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ORIGINAL ARTICLE

Mechanical design of a low cost parabolic solar dish concentrator

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Abstract The objective of this research was to design a low cost parabolic solar dish concentrator with small-to moderate size for direct electricity generation. Such model can be installed in rural areas which are not connected to governmental grid. Three diameters of the dish; 5, 10 and 20 m are investigated and the focal point to dish diameter ratio is set to be 0.3 in all studied cases. Special attention is given to the selection of the appropriate dimensions of the reflecting surfaces to be cut from the available sheets in the market aiming to reduce both cutting cost and sheets cost. The dimensions of the ribs and rings which support the reflecting surface are optimized in order to minimize the entire weight of the dish while providing the minimum possible total deflection and stresses in the beams. The study applies full stress analysis of the frame of the dish using Autodesk Inventor. The study recommends to use landscape orientation for the reflective facets and increase the ribs angle and the distance between the connecting rings. The methodology presented is robust and can be extended to larger dish diameters.

1. Introduction

In the last ten years, oil prices became very high as well as the reserves amounts have been decreased. Moreover, burning fossil fuels such as coal, oil and natural gas for energy generation causes global warming and pollution problems. Based upon these facts, new resources of clean energy are necessarily needed. Renewable energy is the promising solution to this problem. Therefore, significant researches have been reported on how to utilize renewable energy resources efficiently. One of the most important resources of renewable energy is the solar energy which has widely spreading applications. It has been used for water heating, direct electricity generation by means of photovoltaic, and for steam generation using parabolic trough solar collectors. It is estimated that earth receives approximately 1000 W/m^2 amount of solar irradiation in a day [1]. Abbot [2] showed that this amount of irradiation could generate around 8500 TW worldwide and concluded that solar energy alone has the capability to meet the current energy

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Nomenclature

a	the distance from the focal point to the vertex	$\|L$	sum of the perimeter length of unfolded facets to be cut from the reflecting sheets
b	width of the cross-sectional area of the rib	n_f	factor of safety
d	distance between any two consecutive rings	W	weight of the dish frame
D	diameter of the dish	α	angle between any two consecutive ribs
FD	ratio of focal point position to dish diameter	δ	total deflection
FP	position of the focal point	σ	stress
h	height of the cross-sectional area of the rib		

demand. This has been confirmed based on theoretical calculation by Liu et al. [3].

Nowadays, there are numerous projects regarding the implementation of the solar concentrators. These projects have been done by research centers, universities and companies to design and to analyze the reliability and the performance of solar concentrator. Among the solar concentrators, parabolic dishes have the highest efficiency in the conversion of solar energy to electricity with an efficiency of 29.4% achieved [4]. Such systems have high optical efficiencies and low start-up losses leading to highly efficient solar energy engines. This gives the parabolic dishes the potential to eventually become one of the least expensive forms of renewable energy. Moreover, parabolic dish systems are typically designed for small-to-moderate capacity applications of the order of ten kilowatts which are suitable for remote power needs in rural areas and the places far away from the national electricity grid. In case of higher capacity needs, it is easy to install smaller capacities of the solar dishes connected together in a small farm [5]. Although parabolic dish concentrators offer the highest thermal and optical efficiencies among all solar concentrators demonstrated to date, they suffer from a higher cost of construction per unit area, compared to parabolic trough and Fresnel linear systems. Most of the studies carried out on parabolic dish concentrators have been focused on the thermal analysis of the solar energy conversion process including the engine. Johnston et al. [6] studied the optical losses in the spherical elements of the reflective surface of the dish. García et al. [7] presented a detailed thermal analysis of the Eurodish solar Stirling engine developed at the premises of the Sevilla Engineering School, Spain. The dish was designed to produce 10 kW. Wua et al. [8] had designed an improved version of the Solar/Stirling engine that could produce about 20 kW electricity through AMTEC project. It can be noted that the sizes of the parabolic dishes have been increased within the last two decades with the development of new technologies and the progress of automated fabrication. Lovegrove et al. [5] in the Australian National University had worked on dish systems for many years and his team developed 400 m^2 and 500 m^2 [9] dishes in the last decade. However, most of the implemented systems are still expensive and require large parcel of land. Therefore, designing a solar dish with small to moderate size with low cost to generate direct electricity is an important issue. Palavras [10] worked on the development and performance characteristics of a low cost dish solar concentrator and its application in zeolite desorption. An old damaged satellite dish purchased from a scrap yard was used. The reflecting surface was a polymer mirror film. It is concluded that a low cost is achieved compared with other previous work through the use of polymer film instead of curved glass mirror or polished aluminum mirror. A low cost solar steam generating system design, development and performance characteristics were investigated in [11]. Preliminary field measurements and cost as well as performance analysis of the system indicated that a solar to steam conversion efficiency is 70–80% at 450 °C.

Noting that little work has been done in the area of lowering the cost of solar dish, this paper demonstrates the mechanical design of low cost parabolic solar dish concentrator with small-to-moderate size for direct electricity generation. The strategy implemented in this study considers the following directions: (a) minimize the torque required by the motor to track the sun rays, (b) find the optimum distribution of the reflecting sheets which will be cut from stainless steel sheets available in the market, and (c) design a robust dish frame with minimum weight to support the entire engine and dish weight as well as expected wind forces. Different dish diameters have been studied with full stress analysis of dish frame using Autodesk Inventor.

