DOCTORAL THESIS

Novel and Ancient Technologies for Heating and Cooling Buildings

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Luleå University of Technology

Doctoral Thesis

Novel and Ancient Technologies for Heating and Cooling Buildings

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ERRATA PAGE

This corrigendum is to precise that the three first paragraphs of page 47 and 48 and in Figure 6.3 of page 48 in section 6.1 have been taken and translated from French from [1].

[1] Thiers, S (2008) Bilans énergétiques et environnementaux de bâtiments á énergie positive. Thèse de doctorat. Ecole nationale Supérieure Des Mines De Paris

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Il faut distinguer pour les décennies à venir le prévisible, le probable et l'incertain, en tenant compte que très souvent l'inattendu arrive
Edgar Morin

C'est une triste chose de songer que la nature parle et que le genre humain ne l'écoute pas
Victor Hugo

Une nouvelle manière de penser est nécessaire si l'humanité veut survivre
Albert Einstein

Agir dès maintenant, c'est l'urgence
Hubert Reeves

Dans la vie, il y a deux catégories d'individus :
Ceux qui regardent le monde tel qu'il est et se demandent pourquoi.
Ceux qui imaginent le monde tel qu'il devrait être et se disent: pourquoi pas ?
George-Bernard Shaw

In memory of Professor Torbjörn Jilar,
SLU, Alnarp, Sweden

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ABSTRACT

The basic issue of this thesis concerns one of the fundamental problems of the future of our society: How to meet the energy requirements for a large and growing world population while preserving our environment? This question is important for the world and the answers are complex and interwoven.

Conventional energy sources, fossil and fissile, are polluting in the present and in the future: they erode the environment and their resources are limited. Renewable energy (hydro, wind, solar, geothermal) constitutes a minimum of pollution in the different energy systems. The technologies for using renewable energy are well known though further development and progress are made. This development also requires behavioural change, adaptation, and above all political will. The transition from an economy based on fossil energy to an economy based on renewable energy appears necessary for the protection of the environment. The cost of renewable energy is often represented as an obstacle but remains competitive in the long run.
The development and availability of renewable energy, which varies because of its spatial and temporal distribution, require an adaptation of lifestyle, habits, habitat design (passive bioclimatic houses), urban planning and transportation.

The focus of this thesis was to apply renewable energy in an area with hot summers and cold winter, a climate like that in the northwest of Algeria. In order to provide improved comfort in the buildings and also economic development in this area, the energy demand for heating and cooling was analyzed in the ancient city of Tlemcen. To supply domestic hot water and space heating, water must be simultaneously available at two different temperature levels. Cold water temperature, close to that of the atmosphere, and hot water between 50 and 60°C. An interesting feature of the preparation of hot water is the small variation of requirements during the year, unlike that to heating. The preparation of hot water is one of the preferred applications of solar energy in the building for several reasons. For this reason an experimental study of the thermal behaviour of a domestic hot water storage tank was undertaken. The phenomena that affect the thermal behaviour of tank especially the coupling between the solar collector and storage tank was studied. This study included concentrating solar collector in which optical fibers were used to transport the energy to the storage tank. Another technology was introduced and developed for the heating and cooling of buildings in the desert involving an existing ancient irrigation system called Fouggara. The novel idea is to use the Fouggara as an air conditioner by pumping ambient air through this underground system. Then air at a temperature of about 21°C would be supplied to the building for heating in the winter and cooling in the summer. This study shows the feasibility of using this ancient irrigation system of Fouggara and contributes to reducing and eliminating the energy demand for heating and cooling buildings in the Sahara desert.

Key words: Heating/cooling, design, solar energy, simulation, optimization, modelling, optical fiber, concentration, Fouggara.

RESUME

La question fondamentale de cette thèse concerne l'un des problèmes fondamentaux de l'avenir de notre société: Comment répondre aux besoins énergétiques d'une population mondiale importante et croissante tout en préservant notre environnement? Cette question est importante pour le monde et les réponses sont complexes et imbriqués.

Sources d'énergie conventionnelles, fossiles et fissiles, sont polluantes dans le présent et l'avenir: ils érodent l'environnement et leurs ressources sont limitées. Les énergies renouvelables (hydraulique, éolienne, solaire, géothermique) constitue un minimum de pollution dans les différents systèmes énergétiques. Les technologies pour l'utilisation des énergies renouvelables sont bien connues mais encore le développement et le progrès sont réalisés. Ce développement nécessite également un changement de comportement, l'adaptation, et surtout une volonté politique. La transition d'une économie basée sur les énergies fossiles à une économie basée sur les énergies renouvelables apparaît nécessaire pour la protection de l'environnement. Le coût de l'énergie renouvelable est souvent représenté comme un obstacle mais reste compétitif sur le long terme. Le développement et la disponibilité des énergies renouvelables, qui varie en raison de sa distribution spatiale et temporelle, nécessitent une adaptation du mode de vie, les habitudes, la conception de l'habitat (maisons passives bioclimatiques), la planification urbaine et le transport.

L'objectif de cette thèse était d'utiliser les énergies renouvelables dans une zone avec des étés chauds et des hivers froids, tel le climat dans le nord-ouest de l'Algérie. Afin d'apporter une amélioration du confort dans les bâtiments et le développement économiques dans ces secteurs, la demande d'énergie pour le chauffage et le refroidissement a été analysée dans l'ancienne ville de Tlemcen. Pour produire l'eau chaude sanitaire et le chauffage, l'eau doit être disponible simultanément à deux niveaux de température différents. Basse température, eau froide à température proche de l'ambiance; et l'eau chaude entre 50 et 60°C. La préparation de l'eau chaude sanitaire constitue une des applications privilégiées de l'énergie solaire dans le bâtiment pour plusieurs raisons. La première caractéristique intéressante est la faible variation des besoins au court de l'année, contrairement au chauffage. Les phénomènes qui affectent le comportement thermique du réservoir en particulier le couplage capteur solaire et ballon de stockage ont été étudiés. Cette étude comprend un capteur solaire à concentration dans lequel les fibres optiques ont été utilisés pour le transport de l'énergie jusqu'au réservoir de stockage. Une autre technologie a été introduite et développée pour le chauffage et la climatisation des bâtiments dans le désert impliquant un ancien système d'irrigation appelé Fouggara. L'idée originale est d'utiliser la Fouggara comme source d'énergie en pompant l'air ambiant à travers ce système souterrain. L'air à une température d'environ 21°C serait fourni à l'édifice pour le chauffage en hiver et refroidissement en été. Cette étude montre la faisabilité d'utiliser ce système d'irrigation antique de Fouggara et contribue à réduire et à éliminer la demande d'énergie pour le chauffage et le refroidissement des bâtiments dans le désert du Sahara.

