrease was more intense. These items show that in the case of using nanofluid the thermal and velocity boundary layers become thinner. The intense increase in thermal boundary layer based on equation(21) leads to temperature gradient increase within thermal boundary layer and consequently the Nusselt number increases on the absorber surface as well as nanofluid thermal conductivity.

By increasing the nanoparticles volume fraction and Reynolds number the static pressure drop increases from inlet to outlet in collector. The reason for increasing the static pressure drop with nanoparticle volume fraction increase is the nanofluid density and viscosity increment. In addition by increasing the nanofluid volume fraction the friction factor decreases. The reason for this is more intense growth in nanofluid dynamic pressure according to static pressure drop because the nanofluid density is siginificantlyincreasing. Figures 10b and 11b show the heat performance coefficient diagram in the case of using nanofluid in different volume fractions and for various Reynolds numbers in a collector with sinusoidal absorber plate during the first and second six months of the year, respectively. As it is seen the heat performance coefficientincreases by increasing Reynolds number.

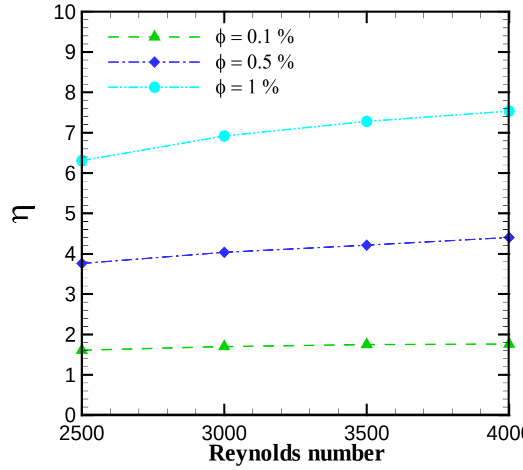
This is due to increasing the Nusselt number and decreasing friction factor with Reynolds number increment. Furthermore by increasing the volume fraction,heat performance coefficientincreases too. The reason for this is similar increase in the Nusselt number and decrease in thefriction factor by Nusselt number increment. Consequently in the case of using sinusoidal absorber plate with aerosol-carbon black nanofluid, the highest heat performance coefficient was obtained in Reynolds 4000 and volume fraction of 1% for both first and second six months of year.

Figures 10c and 11c show the diagram of temperature increase changes from collector inlet to outlet in the range of discussed Reynolds numbers for different nanofluid volume fractions inside the collector with sinusoidal absorber plate during the first and second six months of the year. It is observed that for all volume fractions during both periods of time the temperature increment inside the collector having nanofluid is less than the collector having the base fluid.

In the volume fraction of 1% during both periods of time the least temperature increase was observed from inlet to outlet and after that the volume fractions of 0.5% and 0.1% were like that respectively. The least temperature increment in low Reynolds numbers was 20 K for the first six months of the year and 22 K for the second six months of the year in the volume fraction of 1% and these values were 40 K and 38 K for the volume fraction of 0.1% respectively. While for the collector having the base fluid these amounts were 50 K for the first six months of the year and 54 K for the second six months of the year. As a result if the temperature increase from inlet to outlet is only considered for the collector with sinusoidal absorber plate the nanofluid usage will not be recommended because it substantially decreases the temperature increment from inlet to outlet. In addition it is proposed to use the collector in low Reynolds number since by increasing the fluid velocity the temperature increment from inlet to outlet decreases throughout the year.



(a)



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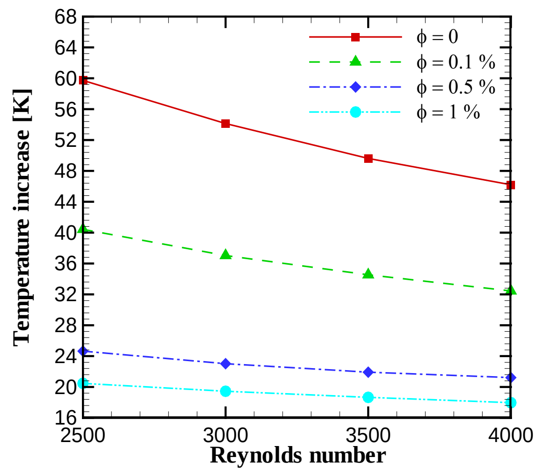
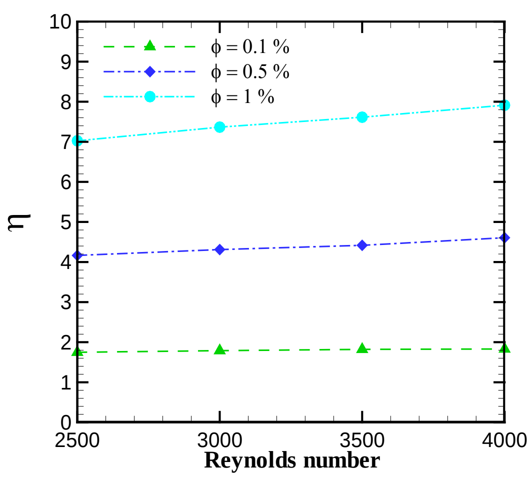


(c)

Figure 10: Variation diagram of (a) average Nusselt number, (b) heat performance coefficient, (c) air temperature increment from collector inlet to outlet, according to different Reynolds number for sinusoidal corrugated absorber in various nanoparticle volume fractions during the first six months of the year



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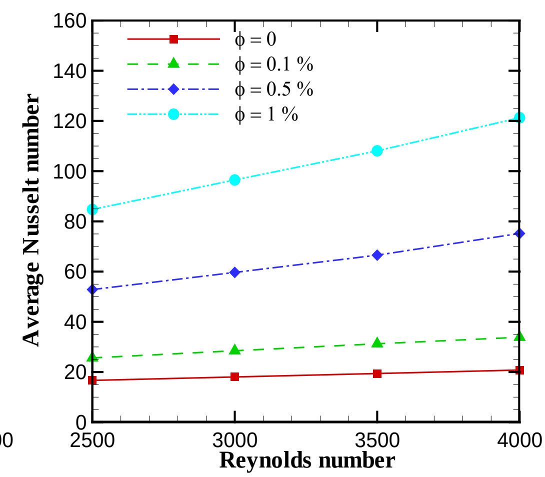

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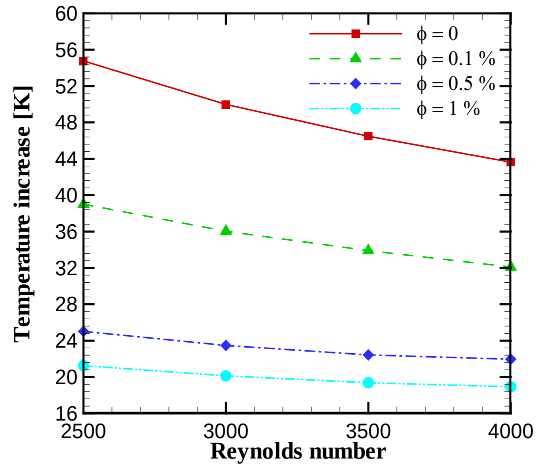
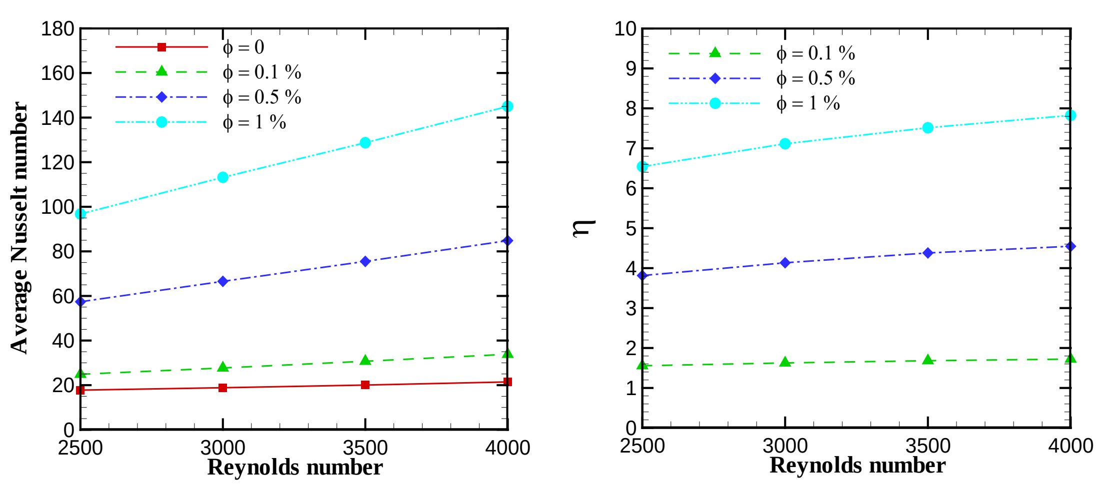