2. Description of the components of the system

The parabolic dish solar concentrator system mainly consists of base support, concave dish frame, reflecting sheets, conversion unit and sun-tracking system as shown in Fig. 1. The tracking system is dual axis tracking system with slew drivers. The first slew driver ensures rotation of the concentrator around the vertical axis through all possible azimuth angles while the second slew driver ensures rotation of the concentrator around the horizontal axis through all possible elevation angles. There are two systems to handle the entire dish with the converting unit. Either to locate the base frame exactly at the CG point of the entire dish in order to reduce the torque on the transmission system or set it behind the entire dish. The present study implements the first technique, which has been designed and implemented by different companies in the market [12,13]. Intentionally, the axes are passing through center of gravity of the dish in order to avoid using counter weight and consequently reducing the needed driving torque. All these parts have to function properly to have an efficient system. However, from cost viewpoint, each of these parts has to have minimum cost as possible. Special attention has been given to lower the cost of preparing reflective sheets as well as to minimize the weight of the dish

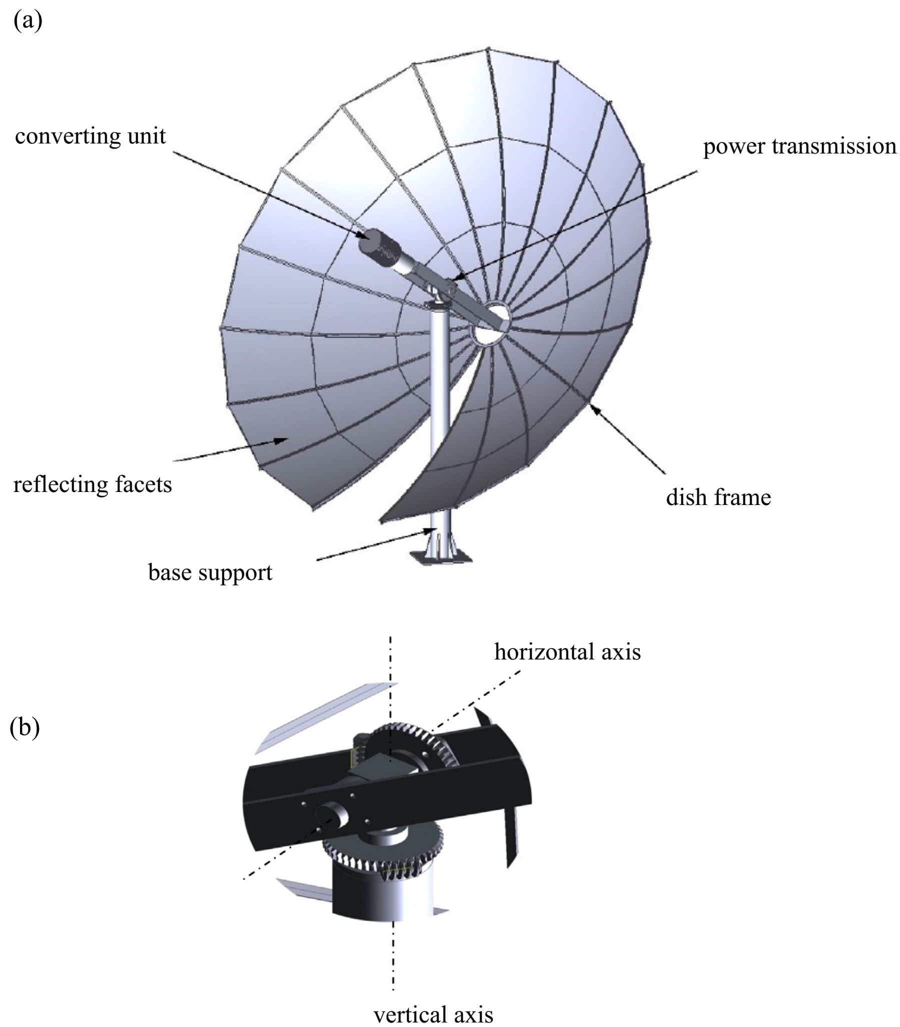

Figure 1 Schematic diagram: (a) dish frame connected to the base, and (b) transmission system.

frame. Lowering the cost of these parts will definitely decrease the total cost of the system.

2.1. Frame of the dish concentrator

The frame of the dish concentrator may have several types such as trusses, tubular bars, T-beams, H-beams, or rectangular beams. In the present study, rectangular beam is selected aiming at simplifying manufacturing and building process. The frame of the dish must bear the weight of the reflecting sheets and the wind forces as well as give minimum possible total deflection in the beams due to the applied mentioned forces. Since high deflection will change position of the focal point which affects the thermal efficiency of the system. Curved ribs are connected to base ring along the radius and they are joined by spaced rings along the circumference for robust structure as shown in Fig. 2.

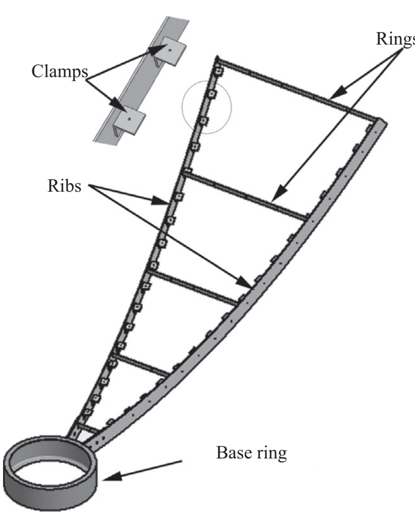

Figure 2 Sector of dish frame.