Mots clés: chauffage / refroidissement, conception, énergie solaire, simulation, optimisation, modélisation, fibre optique, concentration, Fouggara.

Thesis Outline

This thesis is presented as the partial fulfilment of the requirements for the degree of Doctor of Philosophy (PhD) in Water Resources Engineering. The research was carried out at:

- The Division of Architecture and Water, Luleå University of Technology, Sweden.
- The Renewable Energy and Materials Research Unit, Faculty of Sciences, Tlemcen University, Algeria.
- The Renewable Energy Research Unit in Saharan Middle, Adrar, Algeria.

This thesis is articulated into an experimental study of a new conception system for high efficiency solar heat production using the Fouggara system and hot water energy storage for buildings.

The thesis consists of a short introduction and the following papers:

I. **Amara. S, Virgone. J, Bouia. H et Krauss. G,** (2005), Stratification dans les ballons d'eau chaude sanitaire: Comparaison de modèles existants, 12ème Journées Internationales de Thermique (JITH), 15-16-17 Novembre 2005, Tanger Maroc

II. **Amara. S, Zidani. C, Benaissa. D, Benyarou. F, Benyoucef. B,** (2006), Diurnal and nocturnal temperatures modelling for Tlemcen site, Physical and Chemical News, Volume 27, 59-64.

III. **Amara. S, Virgone. J and Benyoucef. B,** (2007), Etude d'un système de production et stockage d'eau chaude sanitaire pour le site de Tlemcen, Comples heliothechnique, 36B, 3-7.

IV. **Amara. S, Benmoussat. A, Bouguima. S, Benyoucef. B, Nordell. B,** (2007), Requirements energy estimate of heating & cooling for a dwelling in the site of TLEMCEN, Revue des Energies Renouvelables, Numéro special, Mai 21-24, 2007, ICRESD_07, Tlemcen, ALGERIE,

V. **Amara. S, Benyoucef. B, Benhamou. M, Moungar. H and Touzi. A,** (2009), Experimental study of a domestic hot water storage tank thermal behaviour, Physical and Chemical News, Volume 46 (2009).

VI. **Amara. S, Nordell. B, Benyoucef. B, Benmoussat. A,** (2011) Concentration heating system with optical fiber supply, Energy Procedia 6 (2011) p.805–814

VII. **Amara. S, Nordell. B, Benyoucef. B,** (2011) Using Fouggara for Heating and Cooling Buildings in Sahara, Energy Procedia 6 (2011) 55–64

1. Introduction

The oil crisis in 1973 raised awareness in industrialized countries of the precariousness of their energy supplies and the need to focus on energy conservation and new technologies. During less than two years the world oil price tripled (from \$14 to \$42 per barrel) and a few years later in July 2008 it reached almost \$144/bbl. (Price per barrel – official oil rate.).

According to the International Energy Agency (IEA), world oil demand is estimated at 88.3 million barrels/day in 2011. The countries of the OECD (Organisation for Economic Cooperation and Development) are the main focus of consumption in the world and represent nearly 60% of global demand. It is instructive to learn that in 1985 the demand in the world was only 10.7 billion TOE (tons oil equivalent) against 4.5 in 1970 which represents an increase of 138% in 15 years (Entreprises, 1973). In 2008 it reached 15.35 billion TOE, as illustrated in Fig.1.1. (IFP, Samuel.B).

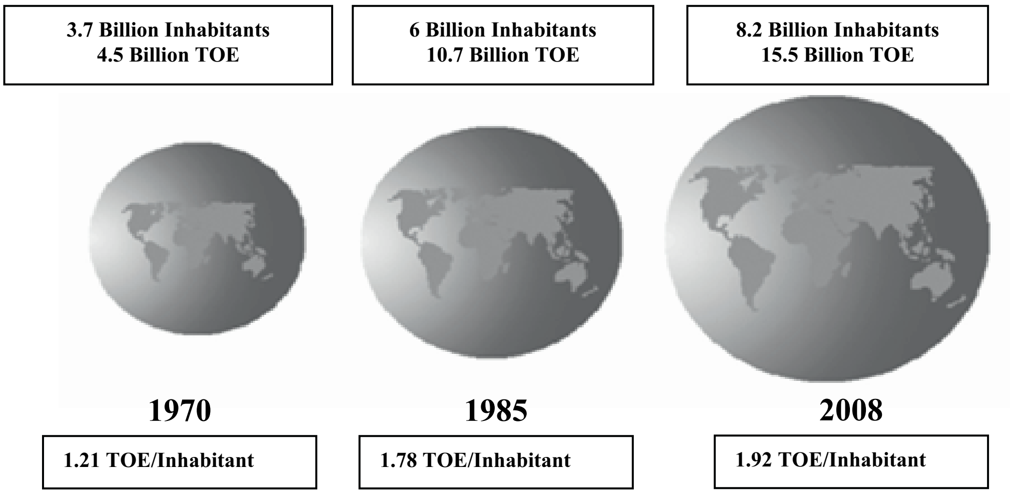


Fig. 1.1 Population growth and global energy demand
(Source : IEA) (Rojey, 2008)

Since 2004, operators in the oil markets have been concerned about the fluctuating demand and supply of oil and the strong trend of increasing prices. The current oil shock (2008) is different from other oil shocks because of its magnitude and duration. The oil shocks in 1973 and 1979 were supply shocks while the current crisis is a demand shock (Kalinowski, 2008).

The evolution of oil consumption is structurally related to economic growth with climatic factors having an influence in the short term. Moreover, the flexibility of demand is becoming weaker.

To address this energy crisis, it is necessary to devise and develop a more efficient use of energy and the replacement of fossil fuels with alternative energy sources. Renewable energies that fall within this context, including reduced consumption of fossil fuels, have lead to other models of energy production, effectively contributing to sustainable development.

The transition from coal to oil and gas in the industry has reduced emissions of carbon into the atmosphere. However, slowly and inexorably, the rate of carbon dioxide continues to increase steadily in the atmosphere. This increase has raised fears of irreversible climate change (global warming) and it is internationally accepted that we have to reduce the greenhouse gas emissions in order to slow down the climate change (Bernard et al. 1983).

The main challenge is found in the sectors of transport, industry and housing, which account for 90% of emissions of greenhouse gases (Fig. I.3), (ECOHABITAT, CITEPA).

For this reason, we will review the energy options for a sustainable future, focusing on options that could reduce or better limit the level of carbon dioxide emissions in buildings.

1.1. The building

Regarding habitat, enhancing the thermal regulation and the emergence of effective insulation materials has already reduced energy consumption from 372kWh/m²/year in 1973 to 245kWh/m²/year today (2007) in France. However, much remains to be done to achieve the performance of low-energy buildings and bioclimatic homes, also known as passive or positive energy or sustainable buildings (Sabonnadière and Quenard (2007), Maugard et al. (2005)).

