Performance Comparison between Three Models of Cavity in Parabolic Dish Collector

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DEDICATION

This Project is dedicated to:

My family whose support and understanding helped make this possible

Abstract

Performance Comparison between Three Models of Cavity in Parabolic Dish Collector

The experimental study and optimization of combined laminar natural convection and surface radiation heat transfer in three types solar cavity receiver is presented in this paper. The three type of cavity are: number one without plate cover; number two is with plate cover and number three is with plate and fin. Minimizing heat loss in cavity receivers is seen as an effective way to enhance the thermal performance and the use of plate fins has been proposed as a low cost means to minimize heat loss. Firstly, the influence of operating temperature, emissivity of the surface, orientation and the geometric parameters on the total heat loss from the receiver was investigated. It was observed that convective heat loss is largely affected by the angle of inclination of the receiver, the presence of fins and the number of fins in the receiver. As for the radiation heat loss it was observed that it is mainly influenced by the properties of the cavity receiver surface. Significant reduction in natural convection heat loss from the cavity receiver was accomplished by using the plate fins whereas radiation heat loss was marginally reduced. The result show that huge enhancing in heat transfer on cavity which equipped with cover plus fin compared with other type of cavity.

الملخص

مقارنة الاداء لثلاثة تصاميم من افران الاستقبال في جامع الحوض المكافيء

يتم تقديم الدراسة التجريبية وتحسين الانتقال الحراري الطبقي الطبيعي المشترك ونقل حرارة الإشعاع السطحي في ثلاثة أنواع من مستقبلات التجويف الشمسي في هذا البحث. الأنواع الثلاثة من التجويف هي: رقم واحد بدون غطاء لوحة ؛ الرقم الثاني مع غطاء اللوحة والثالث مع اللوحة والزعنفة. يُنظر إلى تقليل فقد الحرارة في مستقبلات التجويف على أنه طريقة فعالة لتحسين الأداء الحراري وقد تم اقتراح استخدام زعانف الصفائح كوسيلة منخفضة التكلفة لتقليل فقد الحرارة. أولاً ، تم فحص تأثير درجة حرارة التشغيل ، والانبعاثية للسطح ، والتوجيه والمعلمات الهندسية على إجمالي فقد الحرارة من المستقبل. لوحظ أن فقد الحرارة بالحمل الحراري يتأثر إلى حد كبير بزاوية ميل المستقبل ، ووجود الزعانف وعدد الزعانف في المستقبل. أما بالنسبة لفقدان الحرارة الإشعاعية فقد لوحظ أنه يتأثر بشكل أساسي بخصائص سطح مستقبل التجويف. تم تحقيق انخفاض كبير في فقد الحرارة بالحمل الحراري الطبيعي من مستقبل التجويف باستخدام زعانف اللوحة بينما تم تقليل فقد الحرارة الإشعاعية بشكل طفيف. أظهرت النتيجة زيادة كبيرة في نقل الحرارة في التجويف المجهز بغطاء زائد زعنفة مقارنة بالنوع الآخر من التجويف.

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LIST OF ABREVATIONS

CO_2	Carbon dioxide
3D	Three dimension
PDC	Parabolic dish collector
DNI	Direct normal irradiance
EES	Engineering equation solar
Nu	Nusselt number
Re	Reynolds number

LIST OF NOMENCLATURES

A_a : The available aperture (m²)

A_r : The receiver area (m²)

A_{ro} : The outside receiver area (m²)

A_{ri} : The inside receiver area (m²)

G_b : The direct beam solar direction

C : The ratio of the available aperture to the receiver area

c_p : The specific heat

T_r : Radiation temperature (K)

h_{air} : Heat convection coefficient between absorbed and ambient

V_{air} : Velocity of air (m/s)

h : Heat transfer coefficient

D : Cavity diameter (m)

d : Aperture diameter (m)

D_{ri} : Mean internal diameter (m)

$D_{ri,min}$: The minimum mean internal diameter (m)

L : Length of receiver tube (m)

$\dot{m}$: Fluid mass flow rate (kg/sec)

Nu : Nusselt number

Pr : Prandtl number

Q_s : The available solar energy

Q_{abs} : The rate of absorbed energy

Q_u : The rate of useful energy

Q_{loss} : The rate of thermal losses to the environment

Q_{rad} : The rate of thermal losses radiation

Q_{conv} : The rate of thermal losses convection

Re : Reynolds number

T_w : surface plate temperature (℃)

T_i Inlet temperature of working fluid (℃)

T_o : Outlet temperature of working fluid (℃)

T_{fm} : Mean fluid temperature

f_r : Friction factor

T_{am} : Ambient temperature

u : Velocity of the flow (m/s)

k : Thermal conductivity constant

ΔP : The pressure drop along the tube

Greek symbols

η_c : Collector thermal efficiency

η_{opt} : Optical efficiency of the collector

ε_r : Receiver emissivity

σ : Boltizman constant

η_{th} : Thermal efficiency of the collector

ρ : Fluid density (kg/m³)

CHAPTER 1

INTRODUCTION

1.1 Background

Renewable energy plays an important role in the current continuous increasing energy demand and at the same time too many emissions and greenhouse problems. This incessant request on the energy was one of the main reasons which contributed in expanding the utilization of the solar thermal energy (Sánchez et al. 2016). Moreover, there are many important problems related with the energy domain, as the increasing electricity demand, the high CO2 emissions and the fossil fuel depletion (Iodice et al. 2016). The use of renewable and alternative energy sources is a sustainable way for substituting the fossil fuel with cheap and abundant energy sources (Daabo et al. 2016a). Solar energy utilization is a basic weapon for facing the energy problems, giving efficient, clean and financially viable solutions (Bellos et al. 2016c).

