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K. Bammert
Professor,
Institut for Turbomachinery,
University of Hanover,
Hanover, West-Germany

A. Sutsch
Institute for Computer Assisted
Research in Astronomy,
Alterswil, Switzerland

M. Simon
M.A.N. Maschinenfabrik Augsburg-Nürnberg
AG Division New Technology,
Munich, West-Germany

A. Mobarak
Professor,
Mechanical Power Department,
Faculty of Engineering,
Cairo University, Egypt

Small Gas Turbine with Large Parabolic Dish Collectors

An alternative solution for solar energy conversion to the heliostat-tower and solar farm (parabolic trough) concept is presented in the form of large parabolic dish collectors using small high temperature gas turbines for producing electricity from solar thermal energy. A cost and efficiency comparison for the different solar thermal power plants has shown that the large parabolic dish with gas turbine set is a superior system design especially in the net power range of 50 to 2000 kW. The important advantages of the large parabolic dish concept are discussed. For the important components such as the gas turbo converter, the receiver and the parabolic dish collector, design proposals for economic solutions are presented. An advanced layout for a 250-kW gas turbo converter with recuperator is presented in detail

INTRODUCTION

Solar energy represents a vast source of energy that is available in many areas where population density is low and where plants working with conventional energy resources may not be adequate. For small communities in remote areas, small power plants in the order of a few hundred kilowatts are needed for pumping water for irrigation as well as electric power production. It is important, however, to keep the conversion system as simple as possible with low maintenance, as the cost of a solar thermal power plant is a major deterrent to its use.

For thermal conversion of solar energy into mechanical and electrical energy two main processes are available: For small outputs the solar farm concept can be considered where the working fluid (water) is heated to approx. 300 °C in concentration collectors connected in parallel and/or series (Rankine cycle). Due to the low maximum operating temperature, only low efficiencies can be achieved. If solar radiation is concentrated through a large single parabolic dish (diameter 20 - 100 m) or with an array of heliostats onto a receiver in which the working fluid is heated, very high maximum process temperatures and thus very good efficiencies can be achieved. Because of the materials used in receivers, the maximum process temperature is at present limited to about 950 °C for steel alloys [1]. Under the present state-of-the-art, these temperatures can only be utilized in gas turbine plants (Brayton cycle).

Fig. 1 shows an artist's concept of the parabolic dish with receiver and turbo converter set at the focal plane. The dish exhibits an inherent advantage over the solar farm and solar tower plants [2]. When not in power conversion mode, the dish can easily be changed over for communication. The change over is completed in approximately 20 minutes with communications equipment permanently installed in the foot of the dish and the insertion of a cassegrainian reflector in front of the receiver. This function provides a link to the outside world for remote areas with no infrastructure in power or communications.

A comparison between the different solar power plants follows. It will be shown that for the power range of 50 to 2000 kW, it is advantageous to choose large, parabolic dishes with a gas turbine for solar power generation. Also, selection and design criteria for the dish, the gas turbine set and the receiver are presented and discussed.

COMPARISON BETWEEN DIFFERENT SOLAR THERMAL POWER PLANTS

The large parabolic dish system (LPDS) with a gas turbine can not be directly compared with the solar tower or solar farm plants. This is because we are dealing with completely different systems.

In the solar tower system concept, solar radiation is collected and concentrated optically with tracking mirrors (heliostats) onto a central receiver. In the receiver,

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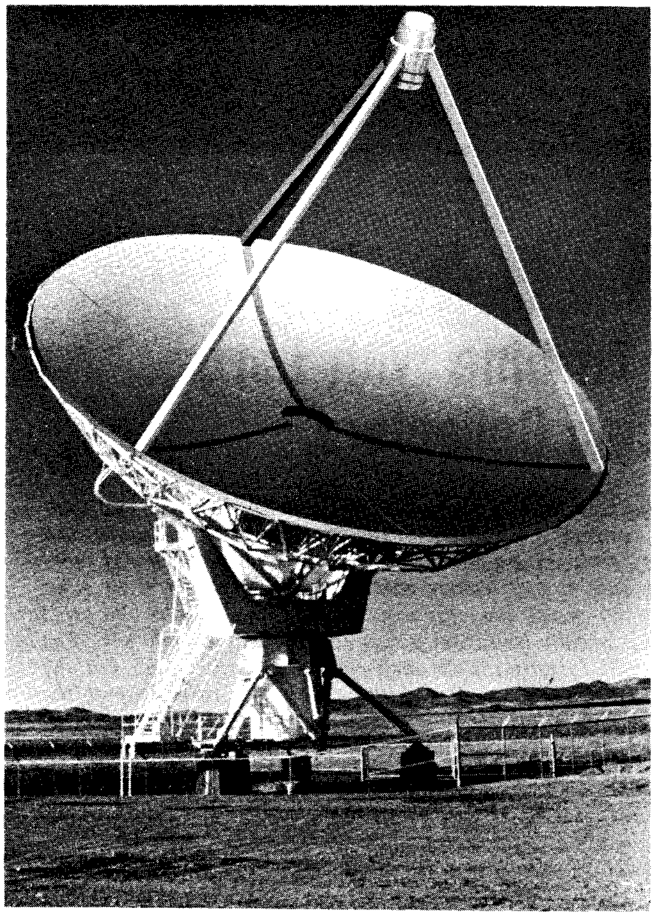


Fig. 1 Large parabolic dish system (LPDS)

the radiation energy is exchanged to a thermal working medium with a high process temperature. In the solar farm concept solar radiation is concentrated by many individual troughs or bowls (cylindrical or spherical concentrators); it is transferred in each absorber onto a working fluid of medium process temperature and passes via a tubing system onto a common conversoin unit. Tower and farm concept use a central energy conversion unit (farm 10 - 1000 kW, Tower 1 - 100 MW).

The LPDS contains within each individual collector a complete gas turbo conversion unit (50 - 2000 kW) and could be connected electrically in the case of larger field array systems. For the LPDS, losses in power are less than in thermal and optical energy collection of the farm and tower systems. Redundancy of single autonomous units (LPDS) provides high availability and flexibility as well as higher efficiency due to higher concentration ratios and smaller field losses.

EFFICIENCY

Efficiency plays an important role when evaluating the three different types of solar power plants. The comparison can be achieved, if we consider three efficiencies linked in the following manner:

$$\eta_o = \eta_{col} \cdot \eta_{plant} \qquad (1)$$

where η_o is the overall efficiency of the entire installation $(P/\dot{Q}_{sol})$; P is the net electric power and $\dot{Q}_{sol}$ is the irradiated solar energy in a given time unit onto the collector area. η_{col} is the total collector efficiency $(\dot{Q}/\dot{Q}_{sol})$ the quotient of absorbed heat in the receiver by the working fluid and irradiated solar energy $\dot{Q}_{sol}$ on the collector. η_{plant} is the plant efficiency $(P/\dot{Q})$, the quotient of electric output P and thermal power $\dot{Q}$ absorbed by the working medium.