2.2. Reflective surface

Reflective surface is used to make the solar concentrator surface brighter in order to reflect as much as radiation to the conversion unit located at the focal point of the dish. In order to improve the thermal efficiency of the system, selection of reflective surface is very important. It may be glass mirrors, aluminum sheets, stainless steel sheets, stretched coated, or

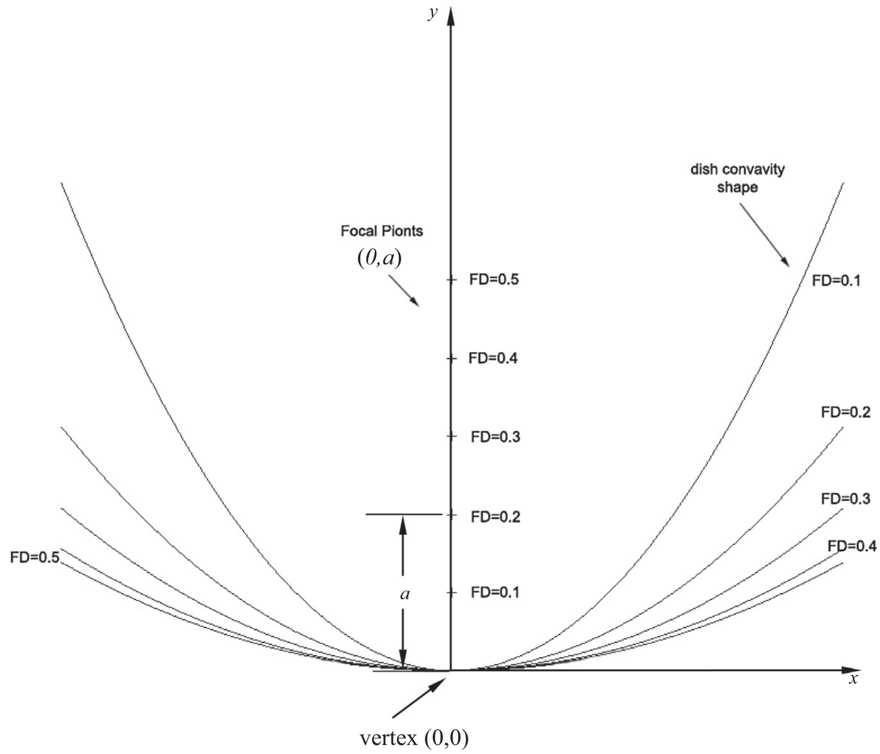

Figure 3 Effect of FD on dish curvature.

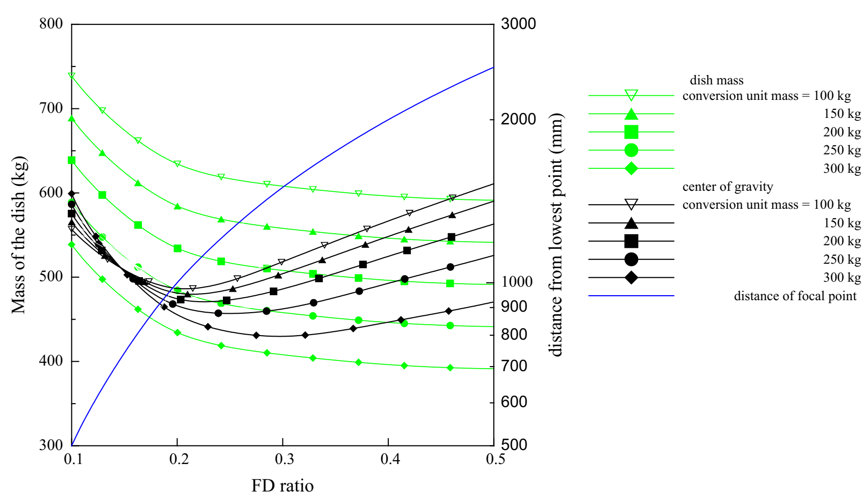

Figure 4 Effect of FD on weight and center of gravity of the dish frame.

stretched membranes [14]. To compromise between cost and efficiency, stainless steel sheets seem to be an appropriate selection. Moreover, stainless steel sheets are easy to be assembled, cleaned, and able to resist severe weather conditions.

3. Selection of the focal point

The focal point is the point where the reflected sunrays intersect. The distance of the focal point, a, measured from the vertex depends on the dish diameter, D, and the concavity of the dish as shown in Fig. 3. The more concave the dish is, the more closer the focal point will be. The ratio between the position of the focal point to the dish diameter is defined by FD (the focal to diameter ratio), where

$$a = FD * D$$

In order to locate the position of the focal point, computer program is conducted at different values of FD, ranging from 0.1 to 0.5, while diameter of the dish is equal to 5 m. The