Solar collectors are the devices which capture solar energy and transform it to useful heat, with satisfying efficiency. For low temperature levels up to 100 ℃, flat plate collectors are the most usual collector type (Bellos et al. 2016a). For medium temperature levels up to 200 ℃, evacuated tube collectors and low quality concentration collector are the most usual collectors (Kalogirou 2004). For high temperature levels, parabolic trough collector, Fresnel collectors and solar dish collectors are the most ideal solution for achieving satisfying results (Pavlović et al. 2016) as shown in fig. 1.1

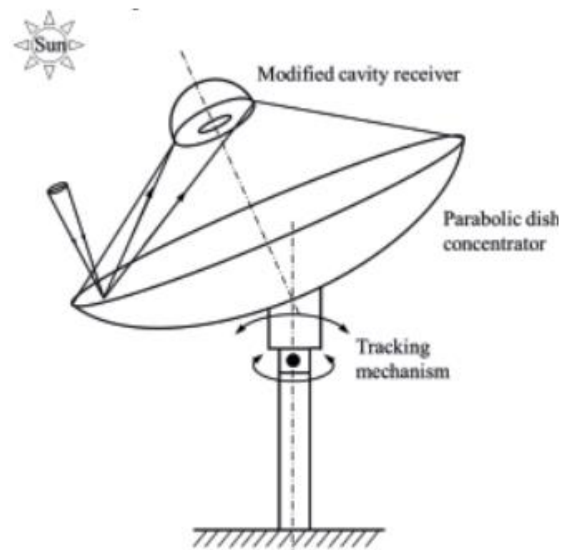


Fig. 1.1 Schematic of solar parabolic dish collector

1.2 Problem Statement

As a result of change in wind velocity which extremely effect on losses heat transfer from absorber tube of receiver cavity in parabolic dish collector, its appear need to reduce heat loses to enhance heat transfer between cavity space and working fluid inside absorber tube.

1.3 Objectives

The aim of this work is to select the optimum design of cavity using in parabolic dish collector the selection based on lowest heat losses from cavity system.

CHAPTER 2

LITERATURE REVIEW

2.1 Overview

Energy demand has been increasing with the increase in population and the rising standard of living. The use of unclean energy sources adversely affects the economy, the environment, and human health. Therefore, many countries prefer to use renewable energy sources, such as wind and solar energy, to fulfil their energy requirements without depending on fossil fuels (Shahin et al. 2016). Solar energy is one of the most promising renewable energy sources available at present. Among different types of solar collectors, the parabolic dish collector (PDC) has been receiving considerable attention for a wide range of applications, particularly in hot water production (Kalogirou 1992) and industrial steam generation (Thomas 1996). Therefore the study will focus on experimental test to evaluate the best cavity based on calculate the lowest heat loss.

2.2 PDC Receiver Applications

Solar dish collectors are a reliable solution for operation in medium and high temperature levels. (Abid et al. 2015) compared a solar dish collector with a parabolic trough collector and the final results proved that the dish technology performs better because of the higher concentration ratio which is fully connected with lower thermal losses and higher thermal efficiency. The solar dish concentrators have been used in a great variety of applications for heat and electricity production. The use of solar dish concentrators in gas turbine systems has been intensively studied during the last years. (Mohammadi and Mehrpooya 2016) and (Daabo et al. 2017) investigated and optimized an integrated micro gas turbine with solar dish collectors to be used between the gas preheater and the combustion chamber. (Loni et al. 2016) examined the use of a solar dish collector with cavity receiver in an organic Rankine cycle. The followed methodology proved that there is optimum concentration ratio which maximizes the work output and the authors proposed the conduction of detailed optical analysis for determining the optimum receiver dimensions in every design. Also, the use of hybrid solar dish collectors in desalination system has been conducted in literature (Omara and Eltawil 2013). Likewise, the use of pure solar energy to desalinate sea salt water for a domestic application was conducted by (Prado et al. 2016). The conjugation of Sterling heat engines with solar dish plates is a promising technology for producing electricity with high performance but also with high investment cost (Xiao et al. 2017). The main purpose of the recent studies regarding this filed is to reduce the cost of the system and to design collectors with higher optical performance. (Li et al. 2011) utilized a Monte-Carlo ray tracing method for determining the heat flux distribution over the receiver. The results proved that the most uniform heat flux profile can be achieved with a shallow semi-ellipsoidal receiver. The design of the solar dish collector is not well-established, with numerous configurations to be studied and to be suggested. Two main parts of the thermal system are analyzed; the reflector and the receiver. The dish reflector size can be varying a lot, from small dishes (for example 0.5 m^2 or 1 m^2) up to huge systems. (Lovegrove et al. 2011) investigated a

500 m^2 solar dish concentrator with 380 identical spherical mirror panels. However, this scale is not always achievable. (Cohen and Grossman 2016) studied a great stationary reflector which can be manufactured with low cost. This reflector was similar to a spherical bowl and the absorber was a cylindrical coil painted with flat-black. The absorber is located in a glass cover while vacuum conditions exist there for minimization of the rate of thermal losses. However, the reflector was fixed on specific position which obviously would not be accounted as an efficient system. Other studies have been focused on the receiver investigation in order to compare various possible candidates. (Daabo et al. 2016b) examined, optically, three cavity receiver geometries; cylinder, cone and sphere without inserting helical tube and then in the next study (Daabo et al. 2016c), a helical tube was used inside in order to capture utilize the solar energy with an efficient way. According to the final results, the conical shape is the best choice among the examined cases. Besides, they proved that the optimum reflector geometry is depended on the selected receiver; an interesting result which is useful in the design of innovative solar dish collectors. A parabolic dish concentrator and cavity receiver with quartz glass cover system were presented in the study done by (Cui et al. 2013). A 2-D simulation model for combined natural convection and surface radiation has been developed. The results of simulation showed that compared with the uncovered receiver, the quartz glass cover largely reduces the natural convection and surface radiation heat losses of the cavity receiver. The total heat flux of the covered receiver at an inclination was only about 36% of that for uncovered receiver. However, neither 3D analysis on the coupled heat transfer process of the cavity receiver nor the heat flux uniformity were conducted. A numerical study on the phase change materials for a vertical cylindrical receiver was