Fig. 2 shows the collector efficiency for different solar thermal power plants as a function of maximum working medium temperature T_4. Due to relatively low concentration ratios in farm plants (C = 50 for trough with line concentrators and C = 200 for spherical bowl with absorber) and relatively large convection losses, collector efficiency is only approximately 55 - 62 %. Maximum working fluid temperatures are between 300 and 400 °C in Rankine cycles. Tower concepts with concentration ratios of C = 500 for a steam process and C = 1500 for a gas turbine process render more advantageous values: collector efficiency in tower plants is approximately 64 %. In the LPDS with special steel tube receivers, upper process temperatures of 900 °C with a concentration factor of C = 3000 can be achieved. For ceramic receivers with concentration ratios of C = 6000, upper process temperatures of approximately 1200 °C are possible. Collector efficiency is approximately up to 67 % in these applications.

In this comparison we have based our calculations on a total collector efficiency η_{col} = 70 % as a limit in farm and tower concepts and 80 % in LPDS. Optical losses, i.e. reflector losses, reradiation losses, cosine losses, etc. have been considered.

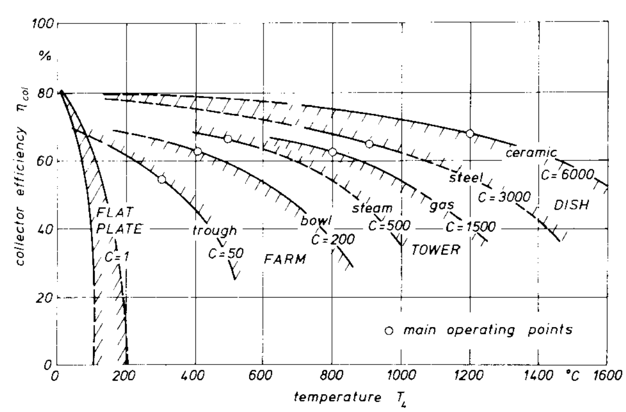


Fig. 2 Collector efficiency versus maximum operating medium temperature

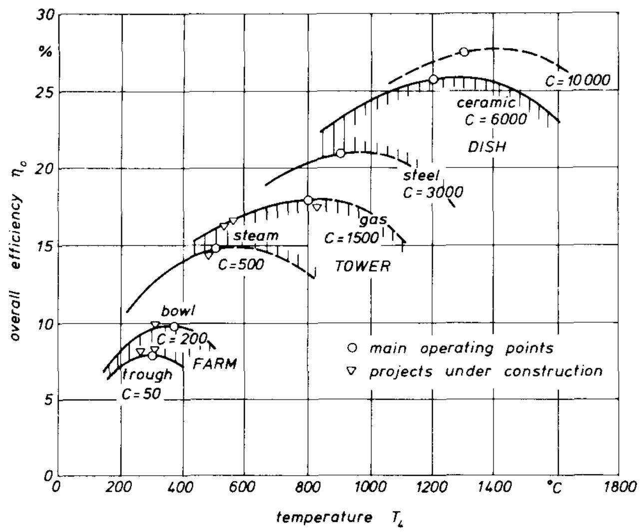


Fig. 3 Overall efficiency versus maximum operating medium temperature

Fig. 3 shows overall efficiency η_o that can be achieved for solar power plant systems as a function of the maximum working medium temperature T_4. Solar farms achieve approximately η_o of 8 % for trough and 10 % for the bowl type. Solar tower concepts achieve 15 % for a steam process and 18 % for a gas process. The LPDS reaches under similar conditions an efficiency of 21 % using steel tube receivers and 26 % with ceramic receivers. A further increase in dish efficiency can be expected when concentration ratios of 5000 to 10 000 are exploited. This comparison only marginally accounts for the most important system parameters: the system size which plays an important role in component efficiency. These figures here apply to systems in their respective power range.

Considering plant efficiency η_{plant} which is a function of component size and

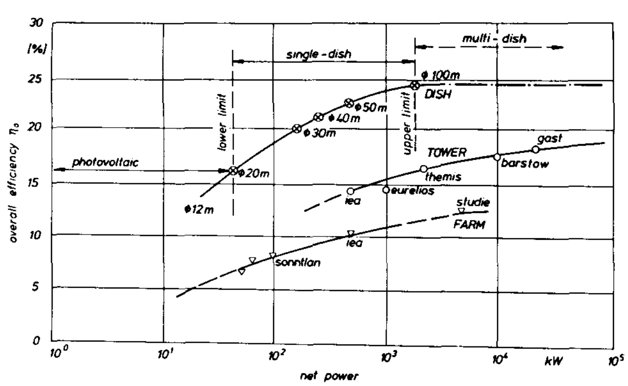


Fig. 4 Overall efficiency versus plant size for different systems

x Planned dish projects
O Tower projects
▽ Farm projects

the corresponding collector efficiency, overall efficiency as defined in equation (1) is the result. Fig. 4 shows η_o as a function of plant size. As can be expected, efficiency rises with increasing plant size. For plants (farm and tower) under construction the respective values for efficiency and net power have been plotted. Highest efficiency values are achieved by the LPDS with gas turbine because of the very high process temperature T_4. In addition, Fig. 4 shows two limits for the LPDS with gas turbine: The lower limit is reached with a dish collector diameter of approximately 20 m with 40 kW net power output; for smaller outputs photovoltaic energy systems should be considered. As an upper limit we have taken a maximum dish diameter of 100 m with an electric net output of approximately 2000 kW. For even larger systems, multi dish systems would be connected electrically in parallel. Therefore, overall efficiency is nearly constant for higher net power output.

COLLECTOR COST

The advantages of higher collector - and overall efficiency can only be judged if considered in conjunction with collector cost accounting for the largest item in cost of the total plant. Fig. 5 shows an estimate of the total collector cost spectrum in US\$/m^2 collector area as a function of the concentration ratio. On top of the diagram the respective ranges are displayed C = 1 (flat plate collector), C = 2 - 6 (Winson-trough), C = 10 - 50 (parabolic trough), C = 100 - 300 (bowl), C = 500 - 2000 (tower) and C = 2000 to 10 000 (LPDS).

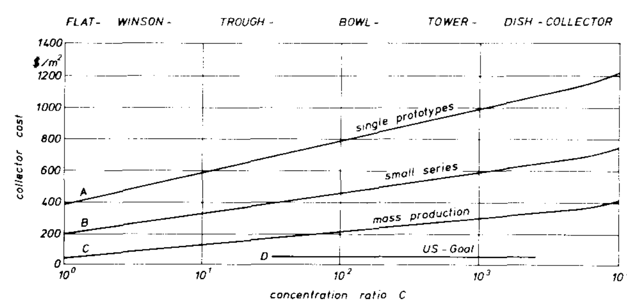


Fig. 5 Collector cost estimates versus concentration ratio

The curves begin with flat plate collectors at C = 1 with an estimated cost around US\$ 50, 200 and 400 per m^2. We differentiate between 4 trend curves:

A single prototype installation
B small series
C mass production
D theoretical design and development goal

Trend curves A and B correspond very well to the experience as of today. Curve C is expected for mass production whereas D reflects the US design goal for mass production of heliostats and parabolic dishes [3, 4].

INVESTMENT COST

Taking the above mentioned figures into consideration, investment cost trends for the different plants can be compared. Fig. 6 shows the cost per installed kW in US Dollars as a function of plant size. As can be expected, cost decreases with increasing plant size. Investment cost relates to estimates for production in series from 1990 - 2000 for solar power plant projects (a to e). Values for fossil power plants (f to h) and for nuclear power plants (i) have been taken from the current literature and doubled [5] for 1990 - 2000.