The best way to save energy in new housing, while retaining a significant degree of comfort for the resident is to define a basic concept based on two main criteria: the bioclimatic parameters of comfort and an adequate system of construction.

1.2. Renewable energy in Algeria

In order to build energy efficient housing, identification of the best environmentally and economically viable energy solutions is essential. Today, all eyes are moving to renewable energy (RE) and to the recovery and recycling of waste for energy.

Renewable energies are involved in the security of our energy supplies and contribute to a better balance in regional planning. They do not generate waste and emissions and are environmentally benign.

Algeria is ranked third in the Mediterranean in terms of capacity of renewable energy. Its potential is estimated at 10 times of the annual world consumption (ECOMED21). The techniques, materials and processes related to renewable energies, are very numerous and have higher levels of technological advancement, price and performance.

1.3. Solar radiation

Because of its location Algeria is very favourable for the utilization of solar radiation. The duration of sunshine over almost the entire national territory exceeds 2000 hours annually and can reach 3370 hours in the Sahara (Adrar). The energy received daily on a horizontal surface of $1m^2$ is about 5 kWh in most of the country or near 1700 kWh/m^2 year in the north and 2250 kWh/m^2 year in the south of Algeria. (MEM-Algeria, Encarta (2008)).

1.4. Wind energy

There are wind turbines of high power (1.2 to 5 MW) that are connected to the electrical grid, but also smaller wind turbines that supply electricity to detached houses. The turbines operate at full power from 2000 to 3000 hours per year, or about 1/3 of the time. Wind maps show that the wind velocity is low in the north (1.5m/s in Algiers) while it is considerably greater in the south (6.3 m/s in Adrar). (MEM-Algeria, Remund and Kunz (2004)).

1.5. Hydropower

Worldwide hydropower accounts for about one quarter of the total production of electricity. This is the main source of electricity in many countries such as Norway (99% of its energy supply), the Democratic Republic of Congo (97%) and Brazil (96%). The plant of Itaipu in Brazil has the largest production capacity in the world (12 600 MW). Among the largest facilities, we can also cite the Grand Coulee Dam in the United States (7 000 MW) and Great Canadian with 7 300 MW (Lako et al. 2003).

In Algeria, there are many dams built for small scale hydro. The capacity of these plants varies between 200 and 300 MW/year according to the intensity of rainfall (MEM-Algeria).

1.6. Geothermal

Geothermal energy meant for a long time high temperature geothermal heat as reservoirs of steam or hot water or hot rocks. Such sources could be used for electricity production or as a direct source for district heating.

Because of the technological development during the last decades on thermal energy storage and heat extraction from the ground the term “geothermal” is now divided into deep and shallow geothermal energy. Still, available data is confusing since “installed geothermal power” sometimes refers to electricity production by hot steam and sometimes by low temperature heat for indoor heating. A heat pump is normally used to raise the water temperature to a suitable level.

When the geothermal reservoir is in a moderate temperature, it is used to generate heat in turn distributed by a pipeline network. The world production of geothermal energy (electricity) was about 5 800 MWe in 1990, over 7900 MWe in 2000, and around 10 700 MWe in 2010 (Bertani, 2007).

In 2007, the United States was the largest producer of geothermal energy, with 2 687 MWe, followed by Philippines (1 970 MWe), Indonesia (992 MWe), Mexico (953 MWe), Italy (810 MWe), Japan (535 MWe). In Algeria more than 200 hot springs have been investigated in the north, including about one third that have temperatures above 45°C, for example in Hammam Meskoutine with a temperature of 98°C which is used to provide heating for greenhouses (Bertani, 2007).

At the end of 2006, the installed electrical capacity for the countries of European Union was 854.6 MWe and the thermal capacity was 9 564.6 MWth, including 7 328.3 MWth of geothermal heat pumps (GEB, 2007).

1.7. Bioclimatic house

The bioclimatic house is a healthy house where one feels comfortable and which fits perfectly within the environmental framework in which one lives. This includes harmonising key elements such as soil, relief, vegetation, orientation, the sun and wind, that is all the natural elements, using simple truths and realities that we have forgotten and that our ancestors reflected. The second focus of the house is bioclimatic architecture; the third is the use of natural materials.

All this leads to a traditional habitat in harmony with the site, climate and the people who will live there. As required, a bioclimatic house offers site specific solutions with renewable energy for example Canadian wells, geothermal, photovoltaic, solar, recovery of rainwater, etc.

To develop these systems, solar homes should be based on three considerations:

• Requirements to be met.
• Available solar (renewable) energy.
• Adaptation the specific site conditions.

2. Objectives and Scope

The overall objective is to study and develop energy-efficient systems for domestic hot water production in climates similar to that of Algeria. The considered energy supply system contains a high-temperature solar collector combined with a buffer tank and an ancient existing irrigation system called Fouggara know used for air conditioning. The objectives are divided into four important sections:

2.1. Characteristic energy load for single-family house in Algeria

In order to utilise naturally stored heat or cold in the ground, seasonal temperature variations are required. In North Africa the mean temperature difference between the coldest and warmest month is about 20°C. In Algeria this difference is even greater, which is favourable. (Grein et al. 2007).

The heating and cooling demand in northern Africa varies considerably in the different countries, with the greatest cooling demand being in Egypt and the greatest heating demand in Algeria. In Section 3, the required heating and/or cooling demand was estimated for a dwelling in the Tlemcen site.

To supply the two heat consumers (domestic hot water and indoor heating), water should be simultaneously available at two different temperature levels. This can be achieved by operating two different storage tanks with a ‘clever’ control unit acting on valves and pumps. However, it is also possible to use a single storage tank if care is taken to avoid mixing water of different temperatures. As hot water has a lower density than cold water, hot water is always located in the upper part of the storage tank; conversely, cold water is found at the bottom. This feature is called stratification of the storage tank.

2.2. Thermal stratification in the buffer tank

The total energy consumption related to domestic hot water is considerable all over the world. The production of hot water is certainly the most interesting application for solar energy systems due to the extensive needs and use throughout the year. Energy used for domestic hot water production corresponds to 40-70% of the households’ energy consumption (ENERGIEPLUS).

The efficient use of solar energy requires the use of accumulator tanks. Efficient performance of such storage tanks means promotion of a stratified water temperature in the tank. Indeed, stratification allows a high temperature at the top, which results in a reduced energy booster. In addition a lower temperature at the bottom of the tank improves the performance of the solar collector. The thermal stratification in the hot water storage tank was therefore paid special attention, see Section 4.

It is necessary to take into account all the phenomena that influence the stratification in solar applications especially the coupling between the solar collector and storage tank. To reach this objective it is interesting to connect the storage tank with a concentrating solar collector using an optical fiber.