examined by (Tao et al. 2013). The feasibility from the techno-economic view point of a 5MWe solar parabolic dish collector field at different areas in India was analysed by (Reddy and Veershetty 2013). Different parameters like percentage of the shadow, spacing between dish collectors and energy yield were numerically investigated. The result showed that there was shadow profile changing with the latitude of (8–35_N). As for the location, their results showed that there are some attractive regions; Direct Normal Irradiance DNI is more than 5 kWh/m^2 day, in the investigated locations which can be used for power generation using the solar parabolic dish. The analysis of a hybrid cooling and heating integrated with Stirling engine and absorption chiller has been proposed and analyzed by (Mehrpooya et al. 2017). (Przenzak et al. 2016) investigated a solar dish collector, with two optical elements and a curved radiation absorber. This collector is designed for operation in high temperature levels and there is a proper design for achieving this goal. The authors of this work performed a parametric investigation in order to determine the optimum receiver location and the most suitable mass flow rate. (Reddy and Kumar 2009) examined a modified cavity receiver of a solar dish collector and they specifically focused on the natural convection losses of the presented collector. Furthermore, they, experimentally, includes a detailed optical analysis of this receiver in (Reddy et al. 2015). Finally, the effect of gravity load on both; the mirror shape and the quality of concentrator for parabolic trough was examined by (Meiser et al. 2017). With the aid of finite element technique and of some specific lab tests, different collector angles were studied. According to their results, the optimum collector angle, with respect to the mirror shape, was 0° (zenith). While, as it is presented, many configurations of receivers and absorbers have been investigated in the literature, very few studies experimentally investigated and validated their works are found in the literature. In this study, a lightweight solar dish concentrator which is consisted of 11 curvilinear trapezoidal reflective petals, coupled with a spiral absorber inside housing is manufactured and experimentally examined. This system is innovative and its design has been presented in an older preliminary study done by (Pavlović et al. 2016). In this study, experimental results of this collector are presented, as well as the results of a developed numerical model in Engineering Equation Solver (EES) (Klein 2015) are presented.

2.3 Receiver Cavity

A low cost solar collector with a dish reflector and spiral absorber is examined in this work. This collector is investigated experimentally and numerically with a developed thermal model in the Engineering Equation Solver (EES). Numerical simulations are performed by the commercial software OptisWorks. The results show that an average flux value of about 2.6 x 105 W/m^2 was absorbed by the helical conical shape with an aperture area of 0.01606 m^2 . The final results show that the thermal performance is about 34%, due to the high rate of thermal losses (Sasa at el.2017) as shown in figure 2.1.

Figure 2.1 solar thermal receiver.

a 3-D numerical investigation is carried out to estimate heat losses from solar parabolic dish with modified cavity receiver used for three different steam generation conditions viz. sub-cooled, saturated, superheated steam. The effect of inclination of the receiver, operating temperature, emissivity of the cavity cover, insulation thickness on the total heat loss from the modified cavity receiver has been investigated. The present model can be used to predict total heat losses from the modified cavity receiver under all conditions more accurately (Vikram and Reddy 2014) as shown in figure 2.2.

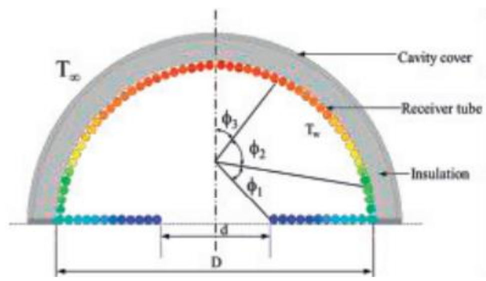


Figure 2.2. Boundary conditions of modified cavity receiver (θ = 90°) for three configurations (sub-cooled, saturated, superheated)

The numerical study of combined laminar natural convection and surface radiation heat transfer in a modified cavity receiver of solar parabolic dish collector is presented in this paper. A two-dimensional simulation model for combined natural convection and surface radiation is developed. convection and radiation heat losses are found respectively 52% and 71.34% of the total heat loss at 0° inclination and 40.72% and 59.28% at 90°.inclination for the modified cavity receiver with an area ratio of 8 and 400 ℃.(Reddy and Kumar 2008) as shown in figure 2.3.

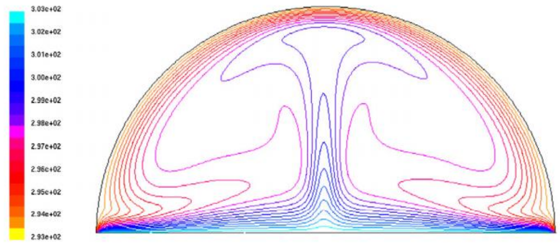


Figure 2.3. Temperature contours of hemispherical enclosure.

This study details the numerical modeling and optimization of natural convection heat suppression in a solar cavity receiver with plate fins. The use of plate fins attached to the inner aperture surface is presented as a possible low cost means of suppressing natural convection heat loss in a cavity receiver. The results indicate that significant reduction on the natural convection heat loss can be achieved from cavity receivers by using plate fins, and an optimal, Reduction of up to a maximum of 20% at 0° receiver inclination was observed.(L.C. Ngo at el 2015). In this work a theoretical and experimental study of heat transfer by natural convection and thermal radiation on a solar open cubic cavity-type receiver is presented. The theoretical study consists on solving the laminar natural convection and the surface thermal radiation on a square open cavity at one end. The overall continuity, momentum, and energy equations in primitive variables are solved numerically by using the finite-volume method and the SIMPLEC algorithm this work a theoretical and experimental study of heat transfer by natural convection and thermal radiation on a solar open cubic cavity-type receiver is presented. The theoretical study consists on solving the laminar natural convection and the surface thermal radiation on a square open cavity at one end. The overall continuity, momentum, and energy equations in primitive variables are solved numerically by using the finite-volume method and the

SIMPLEC algorithm. Experimental results include air temperature measurements inside the receiver. These results are compared with theoretically obtained air temperatures, and the average deviation between both results is around 3.0%, when using the model with variable thermo physical properties, and is around 5.4% when using the Boussinesq approximation.(J.F.HINOJOSA at el. 2015) investigated using 3-D numerical model. The effects of various parameters such as diameter ratio (d/D), angle of inclination (q), operating temperature (T), insulation thickness (t) and emissivity (ε) of the cavity cover on the heat losses from the modified cavity receiver are investigated. The variable boundary conditions are considered for modified cavity receiver to produce sub-cooled hot water, saturated steam and superheated steam. The convective heat losses are greatly influenced by receiver inclination whereas the radiation heat losses are influenced by the cavity cover emissivity. The diameter ratio also plays a major role in heat losses from the cavity receiver. The total heat loss is estimated at 522 W.(Vikram and reddy 2015) as shown in figure 2.4.

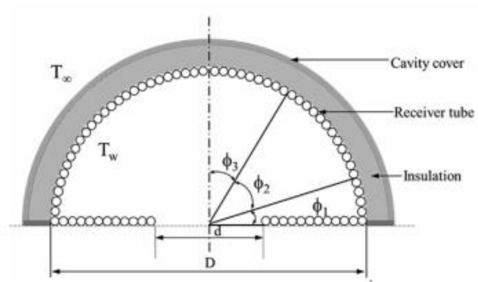


Figure 2.4. boundary conditions: temperature profiles (Tw) for sub-cooled, saturated and superheated conditions.