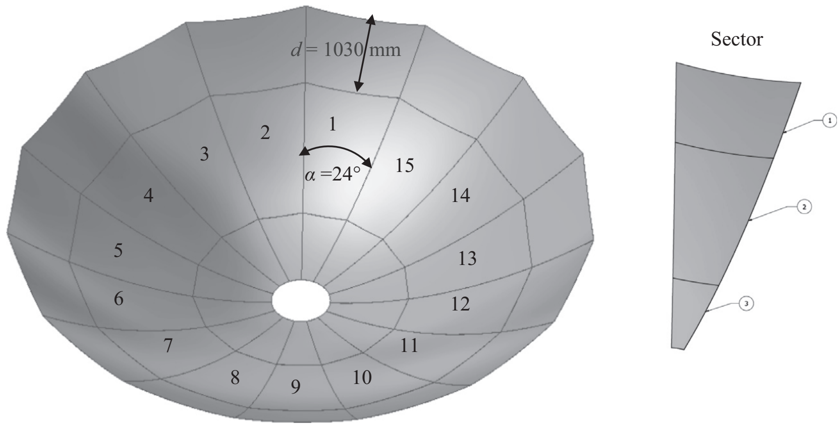

Figure 5 Dish sectors for $\alpha = 24°$ & $d = 1030$ mm (dish diameter = 5 m).

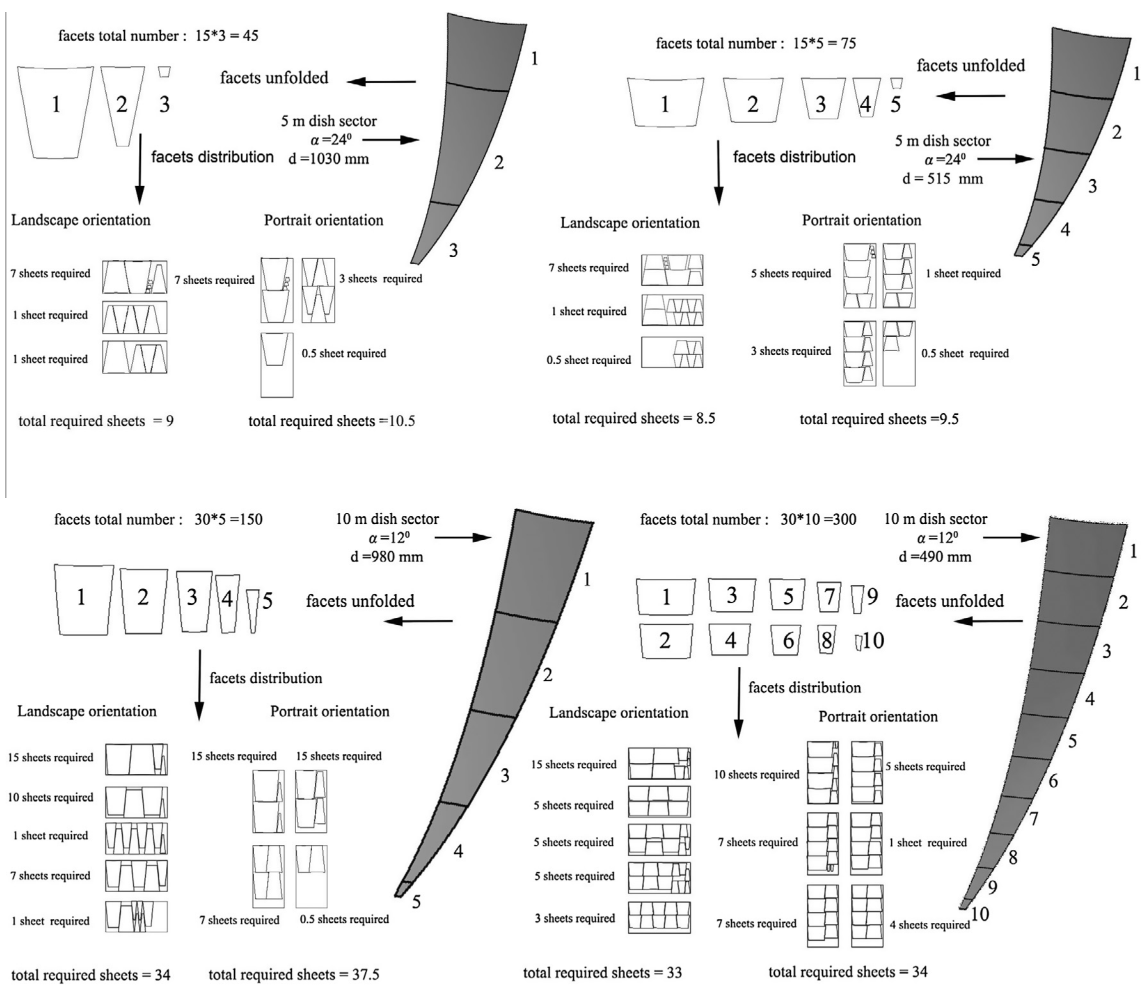

Figure 6 Samples of dish sectors, facets unfolding and distribution.

obtained results are presented in Fig. 3. Next, by using Autodesk Inventor, the dish frame has been drawn for each value of *FD* to calculate the weight of the model as well as the location of the center of gravity of the entire dish including the conversion unit mass which varies from 100 kg to 300 kg. This range of weights is set according to the weights of Stirling engines presented in the literature. The results are presented in Fig. 4. From the results, it is clear that the weight of the dish decreases as the *FD* ratio increases and if *FD* value is less than 0.2, the focal point position will be below the position of the center of gravity. On the other hand, as the *FD* increases, the distance of the center of gravity increases and the dish

Table 1 Optimum cutting of sheets (D = 5 m).