2.3. Solar collector and optical fibers for direct solar radiation

Solar energy systems are used throughout the world in particular for heating of water. Such systems are relatively small and we do not see any development on an industrial scale. Some reasons include the high investment cost and the energy losses during transportation to the user. To solve this latter problem, we transport a concentrated solar energy to the place of conversion.

The idea of transmission of concentrated solar energy via optical fibers was put forward in 1980 by a group of French investigators. The technology of optical fibers was developed and industry offers high quality of fibers with large diameter for light transmission (Cariou et al. 1982). In this work we use optical fibers that have the advantage of weak attenuation and small conversion losses.

Another technology was introduced and developed for the heating and cooling of buildings in the desert involving an existing system called Fouggara as an energy source see section 6.

2.4. Fouggara for heating and cooling buildings in Sahara

The idea of using the Fouggara system as an energy source for heating and cooling buildings in the Sahara desert was a result of ongoing studies on Underground Thermal Energy Storage (UTES) systems at Luleå University of Technology. The Fouggara is an ancient water transportation and irrigation technique that was first used in Iran. It was introduced in Algeria desert by Arabs at the time of their conquest of Algeria about the year 700. This underground system is characterized by a constant mean air temperature about 21°C (ABH Sahara). As the temperature in the desert of Algeria (Adrar) varies from 0°C to 47°C this ancient underground duct system can be used as a source for heating, cooling and ventilation, see Fig.6.4. This part of the work involves experiments and theoretical analyses of an existing Fouggara system (El Faiz and Ruf 2006).

Major findings are to be incorporated in a realistic system simulation model to investigate the feasibility of the concentration heating system with optical fiber supply and ancient Fouggara systems. The model can then be used to design the system under various conditions.

3. Heating and cooling demand at the Tlemcen site

The essential points in the building design are directed towards estimation of energy consumption and more particularly the precise prediction of the interior environment. These criteria are not adapted for the evaluation of natural ventilation and the risk of discomfort when the energy consumption is the principal consideration.

The actual design of the building is mainly based on energy economy criteria. It associates insulation, solar contributions and inertia while often omitting the comfort concept that plays two important functions in energy conservation (Norton 1992, Santamouris 2003):

1. The energy consumption of any building is critically dependent on the indoor temperatures which the building and its services must deliver, because the heat lost from the building either through its surfaces or by ventilation depends essentially on the difference between indoor and outdoor temperature. The sensitivity of occupants to the indoor climate also has deep implications for the manner in which the environment is controlled.

2. The building is considered successful only if a comfortable indoor environment is achieved. Thermal comfort is generally regarded by occupants as one of the most important requirements for any building. In addition, there is evidence that the thermal comfort of occupants is closely linked to their perception of indoor air quality (Humphreys and Nicol (2002)) and productivity (McCartney and Humphreys (2002)). The comfort is also important because the occupants will react to any discomfort by taking actions to restore their comfort. These actions may themselves have an energy cost e.g. opening a window when the heating is on, for instance, can be a costly way to cool an overheated building.

3.1. State of the art of thermal comfort

The thermal interaction between humans and their environment is highly complex and has been the subject of a great deal of studies. The internal processes by which we produce and respond to heat and processes of heat transfer between humans and the immediate environment. The study dealing with thermal comfort therefore has to take all these considerations into account (McIntyre 1980; Parsons 2002; Darby and White 2005).

For any construction type, it is necessary to define the conditions within building which will provide a comfortable indoor climate («comfort conditions» or «comfort temperature»). Because some factors influence the heat balance of the human body, including humidity and air movement in addition to the radiant and air temperatures.

3.2. Thermal comfort

Beyond air temperature, the thermal comfort also depends on wall surface temperatures, calculated from the incidental solar radiations, relative humidity and air velocity.

Humphreys and Nicol (2000) established a relationship between comfort temperature and external average temperature (T_o), the equation for comfort temperature Tc, is given by:

$$T_c = 13.5 + 0.54\,T_0 \qquad (1)$$

They showed that this relationship is markedly stable between Humphreys' (1978) data and deDear's (1998) ASHRAE database.

In the current study, the focus is on dwellings located in Tlemcen, Algeria.

3.3. Geographical characteristics

Tlemcen is located in north-west Algeria, at 580 km from Algiers, with latitude of 34.56°, a longitude of -1.19° and an altitude of 830 m, see Fig.3.1. It is characterized by hot and dry summers and cold and harsh winters. It is important to remark that its climate is somewhat softened by the influence of the Mediterranean Sea, 45 km to the North.

Fig.3.1 Tlemcen site situation (Encarta 2008)

3.4. Solar radiation

Any use of solar radiation must take into account the local and regional climatic conditions. It is widely known that all the observed atmospheric processes are the consequences of the received solar radiation. Thus, the measurement of the solar radiation is of special interest for environmental researchers (Santamouris and Asimakopoulos 2001).

As shown in Fig.3.2, the largest increase in total radiation occurs between February and March and the maximum for the horizontal radiation is obtained in July, with a slight stability for the global collector slope between March and October. The annual values of diffuse solar radiation represent 35 – 44% of the global radiation.

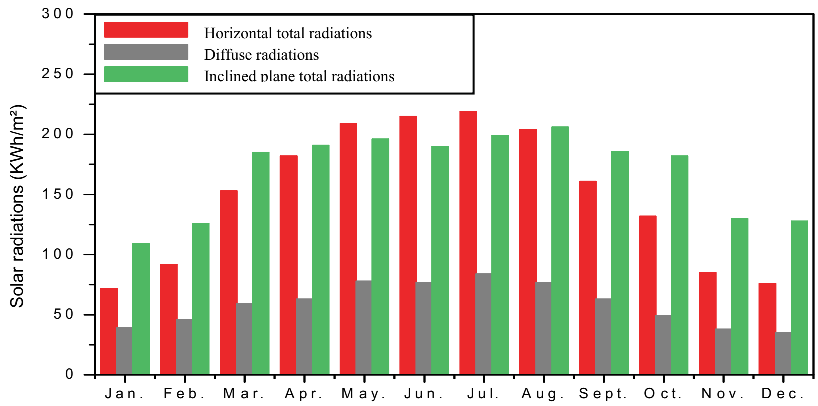


Fig.3.2 Variation of global, diffuse horizontal radiation and global inclined plane, (Meteonorm 6.1, 2008)

3.5. Relative humidity

The prevailing atmospheric humidity also plays a significant role in building comfort. High air temperatures coupled to high values of atmospheric humidity can create discomfort, mainly in the respiratory system.