Fig. 6 shows the dominating role of photovoltaics for smaller power plant units (1-40 kW). For this power range very small gas turbine plants have been also suggested [6]. This range is followed by the LPDS from 40 to 2000 kW for the lowest investment cost. For 10 to 100 MW the solar tower plants present a viable solution. Judging from trends in the curves in Fig. 6 one can expect the LPDS between 50 to 10 000 kW (single and multi dishes) to be one of the most economic sources of solar energy conversion, especially if a further cost increase in fossil fuels for convetional power plants is taken into consideration.

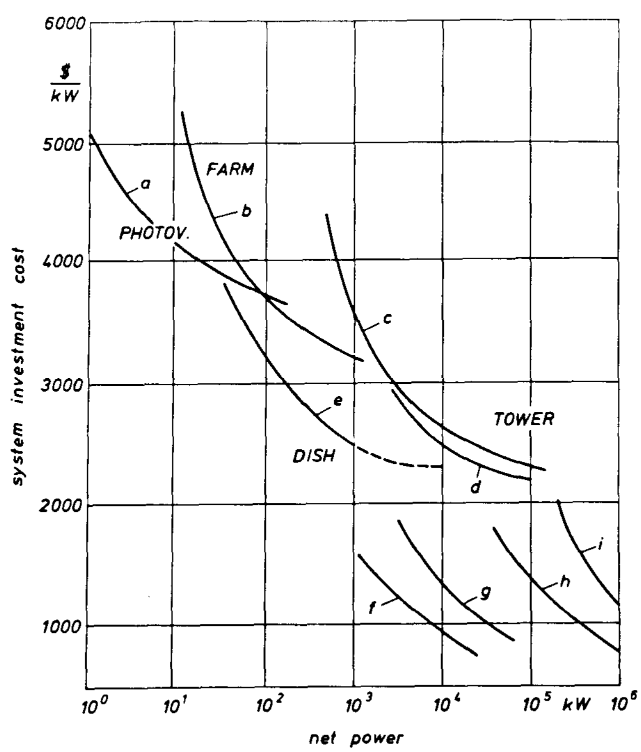


Fig. 6 Investment cost versus plant size for different systems

a Photovoltaic
b Farm
c Tower (steam)
d Tower (gas)
e Dish
f Gas turbine plant
g Diesel plant
h Steam plant
i Nuclear plant

SYSTEM DESCRIPTION

We now present the individual components and features in detail for LPDS and discuss a specific layout for a 250 kW solar energy power plant [7]. This plant utilises a 40 m in diameter parabolic dish collector with a high-temperature metal tube receiver and an open-cycle gas turbine with recuperator. After conversion to communication mode, the dish becomes either an earth station to handle telephone and telex network for international and national communications or can be applied to direct line communication work. This renders a 24 hours system use and when a suitable storage is provided, the advantage of the LPDS over other solar power plants is maintained [8]. We are also working on high temperature storage unit to complement the design features of the LPDS.

PARABOLIC DISH COLLECTOR

Large parabolic dish collector systems are being used for radioastronomy and communication all around the world. These dishes have been manufactured in the past up to 100 m in diameter, fully steerable, with very exact specifications, tracking on objects with far greater accuracy than required in this application. For the INTELSAT telecommunication satellite network alone, over 300 dishes (antennas) with a diameter of approximately 30 m are in service.

Fig. 1 shows such a prototype parabolic dish collector taken from present radio-astronomical antenna designs:

a) the foundations and ground work,

b) a two-axis tracking mount in either altazimuth (azimuth/elevation) or equatorial (right-ascension / declination) configuration,

c) dish support structure made of a tubular grid of aluminium or steel,

d) the dish panels with adjusting screws with either aluminium and SiO_2 protection coating (under high vacuum) or thin glass (reflectivity approximately 92 %).

e) the struts for the focal plane cage and support structure, in this case supporting the receiver, and the secondary reflector for communication purposes.

The main design data for a 40 m parabolic dish and for a 250 kW power plant are listed in table 1. Two versions are presented:
a) the antenna prototype and
b) a future low cost alternative.

Table 1 Design data for 40 m parabolic dish

GENERAL DESIGN DATA:
dish diameter: 40 m
total projected area: 1255 m^2
focal length: 32 m
focal length - diameter ratio: 0.8
receiver and struts shadowing: 1 %
ideal concentration ratio C: 17 000
ideal solar image diameter: 0.300 m
actual concentration ratio: 5 000
actual solar image diameter: 0.564 m
actual area of solar image: 0.250 m^2
power per unit area in focal plane: 3 000 kW/m^2
collector efficiency: 65 %
incident solar radiation: 0.925 kW/m^2
a) ANTENNA PROTOTYPE
panel size: 2.5 x 1.0 m (in 7 concentric rings)
panel thickness: 4 mm
panel accuracy: 0.4 mm from true paraboloid
panel material: aluminium
panel coating: aluminium plus 0.5 mm SiO_2 (surfaced under high vacuum)
panel reflectivity: 92 %
b) FUTURE LOW COST ALTERNATIVE
double membrane typ
membrane material: steel
membrane thickness: 0.6 mm, 20 strips welded together from a 2 m steel roll
membrane forming: deep-drawn with special parabolic load profile
membrane accuracy: 0.4 - 0.6 from true paraboloid
membrane support: ring-structure or self supporting sandwich
membrane coating: thinglass mirror 0.7 mm
membrane reflectivity: 90 %
concentration ratio C: 3000 - 5000

Since panel accuracy is a direct measure for concentration ratio, flux density and temperature in the focal plane, smooth, highly accurate reflective aluminium panels are required as a dish surface. State-of-the-art today is approximately 0.4 mm overall panel accuracy (between 1 to 3 mrad) in a given design position of the dish. Deviation from this base position results in mechanical deformation of the dish panels and structure and is calculated to follow the "best-fit paraboloid" method, i.e. the deformation will result in a different parabola to which the focal plane cage with the receiver should be moved. In the case of a 40 m diameter dish collector, this deformation is within a few millimeters from the true figure for a solar position of $\pm$ 23.5° from the celestical equator and can be neglected.

Because of the relatively high investment cost for such antenna structures, and since the dish collector represents the most cost intensive component, we are looking at various alternative developments for low cost collector systems (future generation) to approach the economic cost goal of 100 to 200 US\$/m^2. One of these developments uses large metal membranes with special load profile inside a gimbal mounted ring structure like the experimental prototype of M.A.N. company [9].

Further design proposals consider self-supporting double membranes with parabolic shape and internal sandwich filling through plastic deformation under a special load profile [10].

TRACKING, GUIDING AND CONTROL

There are two different methods in use today for tracking on celestial objects (the sun):

a) The equatorial mount with one axis pointed to the celestial north or south pole depending on geographic latitude - the right ascension or hour axis - and perpendicular to it the declination axis. A type of equatorial mounting is called the "English equatorial mounting" in astronomy and has always been utilized where heavy loads had to be contended with at low cost, where the polar axis rests on two pillars and allows attachment of dish and power conversion unit to move freely.

b) The altazimuth mount consisting of an azimuth and elevation axis perpendicular to one another without pointing to any specific celestial reference point but mounted to point to the exact site zenith. This type of mount requires sophisticated, constant two-axis process computer control which today, however, represents no problem to electronics [2].

Guiding for both types of mount is always achieved with solar sensor with computer back-up for cloud passages. Guiding accuracy to one arc-second represents no challenge to modern day electronics and has been developed inhouse [8].