In this paper, a numerical investigation is performed to study the natural convective heat loss from three types of receivers for a fuzzy focal solar dish concentrator, namely cavity receiver, semi-cavity receiver and modified cavity receiver. The natural convection heat loss from the receivers is estimated by varying the inclination from 0° (cavity aperture facing sideways) to 90° (cavity aperture facing down). The orientation and geometry of the receiver strongly affect the natural convection heat loss. A comparative study is performed to predict the natural convection heat loss from the cavity, semi-cavity and modified cavity receivers. The convection heat loss is high at 0° and decreases monotonically with increase in angle up to 90° in all three cases. The convection heat losses at 0° and 90° inclination of the modified cavity receiver are 26.03% and 25.42% of the convection heat loss of the cavity receiver, respectively. The influence of area ratio (Aw/A1) on the convective heat loss is investigated for the modified cavity receiver, and an optimum Aw/A1 of 8 is found for minimum natural convection (Reddy and Kumar 2008) as shown in figure 2.5.

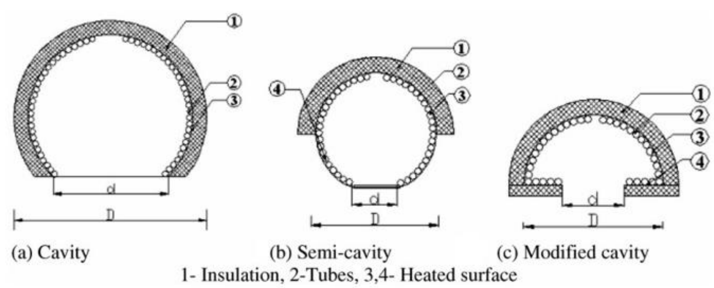


Figure 2.5. Enlarged view of different types of receivers.

The numerical study and optimization of combined laminar natural convection and surface radiation heat transfer in solar cavity receiver with plate fins is presented in this paper. Minimizing heat loss in cavity receivers is seen as an effective way to enhance the thermal performance and the use of plate fins has been proposed as a low cost means to minimize heat loosed. It was observed that convective heat loss is largely affected by the angle of inclination of the receiver, the presence of fins and the number of fins in the receiver. As for the radiation heat loss it was observed that it is mainly influenced by the properties of the cavity receiver surface. Lastly, the overall thermal efficiency of the receiver was presented at different operating temperatures. The overall cavity efficiency marginally increased by approximately 2% with the insertion of fin plates although the convective heat loss was suppressed by about 20%. This is due to the fact that radiation heat loss dominates at high operating temperatures compared to convective heat loss (L.C. Ngo at el. 2015) as shown in figure 2.6.

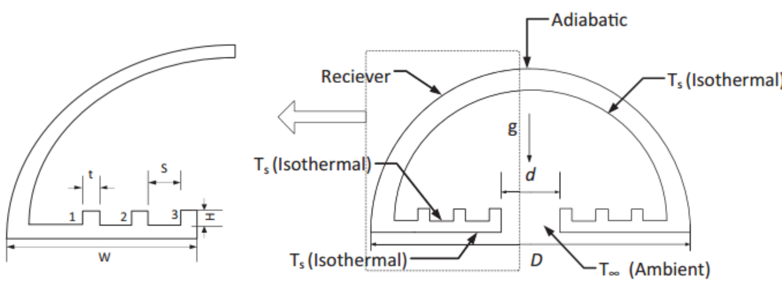


Figure 2.6. Schematic of modified cavity receiver with plate fins.

CHAPTER 3

RESEARCH METHODOLOGY

3.1 . Model description

Fig. 1 shows the schematics of the cavity receiver. The cavity receiver is usually fitted at the focus of the parabolic dish concentrator to capture concentrated sunlight from the dish. Both cavity receivers are made of copper tubing with an opening aperture diameter (d) and cavity diameter (D). Circular plate fins ranging between 1 mm and 6 mm have been installed in the proposed cavity receiver to suppress combined laminar natural convection and radiation heat loss. The cavity receiver diameter and the aperture are given as 180 mm and 100 mm respectively. In order to get the geometry of the cavity receiver, the pipes (made of copper) are wound spirally. An opaque insulating material is used to cover the outer surface of the cavity receiver so as to reduce conduction heat loss. Some assumptions are made for designing the cavity receiver:

1) there is maximum and uniform solar flux distribution inside the cavity receiver
2) the surfaces are considered smooth and uniform for the original modified cavity receiver.
3) the plate fins are made of copper and are installed on the smooth inner aperture surface of the original modified cavity receiver;
4) the temperature of copper tube surface is equivalent to temperature of air flowing in it. However, the copper tubes were not included in the simulation;
5) effects of wind are not considered (i.e. no wind case).

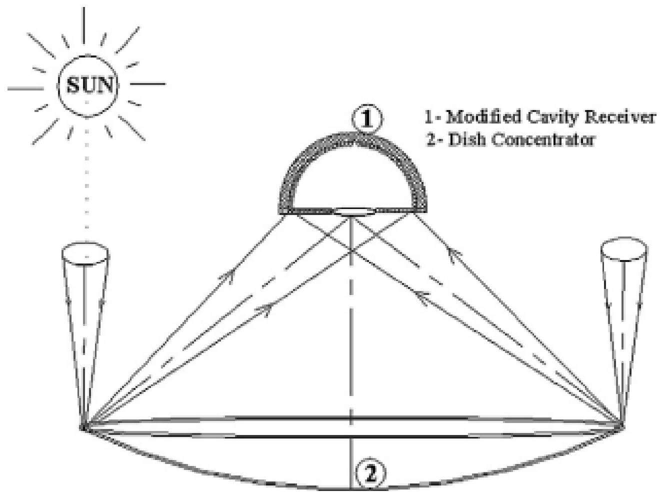


Fig 3.1 Solar Parabolic dish collector

3.2 Mathematical Calculation

The concentrated collectors with high concentrating ratios, as the examined dish reflector, utilize only the direct beam solar radiation (G_b) and the available solar energy is calculated as the product of the effective dish aperture (A_a) and the beam radiation:

$$Q_s = A_a \times G_b \tag{3.1}$$

The concentration ratio of the collector (C) is the ratio of the available aperture (A_a) to the receiver area (A_r), as Eq. (2) shows:

$$C = \frac{A_a}{A_r} \tag{3.2}$$

The rate of absorbed energy from the receiver (Q_{abs}) can be calculated using the optical efficiency of the collector (η_{opt}): ,

$$Q_{abs} = \eta_{opt} \times Q_s \tag{3.3}$$

3.3 Thermal analysis

The developed thermal analysis model is based on the energy balance in the receiver. The rate of absorbed solar radiation is separated to the rate of useful energy (Q_u) and to the rate of thermal losses to the environment (Q_{loss}), as Eq. shows:

$$Q_{abs} = Q_s = Q_u + Q_{loss} \tag{3.4}$$

The useful heat output can be calculated by the energy balance in the fluid volume, according to Eq.:

$$Q_u = m\,.\,c_p(T_{out} - T_{in}) \tag{3.5}$$

The rate of thermal losses is separated to radiation (Q_{rad}) and convection (Q_{conv}) losses. Equations (6); (7) give the formulas for estimating these quantities:

$$Q_{rad} = A_{ro}\,.\,\varepsilon_r\,.\;\sigma(T_r^4 - T_{am}^4) \tag{3.6}$$

$$Q_{conv} = A_{ro}\,.\,h_{air}\,(T_r - T_{am}) \tag{3.7}$$

The heat convection coefficient between absorber and ambient can be calculated by the following Eq. (8), as proposed by (Duffie and Beckman 2013):

$$h_{air} = 2.8 + 3\,V_{air} \tag{3.8}$$

The thermal efficiency of the collector (η_{th}) is calculated as the ratio of the useful energy output to the available solar radiation:

$$\eta_{th} = \frac{Q_u}{Q_s} \tag{3.9}$$

Heat transfer in the flow In the present section the equations related to the heat transfer inside the flow are presented. The rate of useful energy that the fluid gains can be calculated as:

$$Q_u = A_{ri} \,.\, h\left(T_r - T_{fm}\right) \tag{3.10}$$

The mean fluid temperature can be calculated according to Eq. (11). This temperature has also been used for the determination of the thermal properties of the working fluids. ,

$$T_{fm} = \frac{T_{out}+T_{in}}{2} \tag{3.11}$$

The heat transfer coefficient for the examined case is calculated according to Eq. (12) (Zhu et 241 al. 2017):

$$Nu = \frac{\left(\frac{f_r}{8}\right).Re\,.Pr}{1+12.8\,(Pr^{0.68}-1)\sqrt{\left(\frac{f_r}{8}\right)}} \tag{3.12}$$

The friction factor (f_r) has to be determined by a complex equation because the tube is corrugated in the present study. The following equation is suitable for the examined case (Đorđević et al. 2016):

$$f_r = 0.316\,Re^{-0.25} + 0.41\left(\frac{D_{ri,min}}{D_{ri}}\right)^{0.9} \tag{3.13}$$

It is important to state that the mean internal diameter (D_{ri}) is the diameter that is used for Reynolds definition. Equations (14), (15); (16) present the characteristic numbers of Reynolds, Prandtl and Nusselt respectively:

$$Re = \frac{4\, m}{\pi\,.\; D_{ri}\,.\; \mu} \tag{3.14}$$

$$Pr = \frac{\mu\,.c_p}{k} \tag{3.15}$$

$$Nu = \frac{h\,.\,D_{ri}}{k} \tag{3.16}$$

The last important parameter for this study is the pressure drop along the tube, a parameter that is calculated by using the friction factor:

$$\Delta P = f_r\,.\frac{L}{D_{ri}}\,.\left(\frac{1}{2}\,.\,\rho\,.\; u^2\right) \tag{3.17}$$

The velocity of the flow (u) is calculated from the mass flow rate, according to Eq. (18):

$$u = \frac{m}{\left(\frac{\pi}{4}.\; {D_{ri}}^2\right).\rho} \tag{3.18}$$

3.4 Field Measurement

In experimental test several factors may impact on the efficiency of PDC cavity system that has been considered during the test. The factors (ambient temperature, inlet and outlet temperature) can be measured by portable data logger. There are several instruments and accessories device that should be used in experimental test to get reliable results and validation, the equipment that used together to collect data are:

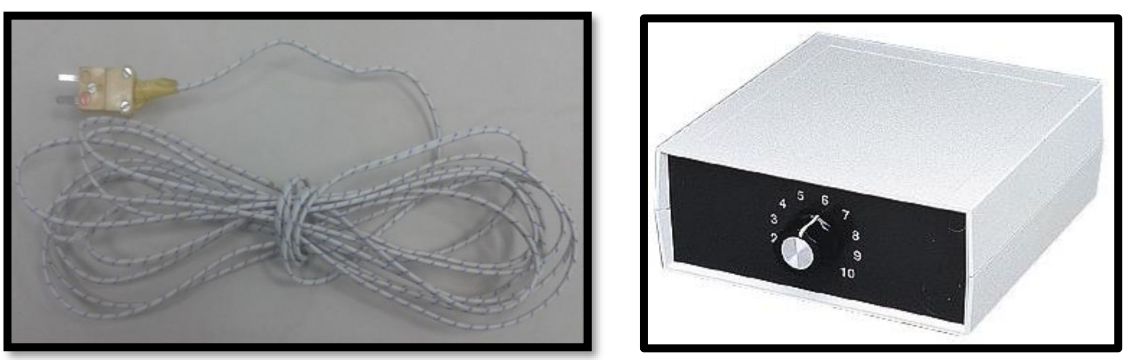
3.4.1 Data logger

Data logger is the instrument to measure temperature directly at any point in the system. This device is type K with temperature range from -50°C to 1300°C and accuracy $\pm$0.5% or $\pm$1°C. This type of data logger can be connected with multi-channal switchbox for measuring temperature in multi-point at the same time. Data logger that used in this study is Digi-Sense 91000-00 as shown in Figure 3.2.

Figure 3.2. Digi-Sense 91000-00 Type K Data Logger (Thermometer 2015)

3.4.2 Thermocouple Wires

In this study, type K thermocouple wires with high temperature insulation (see Figure 3.3 (a)) are used for measuring temperatur in multipint of the receiver heat exchanger. The thermocouple wires are connected to data logger and 10 channal swithbox which is transfer temperature to temperature reader. Type K Ten-Channals Switch-Box (see Figure 3.3 (b)) is used to collecte temperature from several point at the systerm and transfer it to temperature reader (data logger).