Dish diameter (m)	5																											
α (°)	3				6				9				12				15				18				24			
d (mm)	1030		515		1030		515		1030		515		1030		515		1030		515		1030		515		1030		515	
Number of reflecting facets	360		600		180		300		120		200		90		150		72		120		60		100		45		75	
Orientation of the sheet	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V
Number of required sheets	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	9.0	8.0	8.0	8.0	9.0	9.0	8.5	8.0	9.0	8.0	9.5	8.5	10.0	8.0	9.0	9.0	10.5	8.5	9.5
Cutting length of facets, L (m)	644.6		681.4		342.8		379.7		241.0		277.9		191.9		228.6		161.5		198.1		141.2		177.7		115.7		151.8	
Sheets cost ($)	560	560	560	560	560	560	560	560	560	630	560	560	560	630	630	595	560	630	560	665	595	700	560	630	630	735	595	665
Cutting cost (3$/m)	1933.9		2044.1		1028.5		1139.0		723.1		833.6		575.6		685.8		484.6		594.4		423.7		533.2		347.0		455.4	
Total cost (1000$)	2.49	2.49	2.60	2.60	1.59	1.59	1.70	1.70	1.28	1.35	1.39	1.39	1.14	1.21	1.32	1.28	1.04	1.11	1.15	1.26	1.02	1.12	1.09	1.16	0.98	1.08	1.05	1.12

Table 2 Optimum cutting of sheets ($D = 10$ m).

Dish diameter (m)	10															
α (°)	3				6				9				12			
d (mm)	980		490		980		490		980		490		980		490	
Number of reflecting facets	600		1200		300		600		200		400		150		300	
Orientation of the sheet	H	V	H	V	H	V	H	V	H	V	H	V	H	V	H	V
Number of required sheets	33.0	34.0	33.5	33.0	33.0	37.0	33.0	34.0	34.0	35.0	33.5	34.0	34.0	37.5	33.0	34.0
Cutting length of facets, *L* (m)	1416.1		1580.4		788.7		948.5		579.3		738.8		474.3		633.4	
Sheets cost($)	2310	2380	2345	2310	2310	2590	2310	2380	2380	2450	2345	2380	2380	2625	2310	2380
Cutting cost (3$/m)	4248.4		4741.2		2366.1		2845.6		1737.8		2216.4		1423.0		1900.3	
Total cost (1000$)	6.56	6.63	7.09	7.05	4.68	4.96	5.16	5.23	4.12	4.19	4.56	4.60	3.87	4.05	4.21	4.28

Table 3 Optimum cutting of sheets ($D = 20$ m).

Dish diameter (m)	20							
α (°)	3				6			
d (mm)	950		475		950		475	
Number of reflecting facets	1200		2400		600		1200	
Orientation of the sheet	H	V	H	V	H	V	H	V
Number of required sheets	136	137	135	133.5	137.5	150	138.5	139
Cutting length of facets, *L* (m)	3228.12		3887.16		1942.62		2613.06	
Sheets cost($)	9520	9590	9450	9345	9625	10,500	9695	9730
Cutting cost (3$/m)	9684.36		11661.5		5827.86		7839.18	
Total cost (1000$)	19.20	19.27	21.11	21.04	15.45	16.33	17.53	17.57

Table 4 Optimum values of dish parameters.

Dish diameter (m)	α (°)	*d* (mm)	Orientation of the sheets	Total cost ($)	Cost/m^2 ($)
5	24	1030	H	977	49.7
10	12	980	H	3873	49.3
20	6	950	H	15,453	49.2

becomes flatter as shown in Fig. 3. The focal point should be located slightly above the center of gravity in order to decrease the heat loss due to reflected rays deviated out of conversion unit in case of large deflection that may be caused by wind forces. Therefore, a value of 0.3 for the *FD* is thought of as a reasonable compromise in the present study.

4. Reflecting facets, unfolding and optimum distribution of the stainless steel sheets

As shown in Fig. 2, the intersection of ribs and rings forms variable sizes of reflecting facets which shall be cut from the available stainless steel sheets in the market. There are two sizes of the sheets 1 m × 2 m and 1.25 m × 2.5 m. In this study the larger size; 1.25 m × 2.5 m is considered. It is noted that the largest facet is formed at the outmost edge of the sector. Taking into account that the largest facet when being unfolded should fit within the boundaries of the sheet area, the angle between any two consecutive ribs, α, and the distance, *d*, between any two consecutive rings must not exceed specific limits based on the available sheet size. The limits of the facets are bounded by the size of the selected sheet (1.25 m × 2.5 m). This means that the angle α is limited by the outmost arc which should be less than 2.5 m if the sheets are distributed in a landscape manner. On the other hand, the limit of distance *d* should be less than 1.25 m. Fig. 5 shows an example in case of dish diameter equals to 5 m where α and *d* are limited to 24° and 1030 mm respectively. In this case, the dish is divided into 15 sectors and each sector has three reflecting facets.

Based upon the limits mentioned previously, for the dish diameters considered in this study (5 m, 10 m and 20 m) the angle α and the distance *d* must not exceed 24°, 12° & 6° and 1030 mm, 980 mm and 950 mm respectively. A compromise between angle α and distance *d* is presented in order to minimize the number of sheets required and the cutting cost. This is done by changing the angle α from 3° to 24° and the distance *d* to be 515 mm or 1030 mm. The orientation of the sheets is tested for both portrait (V) and landscape (H) layouts.