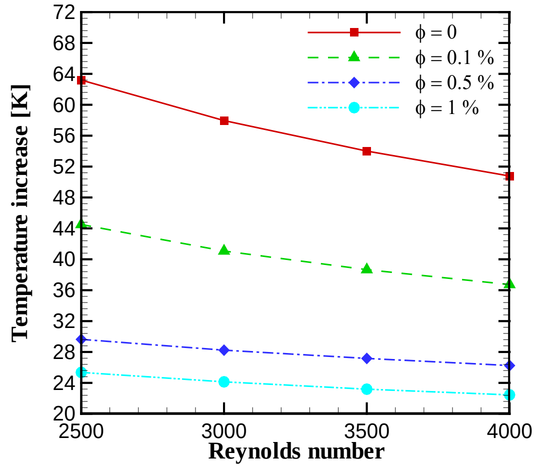
(c)

Figure 11: Variation diagram of (a) average Nusselt number, (b) heat performance coefficient, (c) air temperature increment from collector inlet to outlet, according to different Reynolds number for sinusoidal corrugated absorber in various nanoparticle volume fractions during the second six months of the year

As can be seenin Figures 12a and 13a, by increasing the volume fraction of carbon black nanoparticles in air, the average Nusselt number increases too inside the collector with rectangular absorber plate for the first and second six months of the year.Based on figures 12b and 13b by increasing the Reynolds number and volume fraction the ηvalue increases. Therefore it was found out that in the case of using the rectangular absorber plate with aerosol-carbon black nanofluid the highest thermal performance coefficient is obtained in the Reynolds of 4000 and volume fraction of 1% during the first and second six months of the year.Figures 12c and 13c show the diagram of temperature increase variation from collector inlet to outlet in the range of discussed Reynolds numbers for different nanofluid volume fractions inside the collector with rectangular absorber plate during the first and second six months of the year.It is seen that in all volume fractions during both periods of time, the temperature increment in a collector that has a nanofluid is less than the collector with the base fluid. In the volume fraction of 1% during both periods of time the least temperature increase was observed from inlet to outlet and after that the volume fractions of 0.5% and 0.1% were ranked, respectively. The least temperature increment in low Reynolds numbers for volume fraction of 1% was 25 K during the first six months and it was 26 during the second six months and these values were approximately 44 K for volume fraction of 0.1%. Hence in case of using the model with rectangular corrugated absorber plate, if fluid temperature increment from inlet to outlet is only considered, the nanofluid usage will not be recommended because it considerably decreases the temperature increase from inlet to outlet. Moreover it is advised to use the collector in low Reynolds numbers due to the temperature increase from collector inlet to outlet decreases by velocity increment in all cases throughout the year.

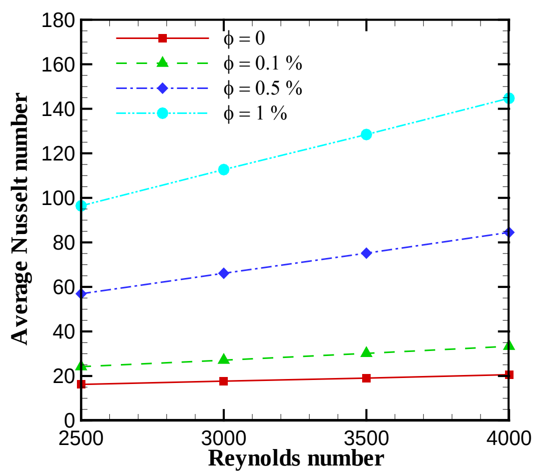


(a) (b)



(c)

Figure 12: Variation diagram of (a) average Nusselt number, (b) heat performance coefficient, (c) air temperature increment from collector inlet to outlet, according to different Reynolds number for rectangular corrugated absorber in various volume fractions during the first six months of the year



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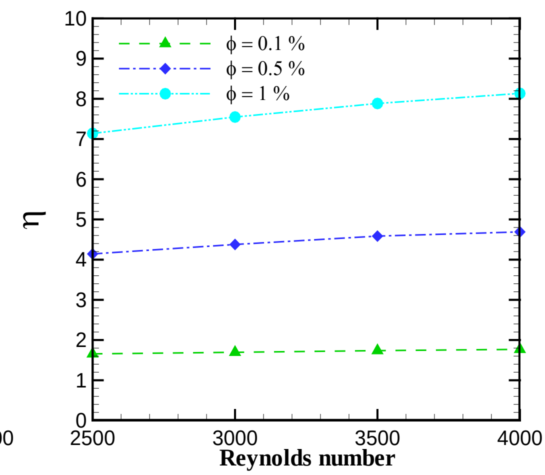

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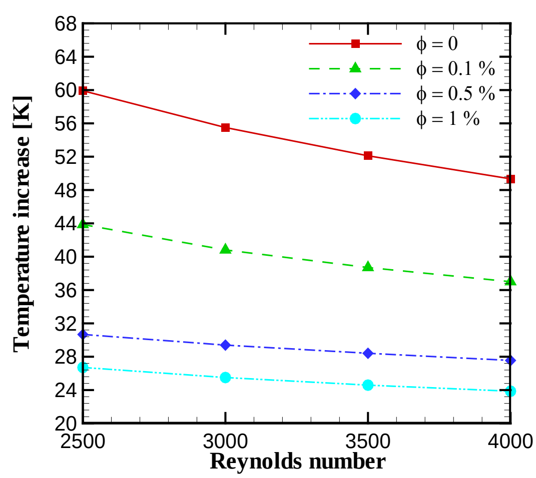

(c)

Figure 13: Variation diagram of (a) average Nusselt number, (b) heat performance coefficient, (c) air temperature increment from collector inlet to outlet, according to different Reynolds number for rectangular corrugated absorber in various volume fractions during the second six months of the year

IV. Conclusion

A numerical study was carried out in order to investigate the thermal-hydraulic behaviors of air forced convective heat transfer inside the collector with dual usage and corrugated absorber plate for turbulent regime in the range of turbulent Reynolds numbers between 2500 and 4000. The solar collector with dual usage means a collector that is able to transfer the heatto the fluid inside the installed pipes under the absorber plate and also the heat transfer to the air passing between the absorber plate and glass cover. The focus of the present study has been on enhancing the heat transfer because of corrugated absorber plate by breaking the laminar sub-layer and producing local wall turbulence due to flow separation and adherence between successive grooves again that decreases the thermal resistance and intensifies the heat transfer considerably. As stated by the results corrugating the absorber plate improves the thermal characteristics like Nusselt number and temperature increment from inlet to outlet but in the case of hydraulic characteristics it enhances the losses. For this reason by defining the PEC that is a compromised point between improving the heat transfer characteristics andpressure drop compensation, it is possible to obtain the optimized model. On that account the results indicate that despite this fact that the collectors with triangular and rectangular absorber plate have the highest Nusselt number as well as highest temperature increase from inlet to outlet throughout the year respectively due to friction factor increase and pressure drop compensation but since the collector with sinusoidal absorber plate has the highest PEC for the whole year, the corrugated sinusoidal model is introduced as the optimized model in the current study. On the other hand if the air temperature increase is only considered the rectangular corrugated model is the optimized one. The results revealed that in the case of using the air base fluid whether in term of temperature increase from inlet to outlet or in term of the highest PEC,the optimized Reynolds number is 2500. For each of the sinusoidal and rectangular corrugated models throughout the year the carbon black nanoparticles were added to the air base fluid in the volume fractions of 0.1% to 1%. The results showed that in sinusoidal model which is used because of transferring more heat to the fluid inside the pipes installed under the absorber plate and also the outlet air temperature increase between the absorber plate and glass cover, the nanoparticle volume fraction increase leads to thermal performance coefficient increment and in Reynolds number of 4000 and volume fraction of 1% the optimized model was obtained for the whole year. In rectangular corrugated model that is simulated to increase the air temperature only the nanofluid usage and Reynolds number increment are not useful at all and lead to outlet temperature decrease. So for this model the base fluid usage and 2500 Reynolds number are recommended.