Tlemcen site is characterized by a relatively moderate humidity varies from 55% in summer to 74% in winter (ranging around 66%), which, in general does not create discomfort. (Meteonorm 6.1, 2008)

3.6. Air temperature

It is difficult to define exact comfort conditions in buildings. Thermal comfort is therefore a function of the change as well as the temperature achieved; the 'comfort zone' width will depend on the balance between the two.

The adaptive relationship between comfort temperature and the outdoor temperature can be used to help design comfortable buildings (Fig.3.3). Here the indoor comfort temperature (T_c) is calculated from the outdoor temperature mean (T_m) and plotted on a monthly basis together with the monthly mean of the daily maximum (T_{emax}), minimum (T_{emin}) and mean outdoor temperature (T_m).

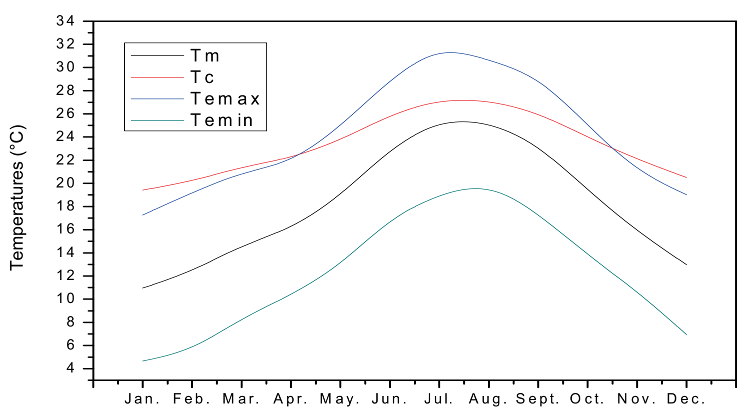


Fig.3.3 Typical variation of monthly temperatures in Tlemcen, (Meteonorm 6.1, 2008)

Such diagrams help the designer to judge whether passive heating and/or cooling is possible for such a climate. The relationship between the comfort temperature and the range of outdoor temperatures shows whether, for instance, night cooling is likely to be a viable way to keep the building comfortable during the day in summer or to calculate whether passive solar heating will be sufficient in winter. This method has been used by Roaf et al, (2001) to define indoor comfort. However, Eq. (1) does not apply in cold climates since an air temperature of -25°C gives the comfort temperature T_c=0°C.

3.7. Energy requirements

3.7.1. Heating and cooling degree days

Independently of the ambient air temperature variations, the temperature variations inside buildings are fundamental for the occupants comfort. For energy need assessment and to maintain indoor comfort the temperature variations are essential. The Degree Day (DD) Method is commonly used to estimate the heating and cooling demand in buildings.

The Degree Day Method is used for various applications e.g. the adequate design of photo thermal generators, greenhouses, and heating and cooling of buildings.

The number of Degree Days (or Degree Hours) for heating (DDh) and cooling (DDc) are calculated as the accumulated difference between a defined indoor base temperature and the ambient diurnal or hourly mean temperatures. Fig.3.4 and 3.5 shows the average of 3 years data of the ambient (Ta) and ground (Ts) temperatures in Tlemcen obtained from National Office of Meteorology (Paper II).

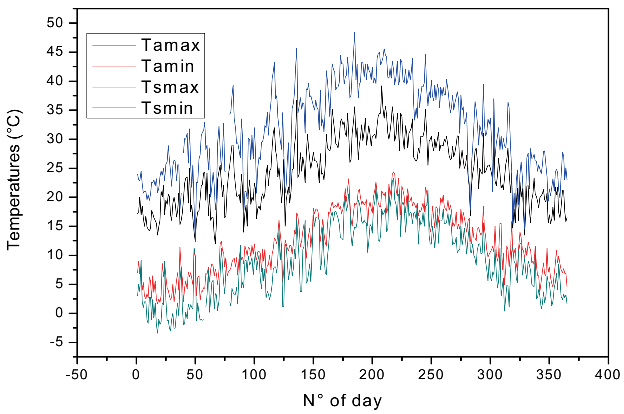


Fig.3.4 Daily ambient air temperature and ground temperature variations (Max and Min) in Tlemcen site.

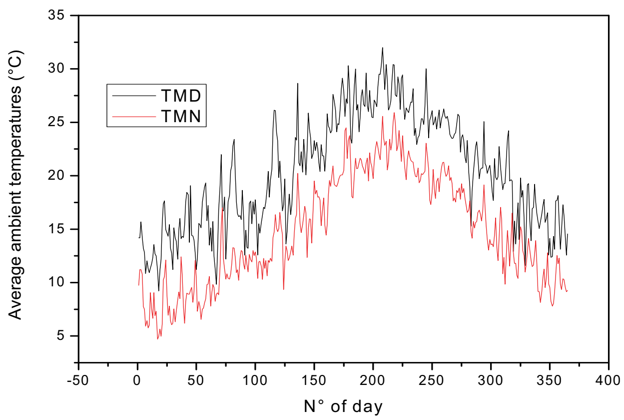


Fig.3.5 Day (TMD) and night (TMN) temperatures in Tlemcen.

In the light of the results obtained and shown in Fig.3.4, it is seen that the difference between the maximum and minimum ambient and in the ground temperatures in Tlemcen site never exceeds 10°C. Fig.3.5 shows the daily mean air temperature used in the DD calculation during the whole year for a considered building in Tlemcen. The result of the DD calculation is summarized in Table 3.1.

Table 3.1 Monthly Degree Days in Tlemcen.

Month	Tc	Average ambient temperatures	DD for heating (DDh)			DD for cooling (DDc)		
			15°C	Tc	18°C	22°C	Tc	25°C
January	19.4	10.4	141.4	192.5	234.4			
February	20.2	12.0	83.5	134.0	167.5			
March	21.4	14.0	50.0	146.0	128			
April	22.2	15.2	24.0	125.5	92.5			
May	23.7	18.0	5.2	91.0	34			
June	26.0	22.0				30.0	30.0	3.5
July	27.3	24.5				78.0	8.0	12.5
August	27.2	24.2				69.0	19.0	12.0
September	26.1	22.4				18.0	27.0	1.5
October	24.0	18.5	13.0	93.5				
November	22.1	15.6	19.0	117.5	76.3			
December	20.5	13.0	65.0	152.0	156.5			
Total			402.0	1052	889	195.2	84.0	30.0

According to this table, the Tlemcen site is characterized by a long duration of heating (October to May), with the following degree day number:

- For comfort temperature (15°C), DD_h = 402
- For comfort temperature (18°C), DD_h = 890
- For comfort temperature calculated every day, DD_h = 1052

And a short duration of cooling from June to September, of which:

- Comfort temperature (22°C), DD_c = 195
- Comfort temperature (25°C), DD_c = 30
- Comfort temperature calculated every day, DD_c= 84

Based on the calculated number of degree days, the building's energy demands for heating and/or cooling were calculated.