For each assigned value of α and *d*, the resulting reflecting facets are unfolded using Advanced Smart Unfolding software integrated with the Autodesk Inventor giving a number of flat patterns as shown in Fig. 6. The figure describes the unfolding procedures in case of dish diameters equals to 5 m and 10 m. The subfigures at the left are for ring distance equals to *d*, while the subfigures at the right are for ring distance equals to 0.5 *d*. In the cases shown, the flat patterns are distributed over landscape/portrait oriented sheets in order to know the required number of sheets for each case. In case of $D = 5$ m, $d = 1030$ mm and $\alpha = 24°$, the dish is divided into 15 sectors

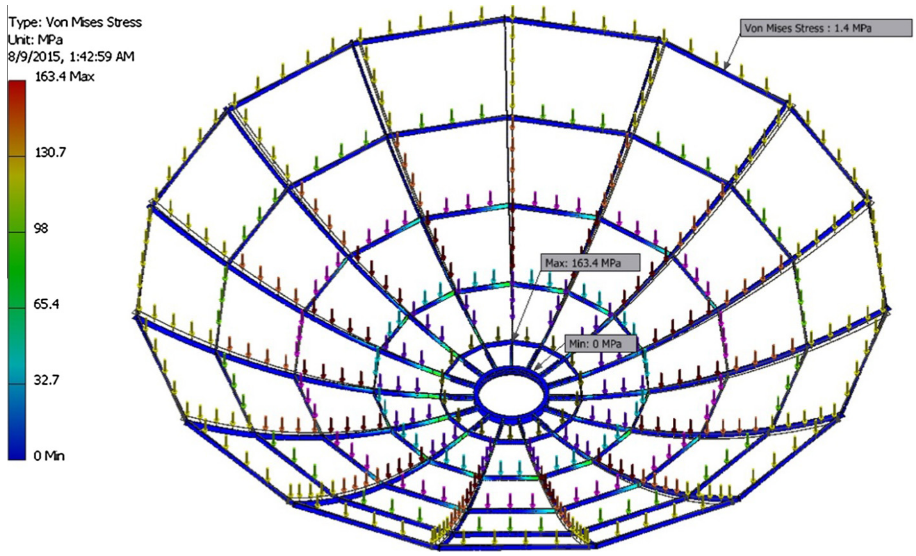

Figure 7 Environment of stress analysis generated by Autodesk Inventor.

having 45 flat patterns; these 45 patterns are divided into three similar groups and each group has 15 identical patterns as shown in Fig. 5. When these patterns are distributed over landscape orientation, 9 sheets are required where 10.5 sheets are required in case of portrait orientation. On the other hand, for the same case but $d = 515$ mm, 8.5 sheets are required for landscape orientation and 9.5 sheets for portrait orientation of the sheet. Next, the total cost of reflective surface is calculated considering the cost of the required sheets as well as the cost of the laser cutting of the reflecting facets from the stainless steel sheets. The results of all cases considered in this study are presented in Tables 1–3.

According to the total cost presented in Tables 1–3, the optimum values of the parameters for each dish diameter are listed in Table 4. The results show that the cost of installing the sheets per m^2 of the dish is about 49.5 $.

5. Stress analysis of the dish frame

Available dish frames vary from small dish diameters to large ones. Most of the dish frames with small diameters use single curved beam arrayed in 360° while the dish frames with large diameters use truss structure in order to have robust design. Of course, design constraints and cost are two important guides for proper design. Design constraints are the total deflections and the loads due to entire dish weight and wind forces. The cost depends on manufacturing, assembling and weight of the frame. Truss structure is more robust than single beam but it is more complex in manufacturing and assembling. In addition, its large weight increases the cost. Therefore, simple structure is selected rather than complex one in this research. The structure is made of rectangular beams which have a wide range of standard sizes and are easy to manufacture.

The stresses acting on the dish frame are generated due to weight of the structure and wind forces induced on the reflecting facets. The wind force is proportional to the projected area of the reflecting facets and the wind speed. In the calculations the wind speed is set to be 60 miles per hour in order to simulate severe weather conditions.

The stress analysis is conducted over the whole dish frame where wind force on the reflecting facets is directly transferred to the supporting clamps and to the ribs as bearing reactions from the bolts that fix the clamps to the ribs. Fig. 7 illustrates the stress analysis environment generated by Autodesk Inventor. Initially, the effect of varying the ring dimensions on the strength of dish frame has been studied. The results are shown in Table 5 which indicate insignificant effect on the strength of the dish frame. Therefore, the dimensions of the rings can be selected as minimum as possible. Next, the value of width, b, and the height, h, of the rib cross-section is varied in order to find their optimum values. Using Inventor software, maximum stress, maximum total deflection and minimum factor of safety are obtained and the results are presented in Figs. 8 and 9 for dish diameter 5 m and 10 m, respectively. For each obtained set of dimensions, total mass of the dish frame can be calculated.

From these figures, the designer can easily find the optimum dimensions of the ribs forming the dish frame that satisfies a specific design target. For example, for a dish diameter of 5 m, if the goal is to design dish frame with factor of safety of 2 associated with minimum mass, simply draw horizontal line corresponding to this factor of safety that intersects the curves at different possible dimensions of the rib. Then, these intersection points are projected to the abscissa line to get the height of

Table 5 Effect of ring dimensions on strength of dish frame of diameter 5 m.