COMMUNICATION MODE

The main difference between existing solar thermal power plant designs and the parabolic dish is its bivalent use as a communication center during non-sunshine hours, i.e. from sunset to sunrise. For the first time, a solar power plant is put to use on a 24 hours basis for different purposes to cut down investment cost and accelerate system repayment time.

With the dish solar plant this is made possible since the parabolic dish collector idea was taken from communications and radio-astronomy. The two modes of operation do not conflict with one another as they are handled be different operators. The changeover from power to communication mode is carried out in less than 20 minutes and consists only of the insertion of a secondary cassegrainian subreflector in front of the receiver cage.

Communication equipment, such as wave guide, low noise amplifiers (LNA), high power amplifiers (HPA), signal processing equipment remain stationary within the parabolic dish vertex and the base. They do not obstruct the system in power mode since the dish vertex is shadowed by the receiver during the day anyway (receiver shadowing losses are typically less than one percent).

THE GAS TURBINE POWER PLANT

Thermodynamic layout of the gas turbine process

Fig. 7 shows a modified open cycle gas turbine in the T,s-diagram. Thermodynamic improvement of the process is possible by introducing one or more intermediate heating stages. The diagram shows an example of a single intermediate heating stage (broken line). The process can also be thermodynamically improved by introducing an intercooling stage. The intercooler will lead to higher efficiencies at higher pressure ratios if compared with simple cycles. Since higher pressures will lead to better dimensions of the heat transfer equipment, an intercooler is attractive as long as no problems are encountered when running the plant (e.g. no available cooling water).

To cool the blade roots in the thermally highly stressed first turbine stages and to cool the rotor, cycle gas is used which is tapped from the HP compressor. The cooling gas flow rate is a percentage of the mass flow of the turbine and is assumed to be 1 % in our case. Also, the total relative pressure drop $\varepsilon = \Delta p/p$ is chosen

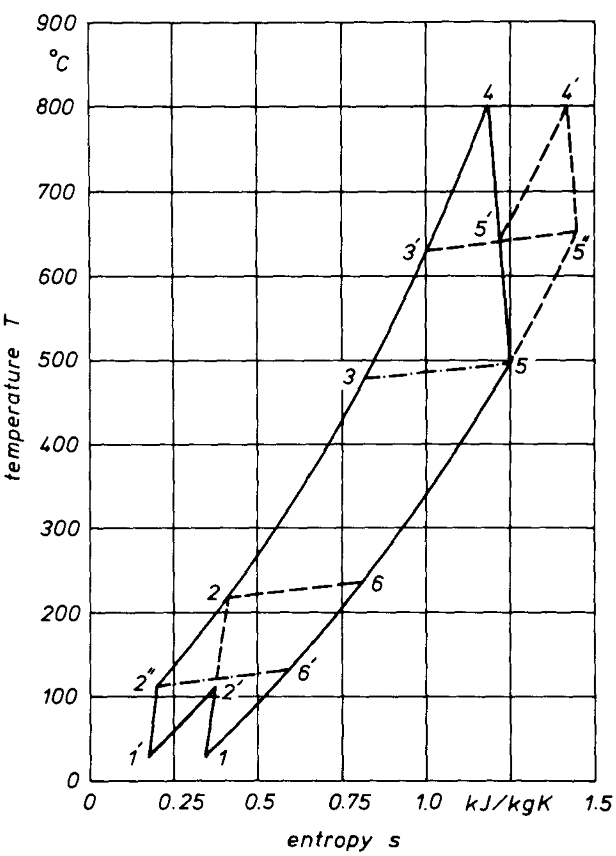


Fig. 7 T,s-diagram for the gas turbine plant

Table 2 Component efficiencies for gas turbo converters

net power	Component efficiency η (%)						
	compressor		turbine	mechanical	gear-box	generator
kW	LP	HP				
25	76	75	84	96	92	85
50	80	79	85	98.5	94	88
100	82	81	86	99	95	90
250	84	83	88	99	96	93
500	85	84	89	99.2	97	95
2 000	86	85	89.2	99.3	97.5	96
20 000	87	86	90	99.7	98.5	98

according to the present state-of-the-art and amounts to 9 %. This value is the sum of the relative pressure drops in the heat exchanging equipment and the piping system.

In addition to the layout parameters for the turbine, the compressors and the heat exchanging equipment, the mechanical and electrical efficiencies of the plant must be defined. For cycle calculations, Table 2 gives the different component efficiencies of the gas turbine set for different plant capacities, based on existing experience and state-of-the-art. For a maximum working medium temperature of T_4 = 800 °C and based on the efficiencies given in table 2, Fig. 8 has been obtained taking into account the real gas behavior. In the case without intercooling, with ambient temperature T_1 = 30 °C, a total relative pressure drop of ε = 8 % (ε = 9 % with intercooling) was assumed. The end temperature difference for the recuperator is taken as ΔT = 40 °C.

Since the parabolic dish collector, with guiding and control equipment, constitutes a large portion of the cost of the plant, it is a general procedure to choose the optimum efficiency point of the gas turbine plant as the working point. This is also the case in Fig. 8, where the optimum efficiency point for

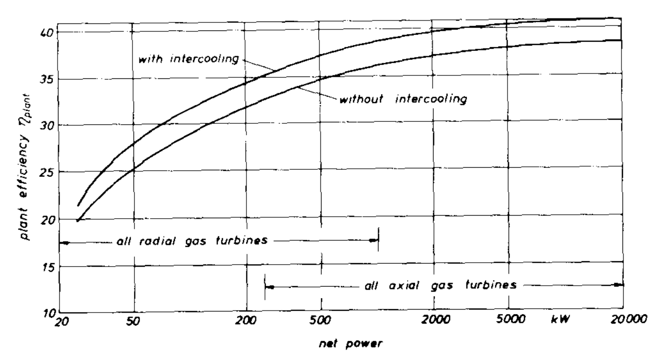


Fig. 8 Plant efficiency versus output for gas turbines with recuperator
T_1 = 30 °C T_4 = 800 °C ΔT = 40 °C