3.8. Building

On the architectural plan the building is designed in the following way (Paper IV). As shown in Fig.3.6, the living surface is 165m^2. In these conditions it is necessary to take into account, on the one hand the liveable volume V_h = 448 m^3, and on the other hand the volumetric coefficient of losses G considering the thickness of the walls, materials used and the number of openings (Table.3.2), (Paper III).

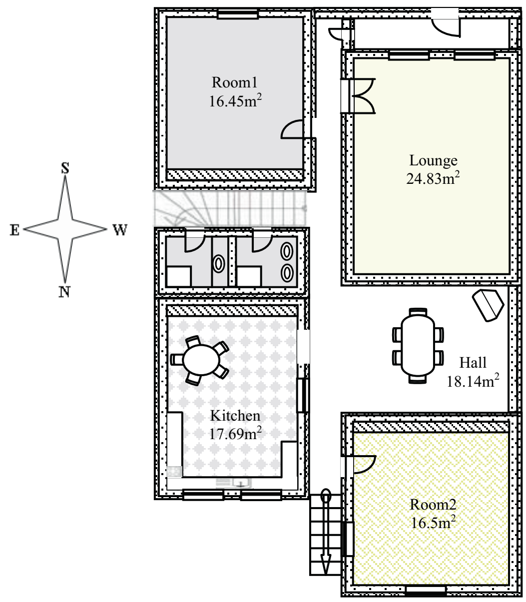


Fig.3.6 Drawing plan of building.

Table 3.2 Total energy losses in building by degree

		Walls	*Windows*	*Floor*	*roofing*	*Total losses (W/°C)*
Lounge	***K***	*3.5*	*6*	*4*	*4*	
	S (surface)	*62*	*7*	*35*	*35*	
	KS	*217*	*39.5*	*140*	*140*	*536.4*
Hall	***S***	*36.5*	*11.2*	*20*	*20*	
	KS	*127.5*	*65*	*80*	*80*	*352.4*
Room 1	***S***	*52*	*4*	*25*	*25*	
	KS	*182*	*23.2*	*100*	*100*	*405.2*
Room 2	***S***	*50*	*6*	*25*	*25*	
	KS	*175.7*	*34.2*	*100*	*100*	*410*
Kitchen	***S***	*53*	*8.7*	*30*	*30*	
	KS	*185*	*50.5*	*120*	*120*	*475.6*
Passageway	***S***	*52.5*	*32*	*15*	*15*	
	KS	*183.5*	*185.6*	*60*	*60*	*489*
Bathroom	***S***	*31*	*4.2*	*15*	*15*	
	KS	*108.5*	*24.4*	*60*	*60*	*253*

Then the total sum of losses (for $\Delta T=1°C$) for the building is P = 2922 W/°C which gives the volumetric heat loss coefficient G as:

$$G = \frac{P}{V_h} = 6.5\ (W/m^3\,°C) \qquad (2)$$

It is difficult to take into account the inhabitants' thermal requirements in the building. Would they agree to not use certain rooms in winter for heating or summer for cooling? Taking all this into consideration, we can evaluate the annual energy requirements C.

$$C = 24 * G * V_h * DD \qquad (3)$$

- Annual requirements for heating :
  - For a comfort temperature of 15°C, October to May: $DD_h = 402$
    Where C = 2815 kWh.
  - For a comfort temperature of 18°C, November to May: DDh = 889
    Where C = 6232 kWh.
  - For a calculated comfort temperature Tc, October to May: DDh = 1052
    Where C = 7375 kWh.

- Annual requirements for cooling :
  - For a comfort temperature of 22°C, June to September: DDc = 195
    Where C = 1369 kWh.
  - For a comfort temperature of 25°C, June to September: DDc = 30
    Where C = 209 kWh.
  - For a calculated comfort temperature Tc, June to September: DDc = 84
    Where C = 589 kWh.

3.9. Adaptation of the resource to requirements

The aim of this section is to have buildings with a surplus energy balance to maintain a stable interior temperature. The solar radiation is the biggest contributor to the superficial and volumetric energy balances during the daylight hours. In fact, it is the main contributor to heat gains in buildings, especially residential buildings.

It becomes necessary, therefore, in many cases to limit these gains for solar radiation, using elements that shadow the external surfaces of the buildings, resulting in improvement of the interior comfort, and the air conditioning costs reduction. The current study focuses on the energy demand calculation for a building situated in Tlemcen, which gave a:

- Maximum energy requirement for heating: 7375 kWh.

- Maximum energy requirement for cooling: 590 kWh.

Solar energy is mainly available when the heat demand is low and this is what constitutes the main disadvantage of solar heating. Fig.3.7 is a good illustration of how the solar energy resource is out of phase with the demand.

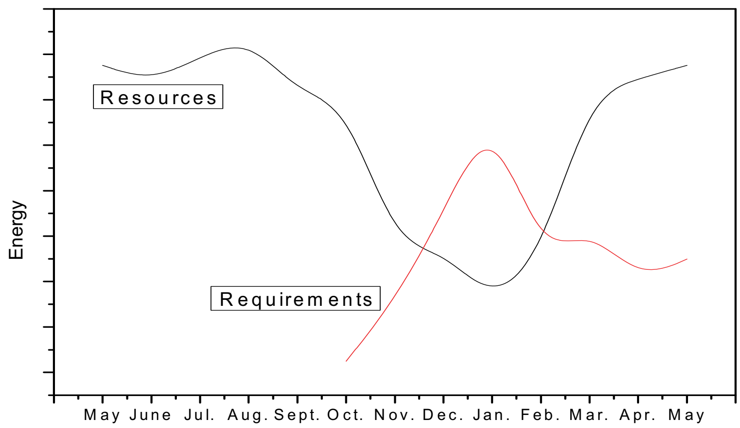


Fig.3.7 Available solar energy is out of phase with the heating demand, (Bernard et al. 1983)

In order to lead to low yearly consumption, an objective accessible in the shorter term is to set up solids bases on the envelope (walls, grounds and roofs). In particular, it will be interesting to introduce the positive energy which fits fully in the current context of energy control, use of renewable energies within sustainable development framework.

If a fraction of the available summer energy could be stored until its utilisation in the winter, the requirements for non-solar energy for heating could be reduced or even eliminated. For this reason it was decided to launch a study of the thermal behaviour of a domestic hot water storage tank.

4. Experimental study of the thermal behaviour of a domestic hot water storage

There are many reports on the necessity of thermal stratification to get the best performance of hot water storage applications. Moreover, Van Koppen et al. (1979) showed that a stratified storage was able to assure a high energetic gain particularly in combination with a reduced flow rate.