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First and Second Laws Analysis and Optimization of a Solar Absorber; Using Insulator Mixers and MWCNTs Nanoparticles

By Soroush Sadripour
University of Kashan

Abstract- In this paper, forced convection flow and heat transfer of MWCNTs-water nanofluid in heat sink collector equipped with mixers are studied. The three-dimensional governing equations are numerically solved in the domain by the control volume approach based on the SIMPLE algorithm. Reynolds numbers are considered in laminar-turbulent range of 50<Re<12,000. The optimization was carried out by comparison of different parameters to reach the optimal case with the maximum exergy efficiency. From this study, it is concluded that in the case of using heat sink, instead of shell and tubes, the time that the fluid is inside the collector increases and leads to outlet temperature increase from the collector the exergy efficiency increases. Also, it is realized that using mixers enhance the outlet fluid temperature, energy efficiency and exergy efficiency.

Keywords: heat sink collector, exergy optimization, radiation, forced convection, mixer, MWCNTS.

GJRE-A Classification: FOR Code: 850599

FIRSTANDSECONDLAWSANALYSISANDOPTIMIZATIONOFASOLARABSORBERUSINGINSULATORMIXERSANDMWCNTSNANOPARTICLES

Strictly as per the compliance and regulations of:

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Soroush Sadripour

Abstract- In this paper, forced convection flow and heat transfer of MWCNTs-water nanofluid in heat sink collector equipped with mixers are studied. The three-dimensional governing equations are numerically solved in the domain by the control volume approach based on the SIMPLE algorithm. Reynolds numbers are considered in laminar-turbulent range of 50<Re<12,000. The optimization was carried out by comparison of different parameters to reach the optimal case with the maximum exergy efficiency. From this study, it is concluded that in the case of using heat sink, instead of shell and tubes, the time that the fluid is inside the collector increases and leads to outlet temperature increase from the collector the exergy efficiency increases. Also, it is realized that using mixers enhance the outlet fluid temperature, energy efficiency and exergy efficiency. Generally, while the trend of exergy efficiency variation with effective parameters is increasing, applying the mixers precipitate the efficiency increment. In addition, for the case that the trend of exergy efficiency variation with changing these parameters is decreasing, the decreasing trend gets slow. Finally, the highest exergy efficiency was obtained for the nanoparticle volume fraction of $\phi=0.10\%$.

Keywords: heat sink collector, exergy optimization, radiation, forced convection, mixer, MWCNTS.

I. Introduction

The use of solar energy offers numerous advantages, especially in Iran where levels of radiation from the sun are much higher than average and where many provinces lack any centralized infrastructure to support a national energy supply. While the demand for energy rapidly increases in Iran, using the necessary technology for converting energy from the sun's rays into useful energy is much important for the vast majority of the population [1]. Solar energy has a remarkably higher potential compared to other renewables energy, such as wind, ocean, hydro, biomass and geothermal. There are many types of systems that employ solar energy collectors as a source of input energy to drive a process. Between these all systems the flat-plate solar collector comparing with other collector types has simple design and low costs of construction. In addition to direct solar radiation absorption they can also absorb the diffuse radiation [2]. So far a lot of numerical and empirical studies related to solar collectors have been conducted. The results of these studies demonstrate that the overall performance of collector is related to many factors including the distance between absorber plate and glass cover and pipe diameter [3, 4], wind velocity [5], solar radiation [6], collector material [7], flow rate [8], and channel depth [9]. But one proper solution to improve the efficiency of solar collectors is to use heat sink below the absorber plate instead of pipes. It can increase the wetted surface between fluid and absorber, and also increase outlet temperature of fluid. Furthermore, employing mixers in the heat exchangers has been one of the frequent approaches to break the laminar sub-layer and create local turbulence due to flow separation and reattachment between successive obstacles, which reduces the thermal resistance and significantly enhances the heat transfer [10]. This paper focuses on energy analysis of heat sink flat plate solar collector equipped with mixers for enhancing the thermal performance and the maximum energy efficiency and exergy efficiency under given operating conditions. Another method is to increase the heat transfer between fluid and solar absorbing plate. One common and suggested way is to add the nanoparticles to the base fluid used in collector.

Baniamerianand et al. [11] studied numerically aerodynamic coefficients of solar troughs considering terrain effects and vortex shedding. Their results show that in order to properly align trough collector in solar farms, it is essential to study the vortices shed created at the behind of parabolic troughs. In another numerical investigation Ziapour and Rahimi [12] investigated natural convection heat transfer in a horizontal wavy absorber solar collector based on the second law analysis. Their results show that with increasing of the cosine wave amplitude, the collector enclosure irreversibility decreases. Ajay and Kundan [13] studied performance evaluation of nanofluid (Al_2O_3/H_2O-$C_2H_6O_2$) based parabolic solar collector using both experimental and CFD techniques. Their results show a close agreement between experimental and CFD result.

A method for establishing the optimal operation mode of solar collectors derives from the exergy analysis of the processes specific for the fluid that passes through the collector's stream tube [14]. The

Author: Department of Mechanical Engineering, University of Kashan, Iran. e-mail: soroushsadripour@hotmail.com

analyzed relevant literature contains studies on the dependence of the exergy efficiency on the fluid flow rate and on the fluid temperature at the entrance into the collector serpentine pipe. The specific exergy of the fluid in the solar collector as depending on the inlet temperature, the parameter being either the solar radiation or the fluid flow rate, presents points of local maximal. These aspects are not highlighted in the energy efficiency equation. Shojaeizadeh and Veysi [15] developed a correlation for parameter controlling exergy efficiency optimization of an Al2O3/water nanofluid based flat-plate solar collector. Said et al. [16] investigated energy and exergy analysis of a flat-plate solar collector using different sizes of Aluminum oxide based nanofluid and they founded that the combination of energy and exergy analysis is an appropriate method to optimize the flat-plate solar collectors.

Mollamahdi et al. [17] investigated flow field and heat transfer in a channel with a permeable wall filled with Al_2O_3-Cu/water micropolar hybrid nanofluid, effects of chemical reaction and magnetic field. Their results show that with increasing the Hartmann number and the Reynolds number, the Nusselt and Sherwood numbers increase. Furthermore, when the hybrid nanofluid is applied rather than pure nanofluid, the heat transfer coefficient will increase significantly. Hemmat Esfe and Saedodin [18] studied numerically of combined convection flow in a cavity subjected to a nanofluid with an inside hot obstacle: effect of diameter of nanoparticles and cavity inclination angles. Their obtained results show that the average Nusselt number for all range of solid volume fraction decreases with increase in diameter of nanoparticles

The literature review elucidates that although usage of stationary obstacle to investigate the efficiency of heat sink flat-plate air heaters has been assessed, but to the best of author's knowledge there is not any study which investigates effect of using mixers and Multi Wall Carbon Nano Tubes (MWCNTs) nanoparticles in water based heat sink solar collectors on the first and second law efficiencies of solar collectors. Therefore, this study is expected to fulfill the research gap about usage of insulator mixers in heat sink solar collector using nanofluid. The other objective of this study is to investigate the effect of different suspended nanoparticles volume fractions on the energy and exergy efficiencies of water based MWCNTs nanofluids numerically using finite volume method.