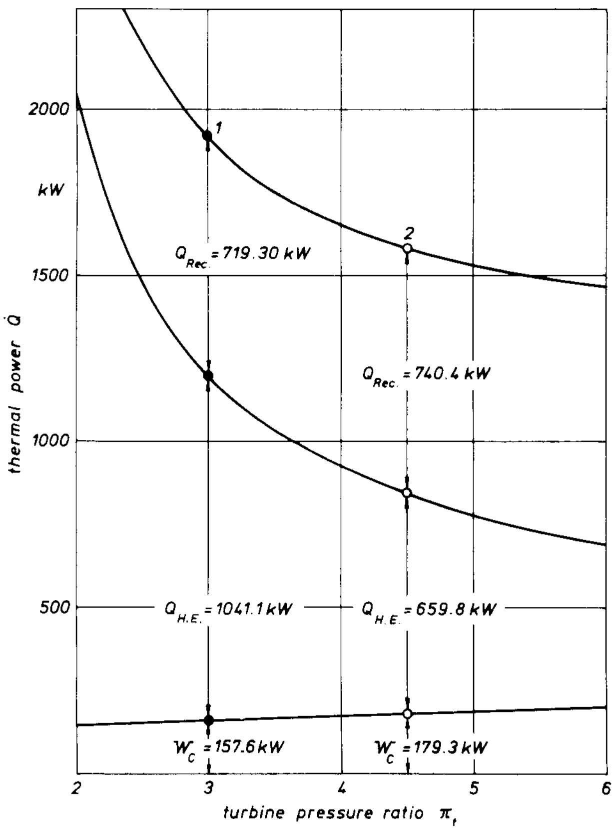


Fig. 9 Heat balance for a 250 kW gas turbine plant

each plant is plotted against the net power. However, this optimization procedure considers only the dish area which is inversely proportional to plant efficiency [11]. Accordingly, the size of the receiver and the recuperator must not be optimized as long as efficiency is the only criterion for selection. The above statement will be explained on the basis of calculations done for a 250 kW solar gas turbine power plant. Fig. 9 shows a heat balance for a 250 kW plant, where the thermal power (processes 1', 2", 3, 4, in Fig. 7) is plotted versus the turbine pressure ratio. Since intercooling is used, the turbine pressure ratio is generally higher than that without intercooler (π_t = 3 for optimum plant efficiency of 35.1 %). If we go to still higher pressure (point 2 in Fig. 9 with π_t = 4.5 and η_{plant} = 34.1 %) it will be observed that the heat transfer in the recuperator $Q_{H.E.}$ is reduced by approximately 37 % than in point 1 with optimum efficiency. A samller heat exchanger is very important for such a plant, since for such a power range the unit is located above the receiver and the entire unit (turbine, recuperator and receiver) moves with the dish. In spite of the fact that thermal power transferred in the receiver is fractionally more for point 2 than point 1, the size of the receiver will not be larger because of the better heat transfer coefficients at the higher pressure of the working fluid (point 2).

DETAILED LAYOUT OF THE CYCLE COMPONENTS

Turbomachinery

When designing smaller solar power plants (up to 500 kW), the component with the major design problem is the turbine, ecpecially if the working fluid is steam [12,13]. For this power range the open-cycle gas turbine offers a better solution, since the overall dimensions of the turbomachinery and the prevailing blade heights could lead to still reasonable efficiency values.

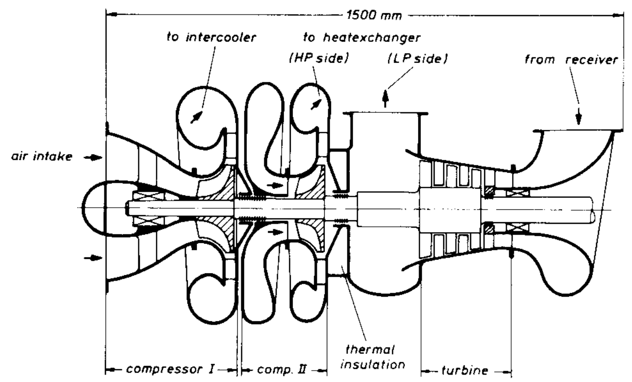


Fig. 10 250 kW solar air turbine set

Since intercooling is selected, we should have two compressors. For volume flow rates and pressure ratios treated here, only centrifugal compressors will be attractive. On the other hand, the turbine should be a multistage axial one to obtain high efficiency. If radial turbines are used, then we should use at least two turbines and the design of the plant and the obtained efficiency will not be any more satisfactory. For such a selection the turbomachinery set will be as shown in Fig. 10. If no cooling water is available, then the plant should be designed with no intercooling and only one compressor. Since the pressure ratio for the turbine will be smaller, a centripetal turbine may be used instead of axial turbine.

The design procedure which is given below for a 250 kW solar gas turbine is only one example and the same procedure may be used to design turbines and compressors for 50, 100 and 500 kW solar gas turbine power plants. More details for the design procedure may be found in [1, 7].

The turbine

The first step in the layout of axial-flow turbomachinery relates to the calculation of the main dimensions. The following data are known from the layout of the cycle: pressure at inlet and outlet (p_i, p_o), temperature at inlet (T_i), rated power, mass flow rate (P, $\dot{m}$) and isentropic efficiency of turbine (η_t). A series of optimizations have been carried out which lead to the following data: speed 27 500 < rpm < 32 000, and the mean diameter 0.18 < d_m < 0.2 m.

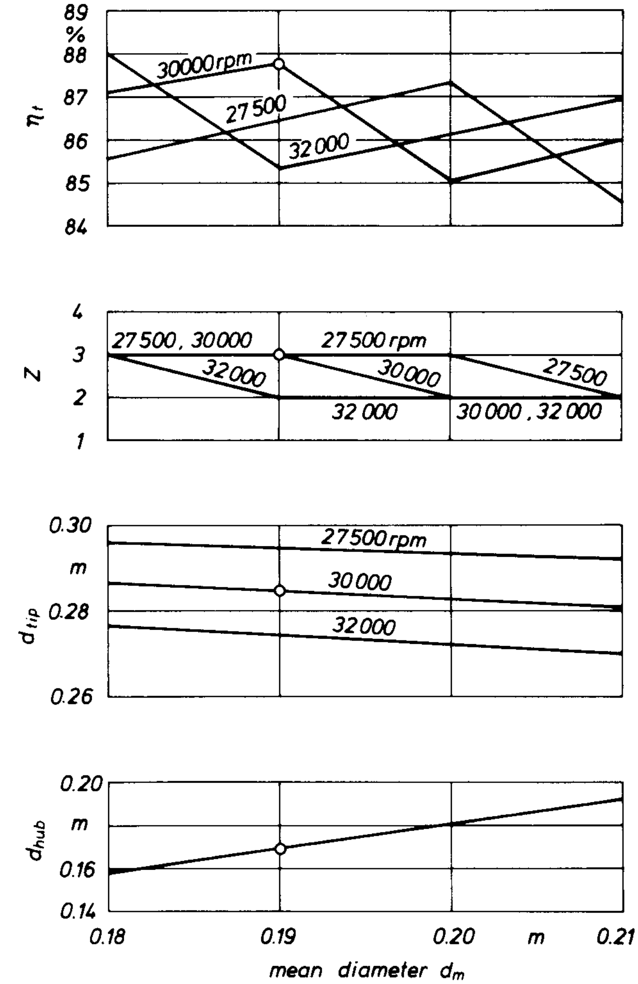


Fig. 11 Turbine design parameters versus mean diameter for different speeds

○ design point

For this design range the turbine blade height and the efficiency are acceptable. Fig. 11 shows how the turbine efficiency η_t, the number of stages Z and the hub and tip diameter (d_{hub}, d_{tip}) of the turbine vary with the mean diameter d_m. The sharp change in the efficiency is due to the sudden decrease of the number of stages, which means high loading and hence lower efficiency. From Fig. 11 it is obvious that 30 000 rpm and 0.19 m mean diameter will lead to good design parameters and is, therefore, chosen as our design point.

The compressors

Centrifugal compressors have been chosen because the mass flow rate is rather small. In the case of axial flow compressors this will lead to short blades at the last stage with rather thick boundary layers. Accordingly, the efficiency will not be much higher than that of a centrifugal compressor. However, the length of the centrifugal compressor will be much less, which means a compact plant (Fig. 10).

The centrifugal compressor may be either of the backward bladed type or radial type. Since the blades are radial, the radial bladed impeller is advantageous, as long as strength is a criterium. On the other hand, the backward bladed impellers have better efficiencies. It will be demonstrated that the assumed compressor efficiencies can be achieved only with backward bladed impellers.