(a) **(b)**

Figure 3.3. (a) Type K thermocouple wire geometry, (b) Ten-Channals Switch-Box (Sellector Cole-Parmer 2015)

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 System Description

At the beginning of the work, we have a device consisting of a dish and a cavity receiver. Several reflective mirrors were placed on the surface of the dish to gain the largest possible amount of solar radiation as shown in Fig 4.1.

4.1 Parabolic Dish Collector with Mirror

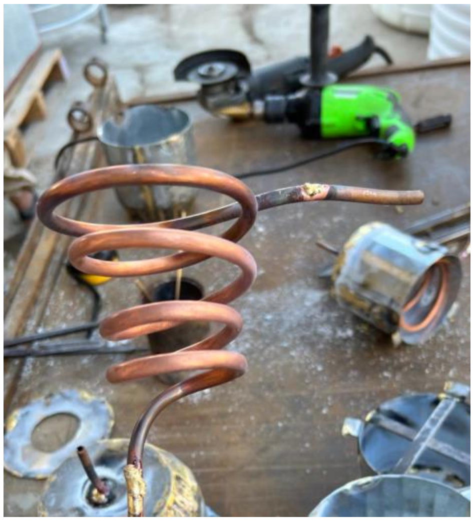
Three cavity receivers were manufactured in different shapes to choose the best shape for the high temperature of water (cavity with no cover -cavity with cover - cavity with cover +fins) as shown in Fig 4.2.

Fig 4.2. Receiver Cavity

The dish is attached to the wall at a certain angle, and we attach the cavity receiver to the dish in succession for three consecutive days. Then we run the device through the flow of water inside the spiral tube made of copper inside the cavity receiver.

The cavity consists of insulation material sandwiched between two stainless steel plate as shown in Fig 4.3 (a). Copper tube with 3mm diameter and 0.5m length were coiled with 4 times, and then the coile placed inside cavity as shown in Fig 4.3 (b)

(a)



(b)

(c)

Fig 4.3 Cavity Manufacture

Three Cavity were `fabricated, one live with no cover, the second covered with stainless steel plate with axial hole to access the solar radiation. The third one is covered with stainless steel covered equipped with one fine. The aim of using cover is to prevent circulation of outside air in the cavity which will led to increase the heat loss as shown in Fig 4.4.

(a) **(b)** **(C)**

Fig. 4.4. Cavity with cover

4.2 Experimental setup

The three cavities were fixed on three parabolic dish collector as shown in Fig. 4.5.

Fig 4.5 The parabolic System

Three thermocouples are placed in copper tube to recorded the temperature of water in inlet and outlet, it also recorded the middle wall temperature of copper tube. Each cavity equipped with three thermocouples which connect with 10-chanel device, and then connected with data logger as shown in Fig. 4.6.

Fig 4.6 The parabolic System

On the first day, data collections take place at outside air velocity of 4m/s that measured by portable air flow rate device as shown in fig 4.7. and 0.005 kg/s of water flow rate. The first data was taken at 3 minutes after starting the system operate, the inlet water temperature were 25. The temperature at inlet and outlet of water flow as well as copper tube wall temperature were recorder each 0.5 min. as shown in table 4.1 the calculation achieved at peak value of water outlet temperature. The result show that the cavity with cover plus fin present low heat loss than other type of cavity as shown in 4.8. The results show in the first test that the highest temperature of the water when reading No. 28 in the Cavity receiver system with cover plus fins, which was about 199℃ see Table 4.1.

Fig 4.7 portable air flow rate device

Table 4.1 Data collection in first day at air velocity $(u = 4\ m/s)$ & water flow rate $(\dot{m} = 0.005\ kg/s)$

No.	Time (sec)	Cavity with no cover			Cavity + cover			Cavity +cover +fins		
		T_{in}	T_{out}	T_{wall}	T_{in}	T_{out}	T_{wall}	T_{in}	T_{out}	T_{wall}
1	3	25	37	32	25	48	37	25	65	50
2	3.5	25	39	33	25	51	38.5	25	70	52.5
3	4	25	41	34	25	54	42.5	25	76	55.5
4	4.5	25	47	37	25	60	45.5	25	86	60.5
5	5	25	51	39	25	65	48	25	95	65
6	5.5	25	55	42	25	70	50.5	25	102	68.5
7	6	25	59	44	25	75	53	25	107	81
8	6.5	25	63	46	25	81	56	25	115	85
9	7	25	70	49.5	25	87	59	25	124	89.5
10	7.5	25	77	53	25	94	62.5	25	134	94.5

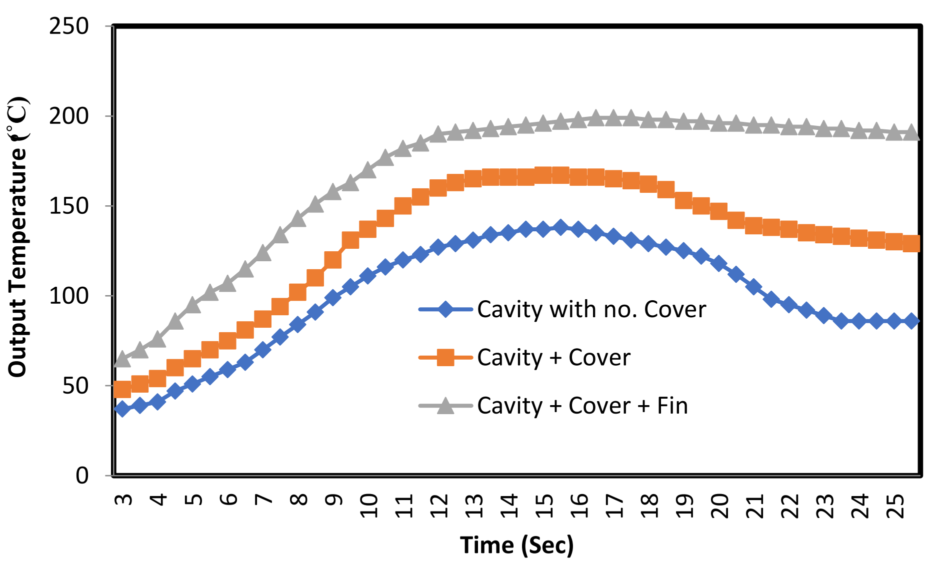


Fig. 4.8 Output Temperature against Time at first day

On the second day, data collections take place at outside air velocity of 9.6m/s and 0.005 kg/s of water flow rate. The first data was taken at 3 minutes after starting the system operate, the inlet water temperature were 25. The temperature at inlet and outlet of water flow as well as copper tube wall temperature were recorder each 0.5 min. as shown in table 4.2 the calculation achieved at peak value of water outlet temperature. The result show that the cavity with cover plus fin present low heat loss than other type of cavity as shown in 4.9.