II. Methodology

a) Physical Model

The three-dimensional schematic diagram of heat sink of a flat-plate solar collector equipped with mixers is shown in "Fig. 1". "Table 1" represents different properties of this heat sink collector. For simulation, useful received energy by collector is calculated based on inlet solar radiation and overall heat loss by analytical relations. Then the three-dimensional heat sink collector investigates numerically and useful received energy by fluid, outlet temperature of fluid and energy and exergy efficiencies obtained. The flow inside the channel is considered steady and turbulent. For the inlet section of the sink the velocity inlet boundary condition in is considered and for the outlet section of the heat sink pressure outlet boundary condition is assumed. The absorber plate is produced from Aluminum with matted black color and is under the uniform heat flux that is calculated with assuming optical properties and overall heat loss of collector for different sunny hours based on empirical measurements results of Khorasanizadeh et al. [2] for a reference collector in Tehran located in Iran ("Table 2"). Due to simulating mixers assumptions with no slip condition are considered. Also, all of these obstacles with diameter of D_o are insulator. Because of considering influences of overall heat loss in calculating of useful received energy by collector other walls of heat sink are assumed insulator.

Table 1: Properties of collectors simulated in present study

Properties	Symbo	Value
Dimensions of collector	$L_c \times W_c$ (mm)	200×92.5
Dimensions of inlet section	$L_i \times W_i$ (mm)	10×20
Dimensions of exit section	$L_e \times W_e$ (mm)	10×20
Height of heat sink	H (mm)	1.5
Slop of collector	B	35°
Number of glass covers	N	1
Emissivity of glass covers	ε_g	0.85
Thickness of plate	δ_p (mm)	0.1
Emissivity of plate	ε_p	0.9
Conductivity of plate	k_p (W·m^{-1}·K^{-1})	211
Optical efficiency	η_0	0.68
Thickness of insulators	δ_{ins} (mm)	2.0
Conductivity of insulators	k_{ins} (W·m^{-1}·K^{-1})	0.05
Number of mixers	n_o	3
Location of first mixer	L_{fo} (mm)	38
Location of other mixers	L_o (mm)	50
Diameter of mixers	D_o (mm)	0.5

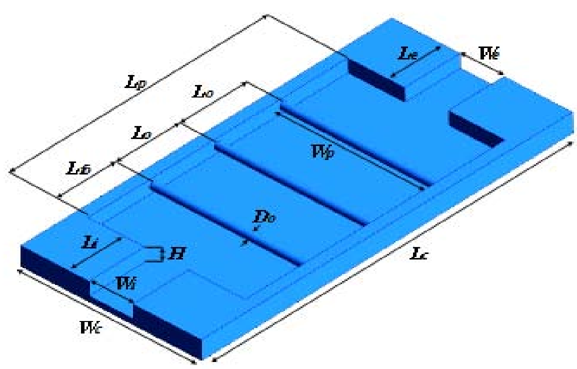

Figure 1: Schematic diagram of the heat sink of a flat plate solar collector equipped with mixers

Table 2: Empirical results of Khorasanizadeh et al. [2] for reference collector installed in Tehran

Time	I_T (W·m^{-2})	T_a (°C)	T_{in} (°C)	V_w (m·s^{-1})
09:00	560	33	44.5	6
09:30	630	33	45	6
10:00	750	34	46	5
10:30	830	35	47	6
11:00	925	36	50	6
11:30	992	37	51	5
12:00	1006	38	53	5
12:30	1020	38.5	54	6
13:00	978	40.5	56	6
13:30	914	40.5	57	5
14:00	834	41	60	5
14:30	780	41	61	4
15:00	734	39.5	62	5
15:30	626	41	63	6
16:00	607	41	64	6

b) Governing Equations

The governing equations for flow and heat transfer in the flat-plate solar collector can be written in the Cartesian tensor system as [19]:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u'_i u'_j}\right) \tag{2}$$

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left[\left(\frac{\mu}{\Pr} + \frac{\mu_t}{\Pr_t}\right)\frac{\partial T}{\partial x_j}\right] \tag{3}$$

where ρ is the density of fluid and u_i is the axial velocity, μ, $ú$ and u_j are the fluid viscosity, fluctuated velocity and the axial velocity, respectively, and the term $\rho\overline{u'_i u'_j}$ is the turbulent shear stress. The Reynolds averaged approach to turbulence modelling requires that the Reynolds stresses $\rho\overline{u'_i u'_j}$ in "Eq. 2" needs to be modelled. For closure of the equations, the k-ε turbulence model was chosen. A common method employs the Boussinesq hypothesis to relate the Reynolds stresses to the mean velocity gradient:

$$-\rho\overline{u'_i u'_j} = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{4}$$

The turbulent viscosity term μ_t is to be computed from an appropriate turbulence model. The expression for the turbulent viscosity is given as "Eq. 5". In the equation of the TKE, k is written as "Eq. 6".

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{5}$$

$$\frac{\partial}{\partial x_i}\left[\rho k u_i\right] = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k - \rho\varepsilon \tag{6}$$

Similarly, in the dissipation rate of TKE, ε is given by the following equation:

$$\frac{\partial}{\partial x_i}\left[\rho\varepsilon u_i\right] = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}G_k + C_{1\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{7}$$

where G_k is the rate of generation of the TKE while $\rho\varepsilon$ is its destruction rate, G_k is written as:

$$G_k = -\rho\overline{u'_i u'_j}\frac{\partial u_j}{\partial x_i} \tag{8}$$

The boundary values for the turbulent quantities near the wall are specified with the enhanced wall treatment method. C_μ=0.09, $C_{1\varepsilon}$=1.44, $C_{2\varepsilon}$=1.92, σ_k=1.00, σ_ε=1.30 and $\Pr_t$=0.90 are chosen to empirical constants in the turbulence transport equations [19].

The fluid is considered to be Newtonian, and the physical properties of the fluid are temperature dependent. Since the temperature variations is higher than 3°C [20]. The following polynomial expansions are used [21]:

$$\rho(T) = 5.3738\cdot10^{-10}T^5 - 9.59976\cdot10^{-7}T^4 + 6.93809\cdot10^{-4}T^3 - 0.255822T^2 + 47.8074T - 2584.53 \tag{9}$$

$$c_p(T) = -4.51782\cdot10^{-8}T^5 + 7.61613\cdot10^{-5}T^4 - 5.12699\cdot10^{-2}T^3 + 17.2363T^2 - 2894.85T + 198532 \tag{10}$$

$$k(T) = 5.15307\cdot10^{-11}T^5 - 8.15212\cdot10^{-8}T^4 + 5.138\cdot10^{-5}T^3 - 1.61344\cdot10^{-2}T^2 + 2.52691T - 157.532 \quad (11)$$

$$\mu(T) = -4.37087\cdot10^{-13}T^5 + 7.38482\cdot10^{-10}T^4 - 4.99292\cdot10^{-7}T^3 + 1.68946\cdot10^{-4}T^2 - 2.86313\cdot10^{-2}T + 1.94641 \quad (12)$$

The spectral radiative transfer equation (RTE) can be written as "Eq. 13".

$$\frac{dI_v(r,s)}{ds} = -(K_{av} + K_{sv})I_v(r,s) + K_{av}I_b(v,T) + \frac{K_{sv}}{4\pi}\int_{4\pi} dI_v(r,s')\cdot\phi(s.s')d\Omega' + S \quad (13)$$

where I_v is spectral radiation intensity which depends on position r and direction s. [22]:

$$I_v(r,s) = \varepsilon_v(r_w)I_b(v,T) + \frac{\rho_w(r_w)}{\pi}\int_{n\cdot s'<0} I_v(r,s')\cdot|n.s'|d\Omega' \quad (14)$$

The commercial available CFD software, FLUENT 15.0 was used to solve the governing equations. The control volume approach was used to solve the system of classical single phase governing equations by using the finite volume method (FVM). The standard k–ε turbulence model with enhanced wall function was selected. The diffusion term in the momentum and energy equations was approximated by second-order central difference. In addition, a second-order upwind differencing scheme was adopted for the convective terms.