A series of optimizations have been carried out to design the two stages of the centrifugal compressor. Fig. 12 shows for a speed of 30 000 rpm (identical with turbine rpm), the compressor design parameters. The inducer hub-tip ratio ν_1, the peripheral speed u_2 and the compressor efficiency η_c are given as a function of the inlet flow relative angle $\beta_{1,tip}$ and for different flow coefficient φ_2 defined as c_{m2}/u_2. The target efficiency of 84 % can be achieved in case of the backward bladed impeller with blade angle $\beta_{2,\infty} = 60^o$. On the other hand, for the impeller with $\beta_{2,\infty} = 90^o$ only 80.5 %

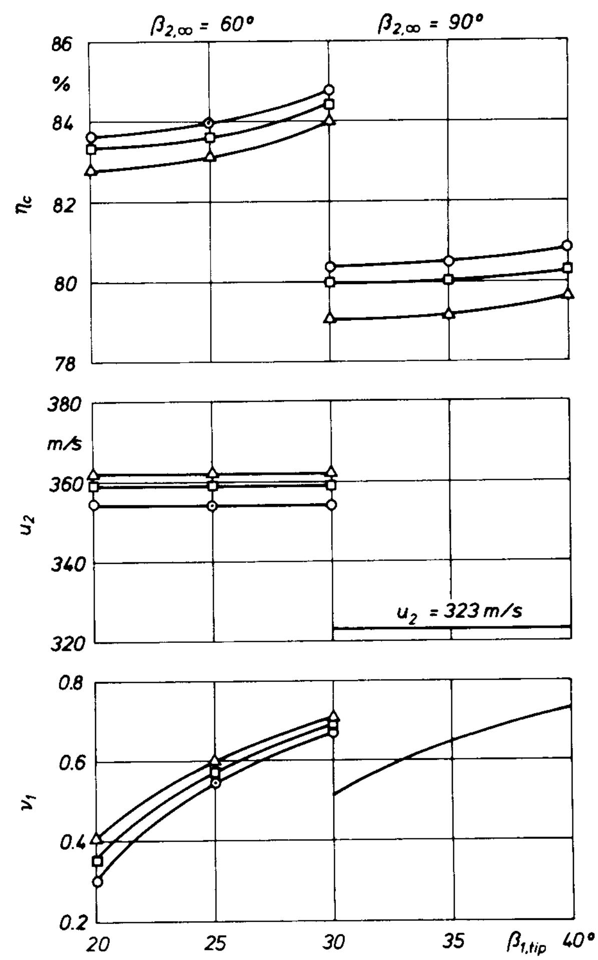


Fig. 12 Compressor design parameters for different flow coefficients

⊙ design point
○ $\varphi_2 = 0.26$
□ $\varphi_2 = 0.30$
△ $\varphi_2 = 0.34$

efficiency can be expected. Therefore, the backward bladed impeller is chosen and the design point is indicated in Fig. 12.

Heat exchanging equipment

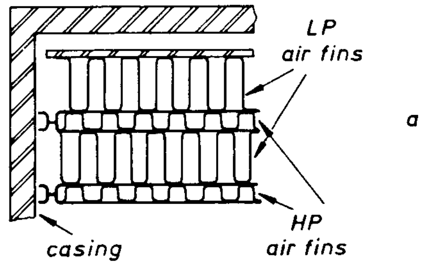
The heat exchangers of an open-cycle gas turbine generally comprise the following units: Intercooler, recuperative heat exchanger and heater (receiver). In the heat exchanger and in the intercooler the heat to be exchanged is transferred from a high-temperature flowing fluid to a low-temperature flowing fluid. Heat absorption by a receiver in a solar power plant differs in such a way from a heat exchanger that a surface is irradiated, thus transferring heat to the working fluid.

The intercooler

The intercooler has the function to reduce the inlet temperature to the second stage of the compressor and hence the total compressor work will be less for the same pressure ratio. Also, the optimum pressure ratio will be higher than without intercooling which leads to better heat transfer coefficients in the receiver and hence smaller dimensions. However, since smaller volume and weight for the intercooler (if any) is important, compact designs such as plate heat exchangers should be used [1, 7].

The heat exchanger

To minimize the pressure losses in the piping system it is advantageous to locate the entire gas turbo converter at the top of the parabolic dish. This implies a compact design for the heat transfer equipment and the turbomachinery. To cope with these requirements a compact recuperator with high effectiveness as described in [14, 15] is chosen. Fig. 13 shows two types compact heat exchangers: the first one is of the plate and fin type (a) and the other is of the herringbone corrugated metal sheet plate type (b). With such designs the heat transfer area A for a certain volume will be several times more than a conventional design. The heat transfer conductance ($k \cdot A$) changes with the effectiveness, where k is the overall heat transfer coefficient. Most conventional heat exchangers have an effectiveness E of approximately 80 %. This is because increasing the effectiveness from 80 to 90 %, the conductance ($k \cdot A$) is increased by almost 100 % [1, 14]. This means doubling the heat transfer area which would imply an expensive and bulky heat exchanger. With the chosen type of heat exchanger, however, the total volume will still be much less than the conventioanl type, if $85\,\% < E < 90\,\%$ is chosen. For the recuperator of a 250 kW gas turbine power plant with intercooling the following values have been obtained: effectiveness 87 %, overall heat transfer coefficient 36 W/m^2K, heat transfer area 300 m^2. The hydraulic diameter for the fin and plate type heat exchanger is estimated to be 4.6 mm (HP) and 10 mm (LP) which results in an estimated volume of 0.5 m^3.



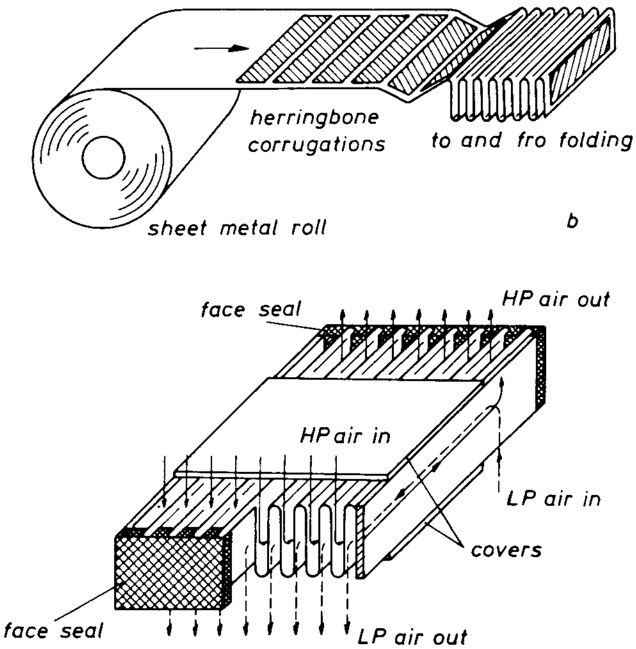


Fig. 13 Compact type heat exchangers

a	plate and fin type	
b	herringbone corrugated metal sheet plate type	
	top:	manufacturing method
	bottom:	method of working

The receiver

Probably the most important component of a solar thermal power plant is the receiver. Basically, the receiver is a heat exchanger which transfers heat absorbed from sunlight (incident solar radiation) to a working fluid flowing in passages or tubes. The receiver design concept is the decisive question for high plant efficiency and cost-effectiveness. Receiver technology and optimization will play an important role in the development of future solar thermal power plants [16].