Other studies (Lavan and Thompson (1977), Phillips and Pate (1977), Wu and Hun (1978), Gupta and Jaluria (1981)) showed that stratification in a thermal storage tank depends mainly on the volume, height, inlet diameter, inlet and outlet design, flow rate, and the difference between inlet and outlet temperatures. Therefore there are four "primary factors of de-stratification" which contribute to loss and/or degradation of stored energy:

- Heat loss to the surroundings;
- Heat conduction within the tank, from hot layer to cold layer;
- Vertical conduction in tank walls and heat loss induced convection (mixing);
- Mixing of water as a result of charge and discharge, which is generally the principal cause of de-stratification.

The performance of hot water storage systems is improved by a highly stratified water temperature, which is made evident by temperature and velocity measurements in such storage. A good knowledge of the thermal behaviour of the storage is essential both for proper design of the storage tank and for prediction of their behaviour when they are installed in a system. Such predictions are needed for design and operation optimization (storage included).

Several numerical and experimental studies have been conducted on hot water storage, where the phenomena which affect the thermal degree of stratification and the thermal behaviour of low flow rate solar water heating are highlighted (Hermansson, 1983). The experimental results show the thermo-hydraulic behaviour in various phases of the storage cycle (injection of the fluid on several levels, drawing-off, auxiliary heaters) (Kenjo, 2003).

Bouhdjar (2005) studied numerically the stratification phenomenon in a thermal storage tank, in order to determine the thermal performances and the best configurations in relation to the thermal storage efficiency. Regarding the inlet temperature influence, a high storage performance is obtained with high inlet temperatures and minimal temperatures in the tank as low as possible (Johannes et al. 2004).

A study undertaken by Ouzzane et al. (1990), permitted to understand the thermo circulation phenomenon through the temperature distribution in the collector.

Sharp and Loehrke (1997) showed that the flows through a stratified solar water heater deliver appreciably more energy than the entirely mixed tanks. A simulation of similar systems shows that an optimum collector flow improves the performance by 37% compared to that of an entirely mixed tank (Wüstling et al. 1985).

Duffie and Beckman (1991) undertook works which led to a much more rigorous model founded on the "entrainment viscosity" theory. Later, Wu and Han (1978) presented another model compatible with TRNSYS. In the same context Chan et al. (1983) studied the mixed convection in a parallelepiped-shaped storage tank and examined the influence of some physical and geometrical parameters (Reynolds number, inlet and outlet positions) on the storage tank performance.

Thermal stratification in tanks requires prevention of convection heat transfer within the water. The mixing depends on tank geometry and functioning conditions (for example flow rate, inlet-outlet water temperature and temperature distribution inside the tank), as well as vertical conduction which can also induce convection streams (Davidson et al. 1994).

Chandreskhar (1961) and Turner (1973) indicated on this subject that the Richardson number has an impact in the formation and thermocline growth. Veltkamp (1981) asserted that the mixture occurs for Richardson numbers lower than 0.25. Experiments led by Loehrke et al. (1979), for a good stratification in tank, showed that the Richardson number was equal to or higher than 4.7.

We limited ourselves in this work to thermal behaviour study of a domestic hot water storage tank (Paper I and Paper V).

4.1. System description

The system considered included all the elements related to Fig.4.1 and Fig.4.2. We measured water temperature for different tank positions in Fig. 4.3. These temperatures are measured using 15 type K thermocouples connected to a FLUKE HYDRA SERIES II recorder. Water flow rate circulating in the system was measured by a flow meter.

Fig.4.1 Experimental set up at Renewable Energy Research Unit in Adrar region

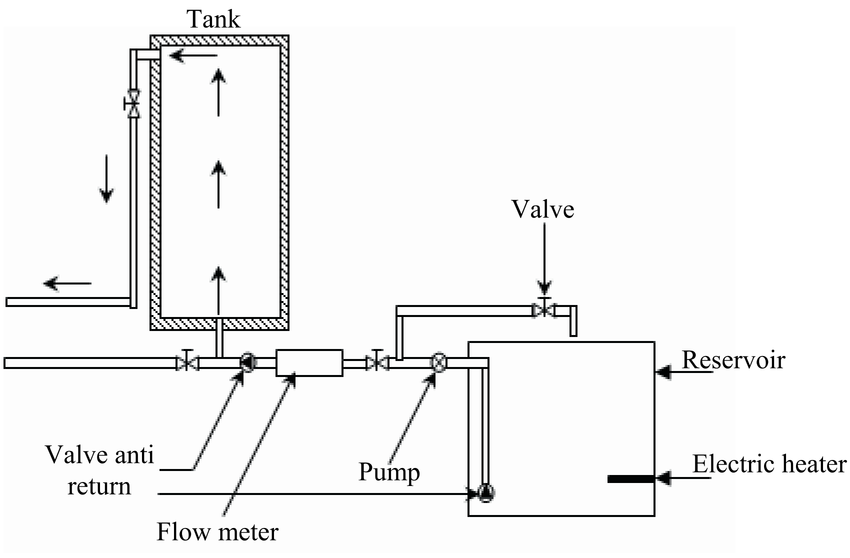


Fig.4.2 System synoptic scheme.

The system consists of:

- A 150 litre storage tank, with a the heat loss coefficient UA=2.75W/K. Primary fluid is injected from the bottom of the tank;
- A pump of a maximum 550 W;
- A reservoir of (60*60*45 cm^3) 160 litre;
- A flow meter;
- Two electric heaters (max 5 kW).

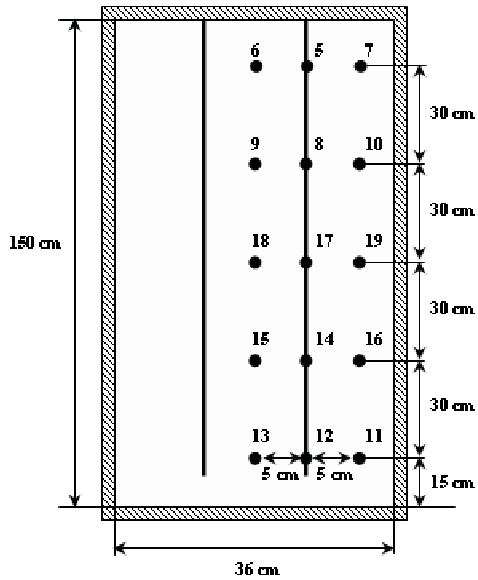


Fig.4.3 Thermocouples distribution in tank

In order to understand the thermal behaviour of the tank experiments were carried out on a traditional tank corresponding to both of the following scenarios:

- Charge of tank with electric heaters;
- Charge and discharge of tank with drawing off water.

4.2. Field experiments

The storage tank is a key element for the performance of the solar-water heater and its thermal behaviour was therefore studied by numerical modelling and experiments.