The convergence criterion was considered 10^{-6} for all variables.

c) *First law modeling*

Useful received energy by fluid in collector is calculated as follow [23]:

$$\dot{Q}_{u,f} = \dot{m}_f \cdot c_p\left(T_{out} - T_{in}\right) \quad (15)$$

where $\dot{m}_f$ is mass flow rate of fluid, c_p is specific heat capacity of fluid and T_{in} and T_{out} are mean temperature of inlet and outlet fluid, respectively.

Useful received energy by collector based on inlet solar radiation and overall heat loss is as follow:

$$\dot{Q}_{u,c} = A_c\left[S - U_L\left(T_{pm} - T_a\right)\right] \quad (16)$$

where A_c is the area of absorber plate, T_a is ambient temperature and T_{pm} is mean temperature of absorber plate. It should be noticed that the temperature of absorber plate is not a constant value and considering a mean temperature for it just is a virtual concept.

In the present study temperature gradients around heat sink can be neglected and a mean temperature can be taken into account for it as far as heat sink has been spread through the absorber plate, and also the thermal conductivity of welding between plate and sink, thermal conductivity of plate and the convection heat transfer coefficient of fluid are high.

Also in "Eq. 16", S is a part of solar radiation per plate area unit that is absorbed by absorber plate and is as follow:

$$S = \eta_0 \cdot I_T \quad (17)$$

where I_T is daily average hourly radiation entered to collector and η_0 is optical efficiency of collector and is calculated as follow:

$$\eta_0 = (\tau\alpha) = 1.01\tau\cdot\alpha \quad (18)$$

Also, I_T is calculated as follow:

$$I_T = I_bR_b + I_d\left[\frac{1+\cos\beta}{2}\right] + I\cdot\rho_{gr}\left[\frac{1-\cos\beta}{2}\right] \quad (19)$$

where I, I_b and I_d are solar radiation on horizontal surface, beam radiation and diffuse radiation, respectively.

Also, R_b is ratio of beam radiation on tilted surface to that on horizontal surface and is calculated as follow:

$$R_b = \frac{\cos(\varphi-\beta)\cos\delta\cos(\omega) + \sin(\varphi-\beta)\sin\delta}{\cos\varphi\cos\delta\cos\omega + \sin\varphi\sin\delta} \quad (20)$$

where φ is latitude of collector location, δ is declination angle and ω is hour angle.

Furthermore, U_L in "Eq. 16" is collector overall heat loss coefficient and is calculated as follow:

$$U_L = U_t + U_b + U_e \quad (21)$$

where U_t is top loss coefficient, U_b is back loss coefficient and U_e is edge loss coefficient. The top loss coefficient is calculated with "Eq. 22 to 26":

$$U_t = \left(\frac{N}{\frac{C}{T_{pm}}\left[\frac{T_{pm}-T_a}{N+f}\right]^e} + \frac{1}{h_w}\right)^{-1} + \frac{\sigma\left(T_{pm}+T_a\right)\left(T_{pm}^2+T_a^2\right)}{\frac{1}{\varepsilon_p + 0.0059N\cdot h_w} + \frac{2N+f-1+0.133\varepsilon_p}{\varepsilon_g} - N} \quad (22)$$

$$f=\left(1+0.089h_w-0.1166h_w\varepsilon_p\right)\left(1+0.07866N\right) \quad (23)$$

$$C=520\left(1-0.000051\beta^2\right) \quad (24)$$

$$e=0.430\left(1-\frac{100}{T_{pm}}\right) \quad (25)$$

$$h_w=2.8+3V_w \quad (26)$$

where N is number of glass covers, h_w is wind heat transfer coefficient, V_w is wind velocity and σ is Stefan Boltzmann constant.

Also the back loss coefficient is defined as follow:

$$U_b=\frac{k}{L} \quad (27)$$

Energy efficiency of collector is defined as follow:

$$\eta=\frac{\dot{m}_f\cdot c_P\left(T_{out}-T_{in}\right)-\mathrm{P}_{pump}}{I_T A_c+\mathrm{P}_{agitator}} \quad (28)$$

where $\mathrm{P}_{agitator}$ is power of agitator and in maximum value is about 15 W per every cylindrical obstacle [24]. Also P_{pump} is power of pump and is defined as follow:

$$\mathrm{P}_{pump}=\frac{P_{flow}}{\eta_{pump}\cdot\eta_{motor}} \quad (29)$$

where η_{pump} and η_{motor} are efficiency of pump and motor, respectively. Also, P_{flow} is dynamic pressure drop of fluid and is calculated as follow:

$$P_{flow}=\frac{\dot{m}_f\cdot\Delta P}{\rho} \quad (30)$$

d) Second law modeling

Exergy is the energy that is available to be used. The rate of exergy equation is defined as follow [25]:

$$\dot{E}_{in}-\dot{E}_{out}-\dot{E}_{loss}-\dot{E}_{des}=\dot{E}_S \quad (31)$$

where $\dot{E}_S$ is rate of storage exergy and with the assumption that the collector operates steady state it is equal to zero. $\dot{E}_{in}$ is rate of inlet exergy and includes rate of inlet exergy by inlet fluid to collector ($\dot{E}_{in,f}$) and rate of inlet exergy of absorbed solar radiation ($\dot{E}_{in,Q}$).

The rate of inlet exergy by inlet fluid to collector is defined as follow [25]:

$$\dot{E}_{in,f}=\dot{m}c_p\left(T_{in}-T_a-T_a\ln\left(\frac{T_{in}}{T_a}\right)\right)+\frac{\dot{m}\Delta P_{in}}{\rho} \quad (32)$$

where ΔP_{in} is difference between pressure of inlet fluid and ambient. The rate of inlet exergy of absorbed solar radiation is defined as follow [25]:

$$\dot{E}_{in,Q}=\eta_0 I_T A_c\left(1-\frac{T_a}{T_s}\right) \quad (33)$$

With the assumption that the sun is a black-body, the temperature of it is about 5777 K. According to influence of atmosphere on debilitation of solar radiation, T_s that is called seeming temperature of sun is about 0.75 of sun temperature and is equal to 4333 K approximately [26].

$\dot{E}_{out}$ is rate of outlet exergy and includes rate of outlet exergy by exiting fluid of collector ($\dot{E}_{out,f}$) [27].

$$\dot{E}_{out,f}=\dot{m}c_p\left(T_{out}-T_a\ln\left(\frac{T_{out}}{T_a}\right)\right)+\frac{\dot{m}\Delta P_{out}}{\rho} \quad (34)$$

where ΔP_{out} is difference between pressure of outlet fluid and ambient. $\dot{E}_{loss}$ is rate of exhausted exergy and includes rate of exhausted exergy from plate to ambient ($\dot{E}_{i,p}$) and exhausted optical exergy ($\dot{E}_{L,optical}$). The rate of exhausted exergy from plate to ambient is defined as follow [28]:

$$\dot{E}_{l,p}=U_L A_c\left(T_{pm}-T_a\right)\left(1-\frac{T_a}{T_{pm}}\right) \quad (35)$$

Because of optical properties of plate, a part of solar radiation does not absorb. Exhausted optical exergy of collector is calculated as follow [29]:

$$\dot{E}_{L,optical}=\frac{\left(1-\eta_0\right)\dot{E}_{in,r}}{\dot{E}_{in,r}}=1-\eta_0 \quad (36)$$