The high concentration ratio (C = 5000 and more) of the LPDS and its uniform heat flux density require very sophisticated receiver designs to withstand the resulting extreme temperatures. Two systems with air as a heat transport medium in two different temperature ranges are considered. The first is an alloyed steel tube receiver which withstands process temperatures up to 950 °C as an immediate solution. Detailed designs for such a receiver are under way. The second is a ceramic receiver with refractory materials which allow temperatures in the range of 1200 - 1400 °C as a solution in the near future [17].

For the 250 kW demonstration plant, a conventional metal tube receiver is being considered. This technology has been employed for years in gas-, oil-, and coal-fired heaters for open- and closed-cycle gas turbines. In the last 25 years extensive theoretical [18, 19] and experimental work [20, 21] has been done for the development of conventionally fired air heaters. We have running experience with several plants in Germany [22, 23] with a lifetime for each plant with more than 120 000 operating hours. This running experience warrants the use of this high temperature technology to be transferred into the receiver of solar thermal power plants. The radiation part of such heaters corresponds to the receiver. The place of the burner (coal, oil, gas) is taken by the incident solar radiation.

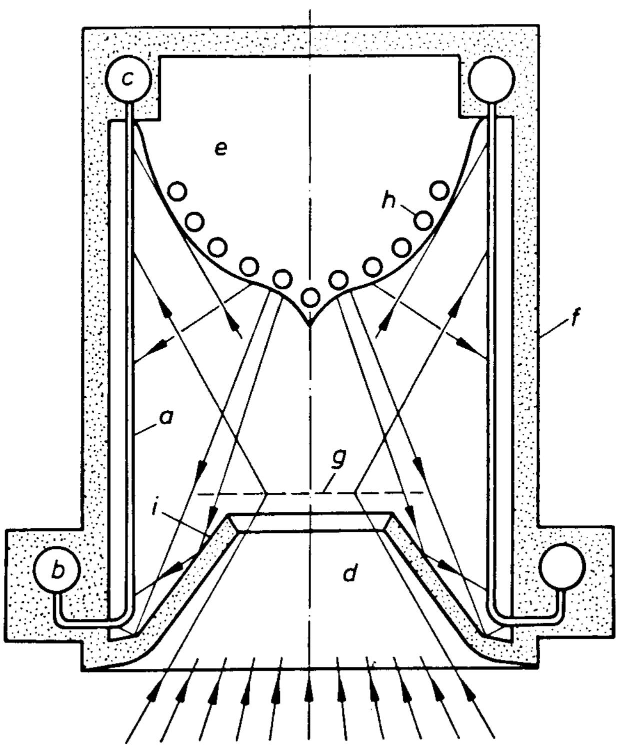


Fig. 14 Diagrammatic sketch of receiver configuration for LPDS

a heater tubes
b inlet header
c outlet header
d aperture
e radiation distribution cone
f receiver cage with insulation
g focal plane
h cooling tubes
i reflective wall

Fig. 14 shows a diagrammatic sketch of a receiver for a solar dish collector. It consists mainly of the heater tubes a, the inlet and outlet headers b and c, the aperture d, the radiation distribution cone e, and finally the receiver cage with insulation f. Solar radiation concentrated by the dish enters the aperture and falls onto the cone, which has the function to reflect them in a certain mode on the heater tubes. The position of the focal plane g is determined by the design point. The cone is generally made of a ceramic material and should be cooled through cooling tubes h. The mode of radiation and reflection is shown in Fig. 14. The aperture wall may be also used to increase the concentration of the heat flux of the tube entry part. The aperture (windowed or open) has to be placed in front of the focal plane with a suitable radiation distribution cone to remain within present-day material strengths and tube temperatures.

A major factor for choosing a parabolic dish collector tracking continuously on the sun is the uniform energy distribution in the focal plane (heat flux distribution) throughout the day. When the parabolic dish collector follows the sun from sunrise to sunset, only the focal plane intensity profile changes due to varying insolation values (morning, noon, evening) but the heat flux distribution over the focal plane remains constant throughout the day (the sun is always held on the optical axis of the dish), the most important consideration for the construction of efficient receiver design and cannot be fullfilled by central receivers of solar towers using heliostat mirrors.

In the calculations for the 250 kW demonstration plant receiver, the characteristics of the heat flux absorbed by the working fluid were varied over the tube length. The tube material is a high temperature steel capable of withstanding tube wall temperatures of up to 950 °C. Optimization studies have shown that neither the heat flux (radiation energy density) along the heater tubes can be kept constant, nor can the maximum tube wall temperature be maintained along the entire length of the individual heater tube [24].

In the first case (constant heat flux along tube length), the austenitic tube material is not used to its optimum tube stresses and a much too large and heavy, thus expensive receiver is the result. In the second case (constant maximum tube wall temperature) one obtains a receiver with minimum dimensions and weight. However, the tube material is being overloaded and this will cause the tubes to rupture after a short lifetime at the entrance of the heater tubes.

The optimum case three is calculated for the maximum stress σ_{max} and the corresponding temperature for any section along the tube length, where the allowable stress σ_{al} is obtained from the creep properties of the material at this temperature. The maximum stress σ_{max} is the combined stress resulting from the three dimensional thermal and pressure loading (v. Mises-Theorem [25]). The ratio of the above mentioned stresses is the stress coefficient $\beta = \sigma_{max}/\sigma_{al}$ and should be kept constant along the length of the heater tube (Bammert-Criteria [26]). The main task of a receiver designer is to find the flux distribution which corresponds to the criterion $\beta = 1$ for the whole tube length.

Fig. 15 shows the heat flux and temperature distributions versus dimensionless tube length for the requirement of $\beta = 1$ (curve a). Curve b gives the heat flux distribution along the length, while curve c is the mean heat flux. Curve d is the maximum wall temperature along the tube length and curve e shows the temperature of the working medium. The curves are based on a tube outer diameter d_o = 18 mm, a spacing ratio t/d_o = 2.3, and a wall thickness ratio s/d_o = 0.11. Table 3 gives the main design data of the 250 kW receiver.

To demonstrate the importance of using an intercooler, the receiver for the plant without intercooling is also calculated.

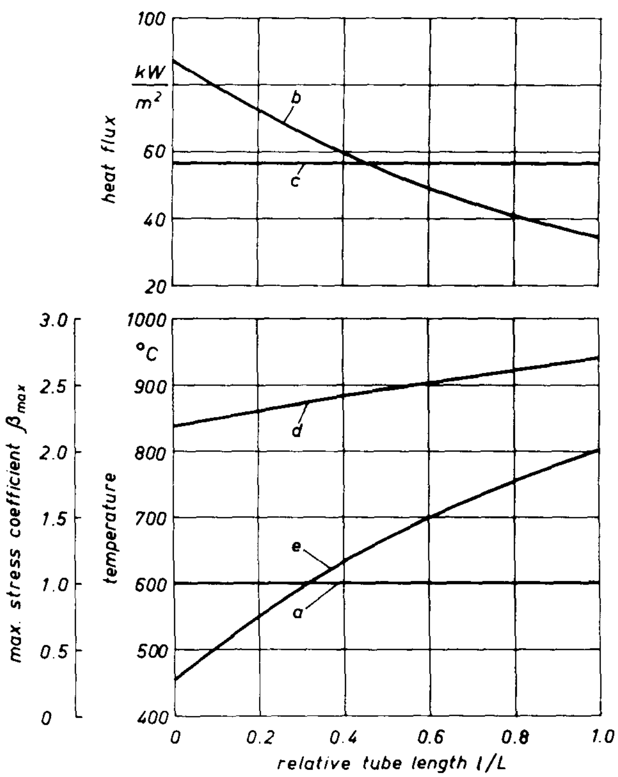


Fig. 15 Heat flux, temperature and stress coefficient versus relative tube length

a maximum stress coefficient
b heat flux
c mean heat flux
d maximum tube wall temperature
e temperature of working medium