Table 4.2 Data collection in second day at air velocity $(u = 9.6\ m/s)$ & water flow rate $(\dot{m} = 0.005\ kg/s)$

No.	Time (sec)	Cavity with no cover			Cavity + cover			Cavity +cover +fins		
		T_{in}	T_{out}	T_{wall}	T_{in}	T_{out}	T_{wall}	T_{in}	T_{out}	T_{wall}
1	3	25	33	30	25	47	36.5	25	64	49.5
2	3.5	25	35	31	25	47	36.5	25	66	50.5
3	4	25	37	32	25	50	40.5	25	72	53.5
4	4.5	25	43	35	25	56	43.5	25	82	58.5
5	5	25	47	37	25	61	46	25	91	63
6	5.5	25	48	38.5	25	66	48.5	25	98	66.5
7	6	25	52	40.5	25	71	51	25	103	79
8	6.5	25	56	42.5	25	77	54	25	111	83
9	7	25	63	46	25	80	55.5	25	117	86
10	7.5	25	70	49.5	25	87	59	25	127	91

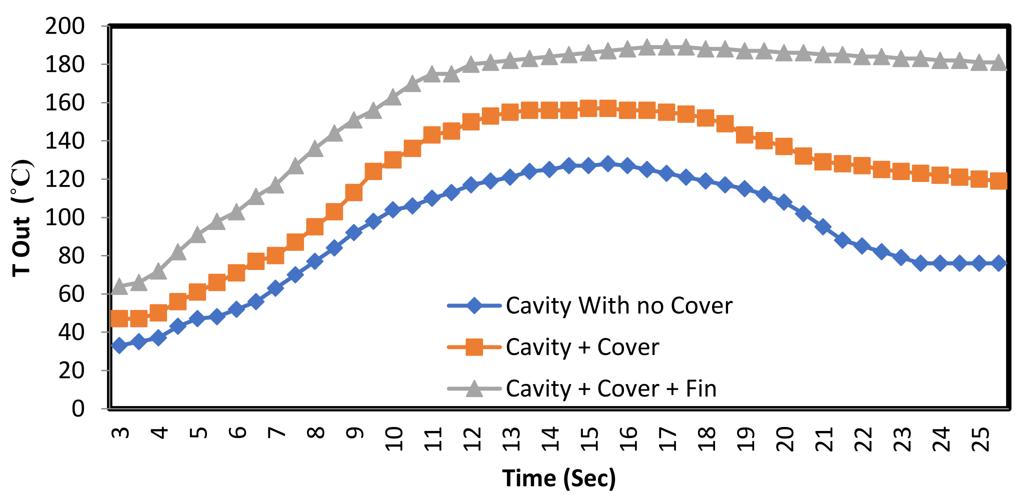


Fig 4.9 Output Temperature against Time at second day

On the second day, data collections take place at outside air velocity of 16.3m/s and 0.005 kg/s of water flow rate. The first data was taken at 3 minutes after starting the system operate, the inlet water temperature were 25℃. The temperature at inlet and outlet of water flow as well as copper tube wall temperature were recorder each 0.5 min. as shown in table 4.3 the calculation achieved at peak value of water outlet temperature. The result show that the cavity with cover plus fin present low heat loss than other type of cavity as shown in 4.10.

Table 4.3 Data collection in second day at air velocity $(u = 16.3\ m/s)$ & water flow rate $(\dot{m} = 0.005\ kg/s)$

No.	Time (sec)	Cavity with no cover			Cavity + cover			Cavity +cover +fins		
		T_{in}	T_{out}	T_{wall}	T_{in}	T_{out}	T_{wall}	T_{in}	T_{out}	T_{wall}
1	3	25	28	27	25	42	34.5	25	59	43
2	3.5	25	30	28	25	42	34.5	25	61	44
3	4	25	32	29	25	45	36	25	67	47
4	4.5	25	38	32	25	51	42	25	77	52
5	5	25	42	34	25	56	44.5	25	86	59.5
6	5.5	25	43	34.5	25	61	47	25	93	63
7	6	25	43	34.5	25	62	47.5	25	94	63.5
8	6.5	25	47	36.5	25	68	50.5	25	102	67.5
9	7	25	54	40	25	71	52	25	108	70.5
10	7.5	25	61	43	25	78	55.5	25	118	75.5

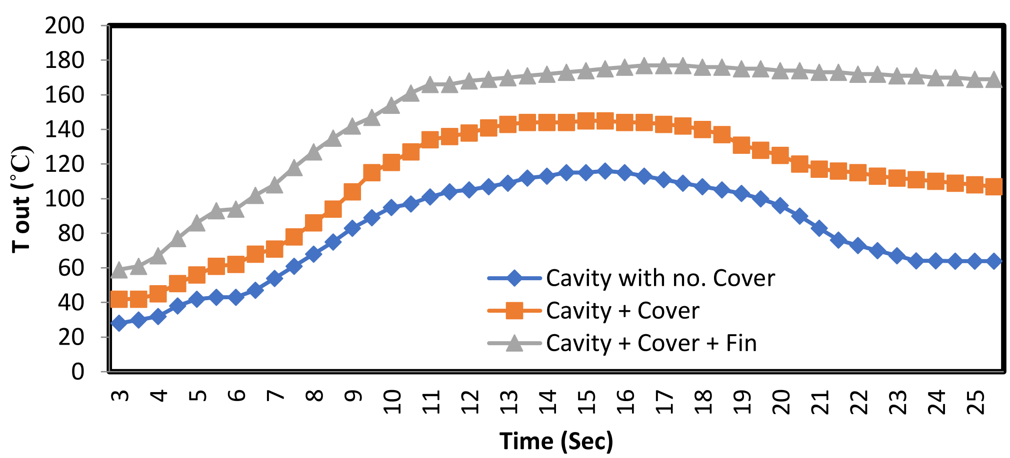


Fig 4.10 Output Temperature against Time at third day

Finaly, by compareson between results of three types of cavity in three different condition days, the heat loss decreas at cavity which equipped with cover and fin less than tis with cover only and cavity with no cover respectively, as shown in Fig 4.11.