$\dot{E}_{des}$ is rate of destroyed exergy because of: temperature gradients between plate and sun ($\dot{E}_{d,\Delta Tp\text{-}s}$), temperature gradients between plate and fluid ($\dot{E}_{d,\Delta Tf}$) and pressure drop from inlet to outlet caused by viscosity of fluid, effects of walls of heat sink and also obstacles ($\dot{E}_{d,\Delta P}$). Theses parameters are calculated as follow, respectively [25]:

$$\dot{E}_{d,\Delta T_{p-s}}=\eta_0 I_T A_c T_a\left(\frac{1}{T_p}-\frac{1}{T_s}\right) \quad (37)$$

$$\dot{E}_{d,\Delta T_f}=\dot{m}c_p T_a\ln\left(\frac{T_{out}}{T_{in}}\right)-\dot{m}c_p T_a\frac{T_{out}-T_{in}}{T_p} \quad (38)$$

$$\dot{E}_{d,\Delta p}=\frac{\dot{m}\Delta p T_a\ln\left(\frac{T_{out}}{T_{in}}\right)}{\rho\left(T_{out}-T_{in}\right)} \quad (39)$$

Exergy efficiency of flat-plate solar collector is defined as rate of exergy increasing of fluid in collector to exergy of entering solar radiation to collector and it is calculated as follow [14]:

$$\psi = \frac{\dot{E}_{out,f} - \dot{E}_{in,f}}{I_T A_C \left(1 - \frac{T_a}{T_s}\right) + \mathrm{P}_{agitator}} \tag{40}$$

By combination "Eq. 27 to 36" the exergy efficiency of water-based flat-plate solar collector equipped with stationary and rotational obstacles is achieved.

e) Nanofluid

To calculate the thermophysical properties of nanofluid with spherical nanoparticle, the following equations are proposed. The effective density ρ_{nf} and specific heat $(c_P)_{nf}$ of the nanofluid at the reference temperature (T_a) are determined by the following equations [19]:

$$\rho_{nf} = (1\text{-}\phi) \cdot \rho_f + \phi \cdot \rho_{np} \tag{41}$$

$$(c_p)_{nf} = \frac{(1-\phi)(\rho c_p)_{nf} + \phi(\rho c_p)_{np}}{\rho_{nf}} \tag{42}$$

The Patel et al. [31, 32] model supposed to be a general tool to predict the thermal conductivity of CNT-Nanofluids. However, the model is not able to predict well at higher temperature of nanofluids.

$$k_{nf} = k_f \left(1 + \frac{k_{np} \cdot \phi \cdot d_f}{k_f \cdot (1-\phi) \cdot d_{np}}\right) \tag{43}$$

Boboo et al. [33] have proposed the viscosity of MWCNTs-Water correlation based on the experimental data valid up to 1.0% volume concentration.

$$\mu_{nf} = \mu_f \left(1 - 0.50437\phi + 1.744\phi^2\right) \tag{44}$$

f) Validation

A grid independence test was performed for the collector with three rotational obstacles (2 rad/s) at 12 p.m. to analyze the effects of grid sizes on the results. As shown in "Table 3", four sets of mesh are considered and by comparing all mesh configurations, the grid size of 3,728,623 nodes has been adopted to get an acceptable compromise between the computational time and the result accuracy. The computer software validation was done based on the geometry and boundary condition of Khorasanizadeh et al. [2].

In their study the properties of a flat-plate and pipe collector were investigated by empirical measurements. Based on "Fig. 2" it is clear that there is a remarkable coincidence between the empirical [2] and numerical results in the term of outlet temperature of fluid.The maximum error between empirical and numerical results in "Fig. 2" is about 12.5% at time of 9 a.m.

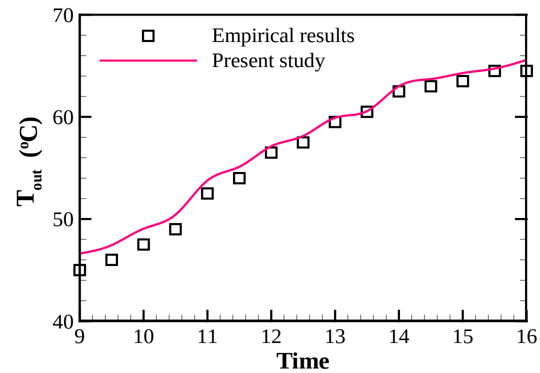

Figure 2: Comparison of the present results with the empirical results of Khorasanizadeh et al. [2], in term of outlet fluid temperature

Table 3: Grid independence test

Nodes	T_{out} (°C)	Error (%)
3,243,983	66.6782	2.22
3,599,007	70.5134	1.05
3,728,623	70.7811	0.02
3,954,131	70.7834	-

III. Results and Discussion

In this section firstly the collector exergy analysis is presented in two different conditions and then the optimization case is investigated.

a) Energy and Exergy Efficiencies

The total heat loss coefficient, mean temperature of absorber plate, collector outlet temperature and energy and exergy efficiencies of simple heat sink (SHS) collector, and heat sink collector equipped with mixers (HSWM) in different hours of day are reported in "Table 4 and 5", respectively. All these values are obtained based on numerical results and analytical correlation. It is realized that energy and exergy efficiencies of SHS collector increase about 30% and 60%, respectively, compared with the reference collector [2] owing to more wetted surface between plate and fluid and more time that it takes the fluid pass the route. Also, the energy efficiency of HSWM increases about 48% in comparison to the reference collector because of the induction of high disturbance and thin boundary layer in the channels equipped with obstacles, leading to higher temperature gradients from inlet to outlet.

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On the other side, the exergy efficiency of HSWM increases about 120% compared with the reference collector. Furthermore, the mean temperature of plate and outlet temperature of collector are increasing during the day incessantly because of the collector inlet temperature of fluid that is taken from the reservoir, is constantly increasing due to collector performance in a closed loop and heat saving in the reservoir. Also, in all conditions the inlet radiation flux rate increases from morning to the middle day hours and then decreases. The energy efficiency has the same trend. However, the reason for decreasing the energy efficiency after the afternoon hours is increasing the inlet fluid temperature and also increasing the absorber plate as the time passes that intensifies the losses.

It is clear from "Table 4 and 5" that the $U_L$change in different hours is significant so that in the condition of the collector with simple heat sink the relative difference of U_L at 10 a.m. is about 9% comparing to 16 p.m. This difference is more for other cases. This fact shows that the assumption of constant U_L that some researchers including [30] have considered is not logical and it is necessary to apply its changes in measurements.

Table 4: Results of simple heat sink collector (SHS)

Time	U_L (W/ m^2·K)	T_{pm} (°C)	T^{o}_{ut} (°C)	η (%)	ψ (%)
09:00	7.33	48.11	58.59	54.29	3.34
09:30	7.37	49.06	59.61	57.48	3.53
10:00	7.32	51.32	61.04	57.78	3.95
10:30	7.49	52.41	62.61	64.40	4.21
11:00	7.56	54.25	65.82	63.00	4.35
11:30	7.52	57.07	67.07	63.68	4.58
12:00	7.56	58.42	69.23	61.91	4.60
12:30	7.77	59.31	70.26	61.54	4.55
13:00	7.76	60.46	72.04	61.48	4.54
13:30	7.40	60.57	72.61	60.16	4.68
14:00	7.73	63.22	75.13	58.64	4.67
14:30	7.55	63.47	75.77	57.57	4.56
15:00	7.77	63.89	76.41	56.91	4.41
15:30	7.94	64.09	76.91	56.28	4.21
16:00	7.97	65.11	77.82	55.62	4.15

Table 5: Results of heat sink collector with mixers (HSWM)