Table 3 Layout parameters for the receiver of a 250 kW LPDS

	unit	with inter-cooling	without inter-cooling
transferred heat	MW	740.7	815.3
mean heat flux	kW/m^2	56.9	47.5
maximum heat flux	kW/m^2	87.8	71.6
mass flow	kg/s	1.9	2.4
receiver inlet temperature	°C	453.8	497.7
receiver outlet temperature	°C	800	800
receiver inlet pressure	bar	4.8	3.7
pressure loss	%	4.5	4.5
maximum tube wall temperature	°C	939.6	941.0
outer tube diameter d_o	mm	18	21
tube wall thickness s	mm	2	2
tube length L	m	1.81	1.98
pitch diameter D	m	1.67	2.01
spacing ratio t/d_o	-	2.3	2.3
length to diameter ratio L/D	-	1.08	0.99
number of tubes	-	127	131
heating surface	m^2	13.0	17.1
mass of tubing	kg	185.3	248.2
cavity volume	m^3	3.96	6.3

The working points have been chosen at higher pressure ratios than the ones corresponding to optimum plant efficiencies of 35.1 % (with intercooling) and 32.4 % (without intercooling). The penalty in efficiency was 1 and 1.4 points respectively. Table 3 also shows the design values for the receiver if intercooling between the compressors is not present. It is interesting to notice, that the cavity volume is 3.96 m^3 with intercooling compared to 6.3 m^3 for no intercooling. In addition the mass of tubing is approximately 34 % less for an intercooled gas turbine set.

ARRANGEMENT OF THE SOLAR POWER PLANT

The parabolic dish

To produce an electric output of P_{el} = 250 kW, thermal power $P_{th} \approx$ 750 kW has to be transferred to the working medium based on a plant efficiency of 0.33. The dish panels should reflect an incident power $P \approx$ 1255 kW based on optical efficiency 0.7. With a maximum direct solar radiation intensity of 0.85 kW/m^2 on the panels, a dish diameter of 40 m is necessary.

The gas turbine plant

For a simple and robust solar dish power plant, the gas turbine set (50 - 500 kW) should be placed above the receiver as a compact power unit moving with the dish tracking the sun. For larger power ranges, the gas turbine set may be put at the vertex of the dish. A special hot gas duct may be installed inside the struts supporting the receiver to carry the air from the gas turbine to the receiver and vice versa. Also, a secondary reflector may be used and the receiver is situated in such a case between the secondary reflector and the vertex of the dish. In this case the gas turbine and/or the receiver act as a counter weight for the dish.

Fig. 16 shows an arrangement of the receiver a and the gas turbine set. The recuperator b is placed directly above the receiver. The turboset c and the generator d arranged axially with the dish to minimize the gyroscopic effects. In case of intercooling, the intercooler e is put next to the turboset. A gear box f transmits the power from the turboset to the generator. The receiver and the turboset are supported by three struts g, also shown in Fig. 1. For the 250 kW solar plant (40 m Ø dish) the

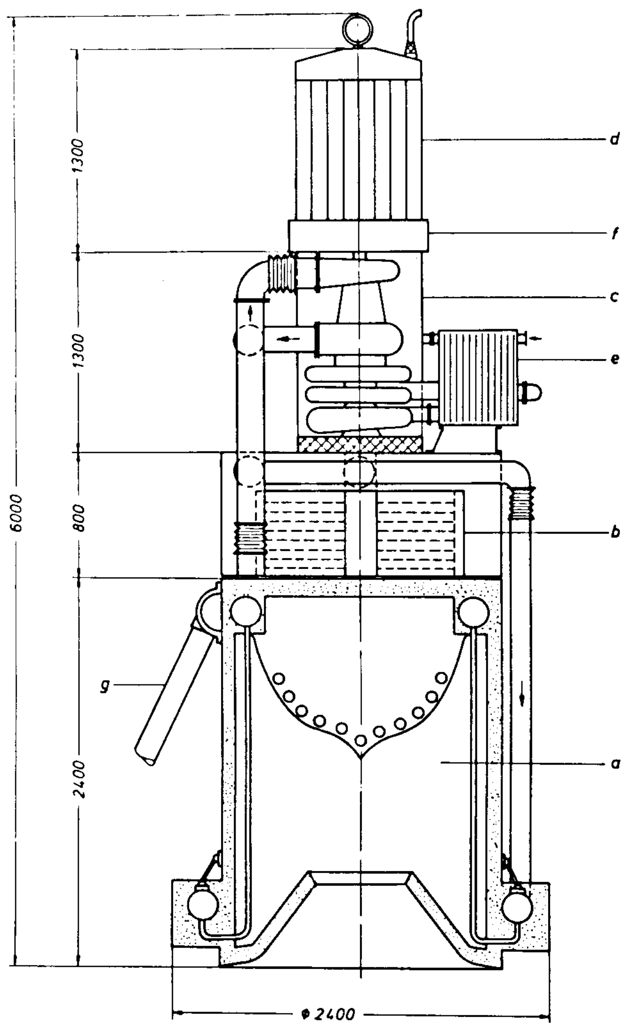


Fig. 16 Receiver and gasturbo-converter set for a 250 kW plant

dimensions in mm

a receiver
b recuperator
c turbo set
d generator
e intercooler
f gear box
g struts

main dimensions are indicated in Fig. 16. The total height is 6 m and the largest diameter of the system (causing the shadowing effect) amounts to approximately 2.4 m.

SUMMARY

A cost and efficiency comparioson for the different solar power plants has shown that the Large Parabolic Dish Systems (LPDS) with a turbo converter is superior to other solar thermal power plants, especially in the power range of 50 to 2000 kW. Due to uniform heat flux distribution in the focal plane the receiver can be optimized to follow the Bammert-Criteria for optimum utilization of the tube material.

An optimization procedure for the layout and component design of a 250 kW gas turbine plant was presented. Cycle optimization has been carried out for a plant with intercooling and no reheat. Reheat was excluded to keep the design of the receiver (especially in this phase) as simple as possible. If cooling water is available intercooling is recommended. This is because it increases the pressure ratio of the compressors and furnishes better heat transfer coefficients for the receiver and the heat exchanging equipment. Therefore, not the best efficiency point has been chosen (as usual) but a point of higher pressure ratio is selected as the design point. The penalty in the efficiency was one point, however, the reduction in size and cost of the heat exchanger and the receiver are much more pronounced. Also, compact heat exchanger units with high effectiveness were chosen to keep the total volume and weight of the plant small, so that it may be possible to install the gas turbine on top of the receiver and the entire power generation set moves with the parabolic dish.

When not in power conversion mode, this proposed dish system can easily be used for various kinds of communication and data transmission as an antenna mainly during the night. Therefore, a 24 h use of the LPDS provides a further reduction on specific installation cost, pay back time and a communication link to remote and rural areas.

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