Rib dimensions (mm)	Ring dimensions (mm)	Max. stress (MPa)
80 × 6	20 × 2.5	99.8
80 × 6	30 × 4	98.7
80 × 6	80 × 4	98.4
40 × 5	20 × 2.5	458.2
40 × 5	30 × 3	457.7
40 × 5	40 × 4	456.2

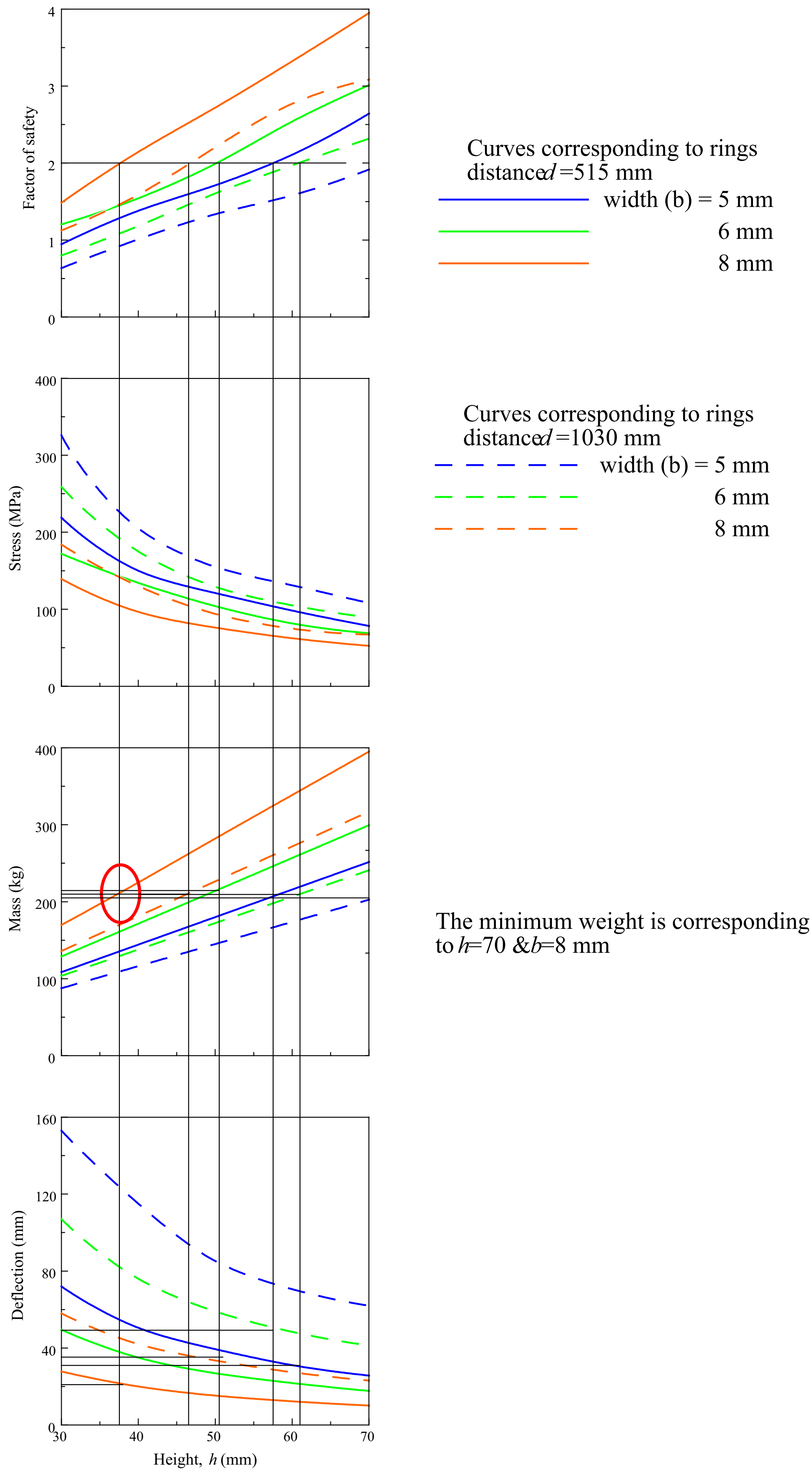

Figure 8 Ribs optimum dimensions curves for dish diameter of 5 m.