Time	U_L (W/ m^2·K)	T_{pm} (°C)	T_{out} (°C)	η (%)	ψ (%)
09:00	7.15	47.08	53.23	64.04	6.34
09:30	7.21	48.02	54.24	67.83	6.88
10:00	7.08	50.27	55.51	68.18	7.77
10:30	7.25	51.33	57.49	75.98	8.34
11:00	7.32	53.21	59.61	74.34	8.71
11:30	7.35	56.00	62.29	76.44	9.21
12:00	7.34	57.32	63.71	74.28	9.98
12:30	7.51	58.14	64.40	73.81	8.75
13:00	7.56	59.32	65.87	73.78	8.62
13:30	7.30	59.52	66.01	72.78	8.93
14:00	7.60	62.13	68.82	69.14	8.87
14:30	7.38	62.42	68.99	69.08	8.81
15:00	7.59	62.64	69.11	66.58	8.51
15:30	7.67	63.01	69.25	65.84	8.01
16:00	7.69	64.12	69.34	65.63	7.88

b) *Using Nanofluid and Exergetic Optimization*

For all two conditions of using the collector, the lowest exergy and energy analysis is related to 9 a.m. either energy efficiency or exergy efficiency are dependent on the I_T and radiation angle. At 9 a.m. the I_T is less and also the angle between the direct sun radiation horizon and vertical to the collector surface is high. Hence the sun radiation absorption is less. In addition, the collector performance due to change in I_T and radiation angle and also the change in temperature of collector inlet water is always transient. These conditions are of high importance in the beginning hours of the day and these are factors of decreasing the efficiency. The effect of changing T_a, I_T, T_{in}, $\eta_0$and $\dot{m}$ parameters on exergy efficiency in different volume fraction of nanoparticles for the optimal condition (HSWM model) in this time was studied to optimize the collector exergically. Therefore, when different values were considered for one parameter, the value at 9 a.m. was assigned to other parameters. The results related to the influence of changing different parameters on the exergy analysis are shown in "Fig. 3 to 7".

In "Fig. 3" the exergy efficiency variation with sun radiation flux for different nanoparticles volume fractions is shown. In the radiation flux changing period, from 300 to 1200 W/m^2, for all conditions the increasing trend for exergy analysis is observed. By increasing the radiation of sun the temperature of collector outlet fluid increases and this increase leads to exergy efficiency increment.

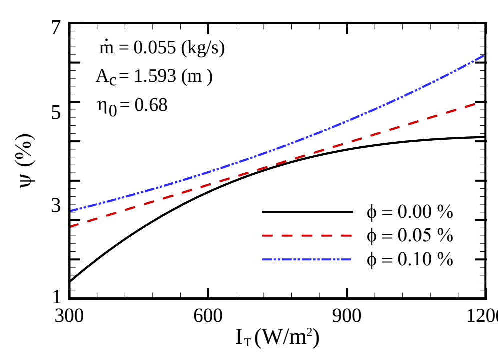

Figure 3: Variation of exergy efficiency of collector with solar radiation for nanifluid in different volume fractions

The exergy efficiency variation with collector inlet fluid temperature for different nanoparticles volume fractions has been demonstrated in "Fig. 4". For base fluid conditions, primarily the exergy efficiency increases until the temperature reaches 65 to 70°C and then it has decreasing trend. On one hand by T_{in} increase, the outlet temperature increases that leads to exergy efficiency increment. On the other hand, T_{in} increase means the fluid temperature inside the collector which raises the thermal loss. So there is one optimum T_{in} that for higher temperatures than it, the effect of exergy efficiency reduction due to higher thermal loss than its increase effect, that is because of fluid outlet temperature increment. But, for nanofluid conditions the exergy efficiency increases by increasing of inlet temperature.

The variation of exergy efficiency with ambient temperature for different nanoparticles volume fractions has been shown in "Fig. 5". For all three conditions of exergy efficiency it has decreasing trend by ambient temperature increase. In this figure the effect of using mixers in exergy efficiency increase due to the heat transfer rate between fluid and collector is perfectly clear.

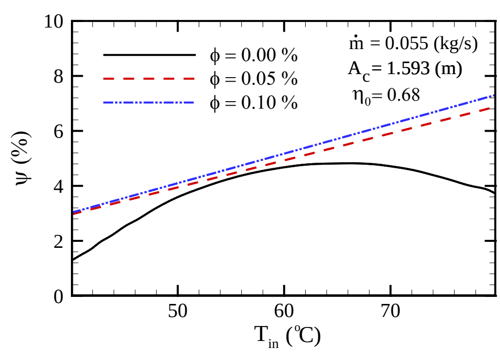

Figure 4: Variation of exergy efficiency of collector with temperature of inlet nanofluid in different volume fractions

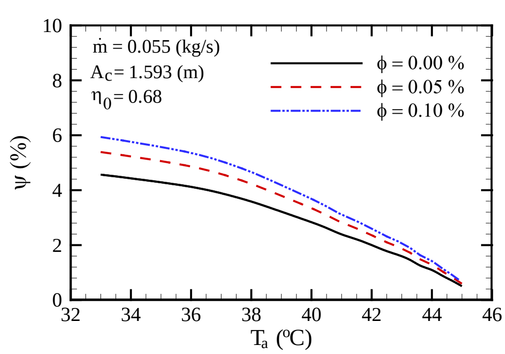

Figure 5: Variation of exergy efficiency of collector using nanofluid in different volume fractions with ambient temperature

In "Fig. 6" the influence of increasing optical efficiency on exergy efficiency for different nanoparticles volume fractions has been demonstrated. By optical efficiency increment for all three collector conditions, the radiation absorption by the absorber plate enhances and causes the fluid temperature inside the collector to increase and therefore the exergy efficiency increases.

In "Fig. 7" the effect of changing the fluid mass flow rate passing through the collector is shown in different nanoparticles volume fractions for mass flow rates from 0.0 to 0.1 kg/s. The applied mass flow rate for three conditions was about 0.055 kg/s. by referring to the results presented in "Fig. 7", it is understood that in the simulations conditions which parameters such as ambient temperature, inlet fluid temperature, optical efficiency, radiate flux and collector cross-section have the same values mentioned in "Table 4 and 5" that are related to 9:00 a.m. For the collector with base fluid, the optimum mass flow rate that causes the exergy efficiency to be maximum, should be ten times lower that means 0.005 kg/s. consequently the exergy efficiency is 5.3% instead of being 4%. Nevertheless, for the condition of using nanofluid the maximum exergy efficiency occurs in the highest mass flow rate of 0.1 kg/s.

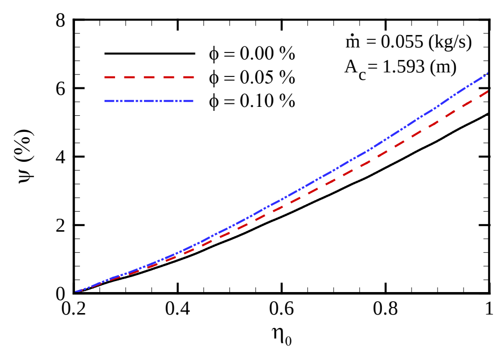

Figure 6: Variation of exergy efficiency of collector with optical efficiency in different nanofluid volume fractions

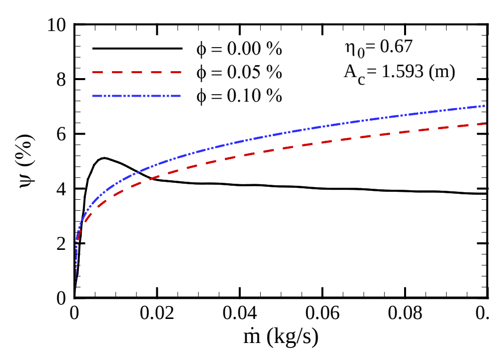

Figure 7: Variation of exergy efficiency of collector with mass flow rate of nanofluid in different volume fractions

IV. Conclusions

Specifying some values of mass flow rate and other parameters that the exergy efficiency get maximum due to them is difficult but in the concept of exergy efficiency, the effect of these parameters is clearer. In this study, the optimization of a solar collector in a closed circuit for three conditions in the viewpoint of exergy analysis by assuming that U_L is the only variable parameter and the fluid temperature is not equal to ambient temperature. The effect of using the mixers and nanofluid through fluid passage was studied and these results were obtained:

- Solar radiation flux increase and optical efficiency increase lead to exergy efficiency increase for all conditions.
- The exergy efficiency decrease by ambient temperature increase but by increasing the collector inlet fluid temperature the exergy efficiency increases to the certain temperature and then decreases. But, by using nanofluid the exergy efficiency always increases by increasing inlet temperature.
- For each special collector there is unique mass flow rate that the exergy efficiency gets maximum. For higher mass flow rates of base fluid, primarily the efficiency slightly decreases and then remains unchanged. But, by using nanofluid the maximum exergy efficiency occurs in the highest mass flow rate.
- Generally, using mixers and nanofluid enhance the exergy efficiency. In fact, while the trend of exergy efficiency variation with effective parameters is increasing, applying the obstaclesprecipitate the efficiency increment.in addition for the case that the trend of exergy efficiency variation with changing these parameters is decreasing, the decreasing trend gets slow.
- The collector performance in a closed circuit causes the collector inlet fluid temperature to increase constantly in the condition that the reservoir temperature increases due to not using the stored heat in it. The temperature increase leads to exergy efficiency increase to a certain point and then decreases this efficiency for higher values.