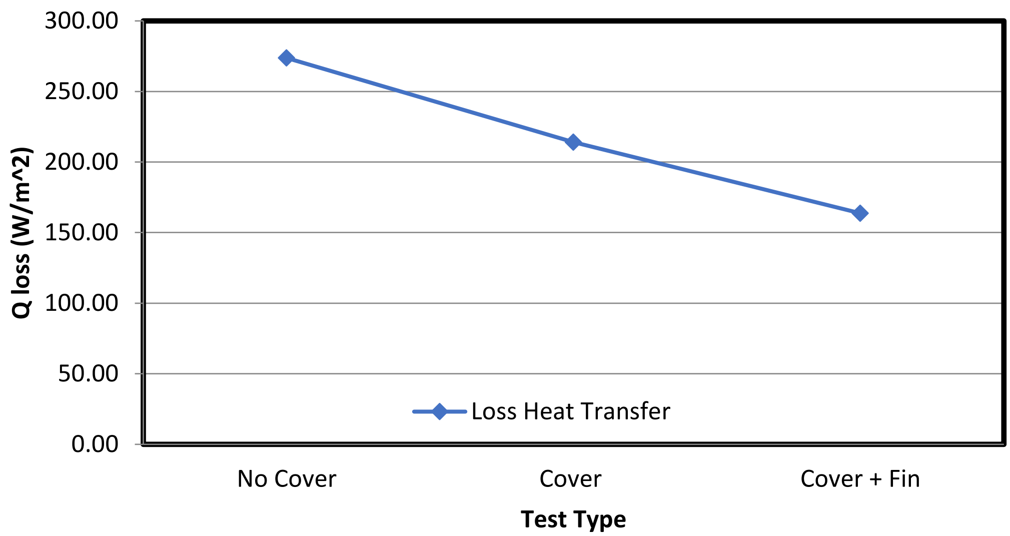


Fig. 4.11 compareson between heat loss of three cavity

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

A low cost solar collector with a dish reflector and spiral absorber is examined in this study. The experimental design was tested in this study. The solar ray distribution inside these receiver geometries, including the helical coil used for the heat transfer fluid, was determined using this tool. The final results show that enhancing in thermal performance as a result of reduce of thermal losses in cavity equipped with cover plus fin compared with cavity with cover and cavity with no cover.

5.2 Recommendation for future works

1) Dying the absorber tube by black color.
2) Increase length of absorber tube.
3) Increase number of coil in helical tube.
4) Increase number of fin per cover in receiver cavity.

APPENDIX B

Properties of saturated water (Cengel et al. 2002)

Temp. T, ℃	**Saturation pressure P_{sat}, kPa**	**Density ρ, kg/m^3**	**Enthalpy of Vaporization h_{fg} kJ/kg**	**Specific Heat C_p, J/kg .K**	**Thermal Conductivity k, W/m .K**	**Dynamic Viscosity μ, kg/m .s $\times 10^3$**	**Prandtl number Pr**	**Expansion coefficient β, $\times 10^6$ $1/K$**
0.01	0.6113	999.8	2501	4217	0.561	1.792	13.5	
5	0.8721	999.9	2490	4205	0.571	1.519	11.2	46.04
10	1.2276	999.7	2478	4194	0.580	1.307	9.45	114.1
15	1.7051	999.1	2466	4185	0.589	1.138	8.09	174
20	2.339	998.0	2454	4182	0.598	1.002	7.01	227.5
25	3.169	997.0	2442	4180	0.607	0.891	6.14	276.1
30	4.246	996.0	2431	4178	0.615	0.798	5.42	320.6
35	5.628	994.0	2419	4178	0.623	0.720	4.83	361.9
40	7.384	992.1	2407	4179	0.631	0.653	4.32	400.4
45	9.593	990.1	2395	4180	0.637	0.596	3.91	436.7
50	12.35	988.1	2383	4181	0.644	0.547	3.55	471.2
55	15.76	985.2	2371	4183	0.649	0.504	3.25	504
60	19.94	983.3	2359	4185	0.654	0.467	2.99	535.5
65	25.03	980.4	2346	4187	0.659	0.433	2.75	566
70	31.19	977.5	2334	4190	0.663	0.404	2.55	595.4
75	38.58	974.7	2321	4193	0.667	0.378	2.38	624.2
80	47.39	971.8	2309	4197	0.670	0.355	2.22	652.3
85	57.83	968.1	2296	4201	0.673	0.333	2.08	697.9
90	70.14	965.3	2283	4206	0.675	0.315	1.96	707
95	84.55	961.5	2270	4212	0.677	0.297	1.85	728.7
100	101.33	957.9	2257	4217	0.679	0.282	1.75	750.1
110	143.27	950.6	2230	4229	0.682	0.255	1.58	788
120	198.53	943.4	2203	4244	0.683	0.232	1.44	---
130	270.1	934.6	2174	4263	0.684	0.213	1.33	---
140	361.3	921.7	2145	4286	0.683	0.197	1.24	---
150	475.8	916.6	2114	4311	0.682	0.183	1.16	---
160	617.8	907.4	2083	4340	0.680	0.170	1.09	---
170	791.7	897.7	2050	4370	0.677	0.160	1.03	---
180	1,002.1	887.3	2015	4410	0.673	0.150	0.983	---
190	1,254.4	876.4	1979	4460	0.669	0.142	0.947	---
200	1,553.8	864.3	1941	4500	0.663	0.134	0.910	---
220	2,318	840.3	1859	4610	0.650	0.122	0.865	---
240	3,344	813.7	1767	4760	0.632	0.111	0.836	---
260	4,688	783.7	1663	4970	0.609	0.102	0.832	---
280	6,412	750.8	1544	5280	0.581	0.094	0.854	---
300	8,581	713.8	1405	5750	0.548	0.086	0.902	---
320	11,274	667.1	1239	6540	0.509	0.078	1.00	---
340	14,586	610.5	1028	8240	0.469	0.070	1.23	---
360	18,651	528.3	720	14,690	0.427	0.060	2.06	---
374.14	22,090	317.0	0	---	---	0.043	---	---

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