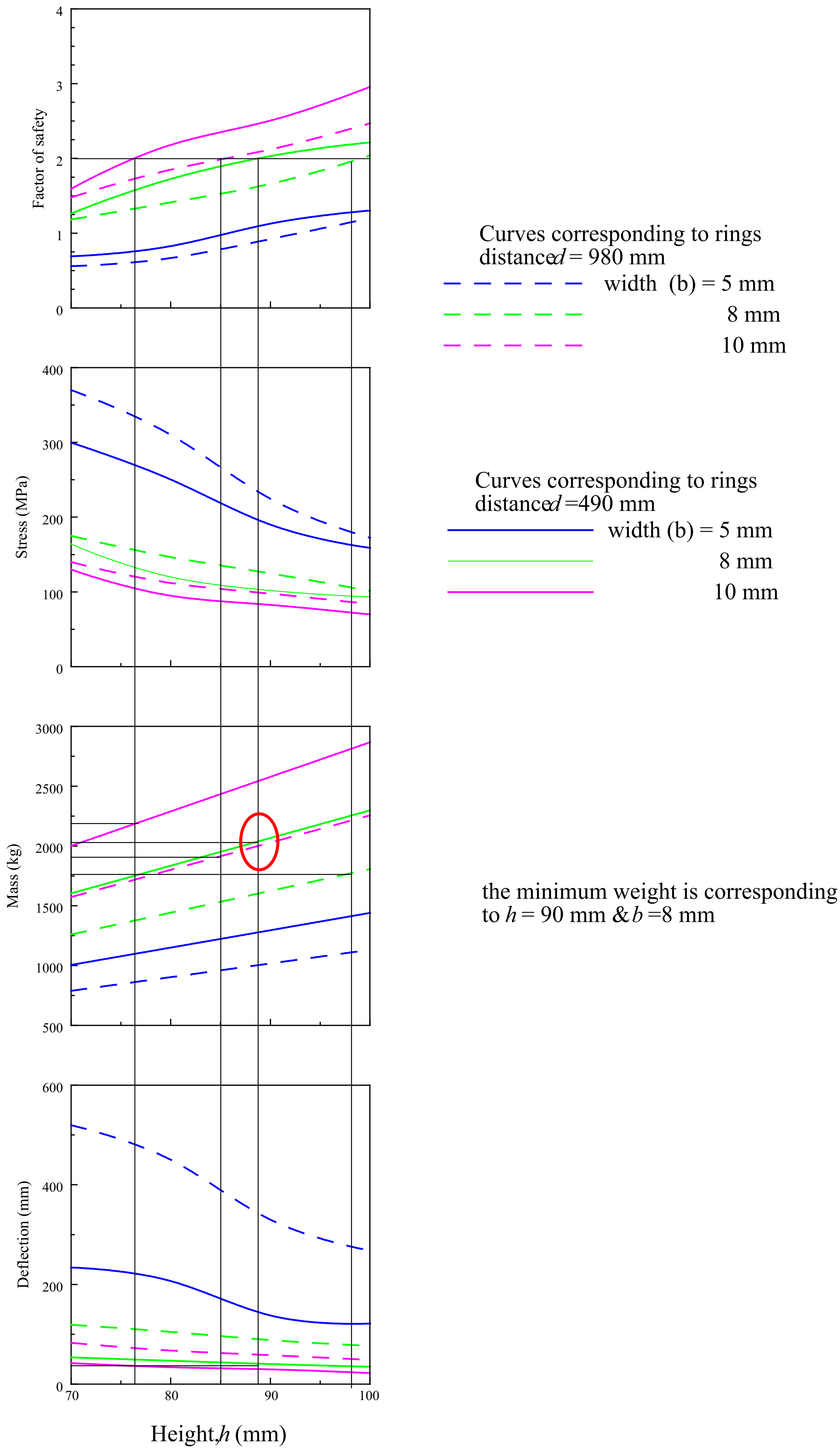

Figure 9 Ribs optimum dimensions curves for dish diameter of 10 m.

the ribs as shown in Fig. 8. It is clear that many values of width, b, and height, h, of the ribs satisfy the desired factor of safety, but only rib dimensions equal to $40\,\text{mm} \times 8\,\text{mm}$ (in case of 5 m diameter) give minimum weight for the dish frame. The same procedure can be applied to dish diameter of 10 m as shown in Fig. 9. Moreover, these figures are convenient for a designer to know the applicability of using ribs with known dimensions in the manufacturing process by determining the expected total deflection and the factor of safety for such ribs.

6. Summary and conclusion

The present study aims at designing a low-cost solar parabolic dish concentrator from the mechanical perspective for direct electricity generation. Minimizing the cost necessitates the reduction of the entire structure weight and minimizing the cost of the reflective surface. To compromise between cost and efficiency, stainless steel sheets with standard dimensions available in the market are used as the reflective surface. Advanced Smart Unfolding software integrated with the Autodesk Inventor is used to find out the proper dimensions and orientation of the reflective facets that should be cut from the stainless steel sheets. The results show that as the angle between the ribs and the distance between rings increase to their upper limits, the cost of the reflective surface will be decreased. In addition, the unfolding of the sheets shows that landscape orientation gives lower number of sheet hence lower cost of the reflective surface. The focal to diameter ratio is selected to be 0.3 and three different diameters; 5, 10 and 20 m are investigated. Stress analysis of the dish frame has been carried out using Autodesk Inventor for optimizing the dimensions of ribs and rings forming the dish frame under the effect of the dish weight and the wind force. The results show that for dish diameter equals to 5 m a rectangular beam with $40\,\text{mm} \times 8\,\text{mm}$ cross-sectional area satisfies the desired factor of safety with minimum mass and acceptable value for the deflection. On the other hand, a rectangular beam with $90\,\text{mm} \times 8\,\text{mm}$ cross-sectional area can be selected in case of 10 m dish diameter. The presented results reveal a robust design of the dish concentrator that can deal with various applied forces and weights to provide the minimum possible cost. The methodology applied in the present study can be extended to any proposed dish diameter. The future plan is to integrate the dish with a converting unit such as Stirling engine and consider the vibration resulting from the engine operation with possible application to larger diameters.

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