Nomenclature

A	area (mm^2)
B	slop (deg)
c_p	specific heat capacity (J/kg·K)
D	diameter (mm)
$\dot{E}$	energy rate (kW)
H	height (mm)
I	irradiation (W/m^2)
k	thermal Conductivity (W/m·K)
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
N	number of glass covers
n_o	numberof mixers
P	pressure (Pa)
P	power (W)
Pr	Prandtlnumber
R	ratio of beam radiation
S	solar radiation per plate area unit (W/m^2)
T	temperature (°C)
U	heat loss coefficient (W/m^2·K)
u	velocity (m/s)
V	velocity(m/s)
W	width (mm)
x	axial direction
Greek symbols	
□	absorption coefficient
□	slop (deg)
□	thickness (mm)
□	desalination angle (deg)
□	emissivity coefficient
□	energy efficiency
□□	optical efficiency
□	dynamic viscosity (N·s/m^2)
□	density (kg/m^3)
□	Stefan Boltzmann constant
□	transmissioncoefficient
□	volume fraction
□	latitude of collector location
□	exergetic efficiency
□	hour angle (deg)
Subscripts	
a	ambiaent
c	collector
e	exit of heat sink
f	base fluid
fo	first mixer
g	glass cover
i	inlet to heat sink
in	inlet
ins	insulator
m	mean
nf	nanofluid
np	nanoparticle
o	mixer (obstacle or agitator)
p	absorber plate
T	tilted surface
t	transient
w	wind

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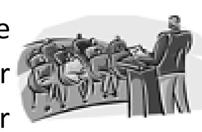

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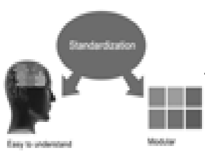

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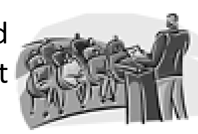

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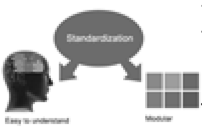

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References

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Key points to remember:

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A purpose of organizing a research paper is to let people to interpret your effort selectively. The journal requires the following sections, submitted in the order listed, each section to start on a new page.

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Approach:

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- Explain materials individually only if the study is so complex that it saves liberty this way.
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- If use of a definite type of tools.
- Materials may be reported in a part section or else they may be recognized along with your measures.

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- Report the method (not particulars of each process that engaged the same methodology)
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What to keep away from

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The page length of this segment is set by the sum and types of data to be reported. Carry on to be to the point, by means of statistics and tables, if suitable, to present consequences most efficiently.You must obviously differentiate material that would usually be incorporated in a study editorial from any unprocessed data or additional appendix matter that would not be available. In fact, such matter should not be submitted at all except requested by the instructor.

Content

- Sum up your conclusion in text and demonstrate them, if suitable, with figures and tables.
- In manuscript, explain each of your consequences, point the reader to remarks that are most appropriate.
- Present a background, such as by describing the question that was addressed by creation an exacting study.
- Explain results of control experiments and comprise remarks that are not accessible in a prescribed figure or table, if appropriate.
- Examine your data, then prepare the analyzed (transformed) data in the form of a figure (graph), table, or in manuscript form.

What to stay away from

- Do not discuss or infer your outcome, report surroundings information, or try to explain anything.
- Not at all, take in raw data or intermediate calculations in a research manuscript.
- Do not present the similar data more than once.
- Manuscript should complement any figures or tables, not duplicate the identical information.
- Never confuse figures with tables - there is a difference.

Approach

- As forever, use past tense when you submit to your results, and put the whole thing in a reasonable order.
- Put figures and tables, appropriately numbered, in order at the end of the report
- If you desire, you may place your figures and tables properly within the text of your results part.

Figures and tables

- If you put figures and tables at the end of the details, make certain that they are visibly distinguished from any attach appendix materials, such as raw facts
- Despite of position, each figure must be numbered one after the other and complete with subtitle
- In spite of position, each table must be titled, numbered one after the other and complete with heading
- All figure and table must be adequately complete that it could situate on its own, divide from text

Discussion:

The Discussion is expected the trickiest segment to write and describe. A lot of papers submitted for journal are discarded based on problems with the Discussion. There is no head of state for how long a argument should be. Position your understanding of the outcome visibly to lead the reviewer through your conclusions, and then finish the paper with a summing up of the implication of the study. The purpose here is to offer an understanding of your results and hold up for all of your conclusions, using facts from your research and generally accepted information, if suitable. The implication of result should be visibly described. Infer your data in the conversation in suitable depth. This means that when you clarify an observable fact you must explain mechanisms that may account for the observation. If your results vary from your prospect, make clear why that may have happened. If your results agree, then explain the theory that the proof supported. It is never suitable to just state that the data approved with prospect, and let it drop at that.

- Make a decision if each premise is supported, discarded, or if you cannot make a conclusion with assurance. Do not just dismiss a study or part of a study as "uncertain."
- Research papers are not acknowledged if the work is imperfect. Draw what conclusions you can based upon the results that you have, and take care of the study as a finished work
- You may propose future guidelines, such as how the experiment might be personalized to accomplish a new idea.
- Give details all of your remarks as much as possible, focus on mechanisms.
- Make a decision if the tentative design sufficiently addressed the theory, and whether or not it was correctly restricted.
- Try to present substitute explanations if sensible alternatives be present.
- One research will not counter an overall question, so maintain the large picture in mind, where do you go next? The best studies unlock new avenues of study. What questions remain?
- Recommendations for detailed papers will offer supplementary suggestions.

Approach:

- When you refer to information, differentiate data generated by your own studies from available information
- Submit to work done by specific persons (including you) in past tense.
- Submit to generally acknowledged facts and main beliefs in present tense.

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	A-B	**C-D**	**E-F**
Abstract	Clear and concise with appropriate content, Correct format. 200 words or below	Unclear summary and no specific data, Incorrect form Above 200 words	No specific data with ambiguous information Above 250 words
Introduction	Containing all background details with clear goal and appropriate details, flow specification, no grammar and spelling mistake, well organized sentence and paragraph, reference cited	Unclear and confusing data, appropriate format, grammar and spelling errors with unorganized matter	Out of place depth and content, hazy format
Methods and Procedures	Clear and to the point with well arranged paragraph, precision and accuracy of facts and figures, well organized subheads	Difficult to comprehend with embarrassed text, too much explanation but completed	Incorrect and unorganized structure with hazy meaning
Result	Well organized, Clear and specific, Correct units with precision, correct data, well structuring of paragraph, no grammar and spelling mistake	Complete and embarrassed text, difficult to comprehend	Irregular format with wrong facts and figures
Discussion	Well organized, meaningful specification, sound conclusion, logical and concise explanation, highly structured paragraph reference cited	Wordy, unclear conclusion, spurious	Conclusion is not cited, unorganized, difficult to comprehend
References	Complete and correct format, well organized	Beside the point, Incomplete	Wrong format and structuring

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