The first scenario was to inject fluid from the bottom of tank. At the end of manipulation, the temperature is nearly uniform whatever the inlet temperature. However, we noted the absence of stratification in the enclosure (Paper I).

In order to determine the fluid inlet temperature effect in tank, we carried out three tests which characteristics are given below in tables 4.1, 4.2 and 4.3.

Table 4.1: Test N°1 characteristics

Date	31/12/2007
Test duration	3 hours
Time step	30 s
Temperature constraints	55 ± 3°C
Flow constraints	1.6 ± 0.1 l/min

Table 4.2: Test N°2 characteristics

Date	02/01/2008
Test duration	2 hours 30 min
Time step	30 s
Temperature constraints	60 ± 3°C
Flow constraints	1.6 ± 0.1 l/min

Table 4.3: Test N°3 characteristics

Date	06/01/2008
Test duration	2 hours 30 min
Time step	30 s
Temperature constraints	40 ± 3°C
Flow constraints	1.6 ± 0.1 l/min

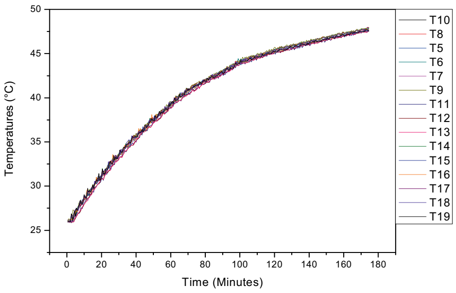


Fig.4.4: Temperature evolution in tank for 55°C inlet temperature

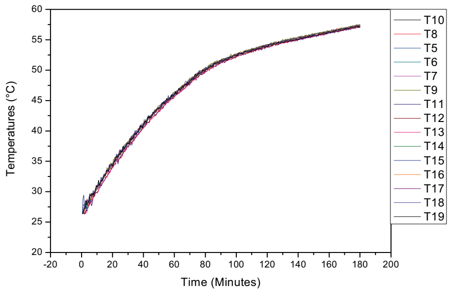


Fig.4.5: Temperature evolution in tank (inlet temperature: 60°C)

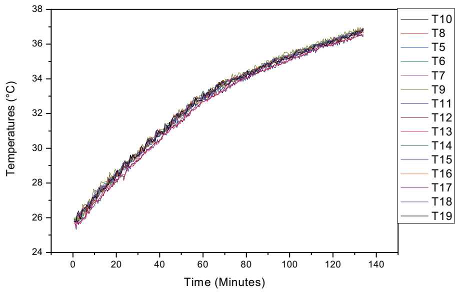


Fig.4.6: Temperature evolution in tank (inlet temperature: 40°C)

Figs.4.4, 4.5 and 4.6 present the measured mean water temperature variation in the tank for various inlet water temperatures (55°C, 60°C and 40°C). Here, we can immediately notice that the curves shapes are very similar. However, to well appreciate the system dynamics, we traced and modelled the temperature evolution for these tests. Since we can assimilate the tank to only one node, the heat balance could be written:

$$M\,C_p\,\frac{dT}{dt} = \dot{m}\,C_p\,(T_e - T) \tag{4}$$

The solution of this equation led us to the theoretical model:

$$T = T_0 + (T_e - T_0)\left\{1 - Exp((-\dot{m}/M)*t)\right\} \tag{5}$$

However, the experimental results gave a second model (two exponentials):

$$T = T_e + A_1 * Exp(-\frac{t}{t_1}) + A_2 * Exp(-\frac{t}{t_2}) \tag{6}$$

where:

T_e :	Inlet fluid temperature
r^2 :	Correlation factor.
A_1, A_2, t_1, t_2	Constants given in Table 4.4:

Table 4.4: Constants variation according to the inlet temperatures

	T_e	A_1	A_2	t_1	t_2	R^2
40°C	39.5	-6.96	-6.96	80.91	80.91	0.99
50°C	50	-12.34	-12.34	71.11	71.10	0.99
60°C	59.1	-16.76	-16.76	62.52	62.52	0.99

The results represented in Fig.4.7, permitted to validate the two exponentials model behaviour towards the theoretical model. The system dynamics is correctly reproduced, contrary to the theoretical model, where the maximum gap observed is about 3.02°C for a temperature of 50°C, and a very significant gap after 70 minutes for a temperature of 60°C.

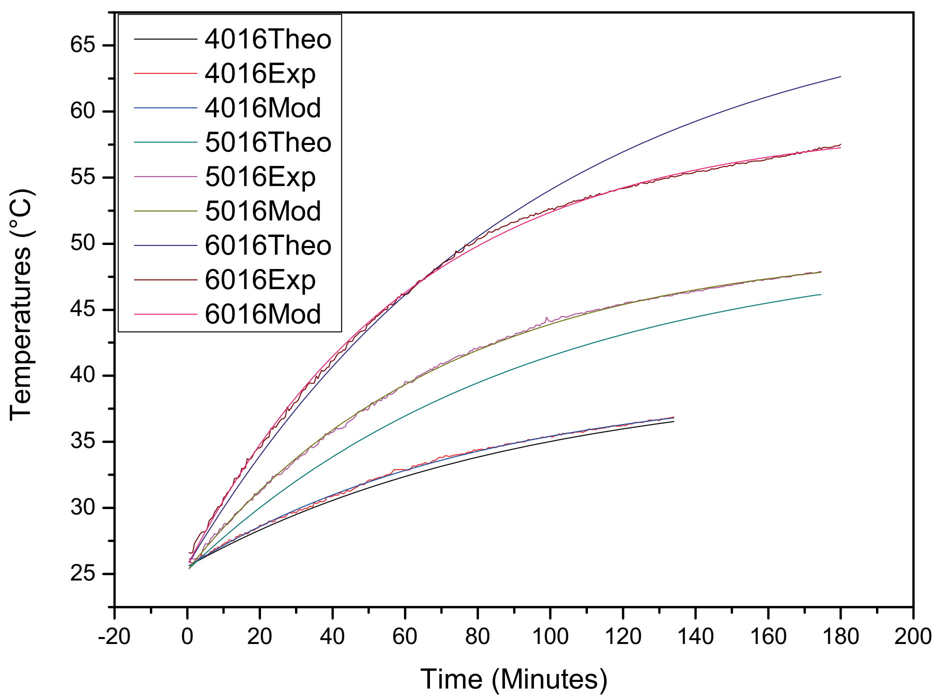


Fig.4.7: Temperature temporary evolution at top centre of tank (T6) for different inlet temperatures

4.3. Drawing off influence in tank thermal behaviour

The temperature profiles in Fig.4.8 and Fig.4.9 result from two hot water extractions of 20 min each, with a flow rate of 2.8 l/min. This means that the quantity of drawn water is 112 litres.

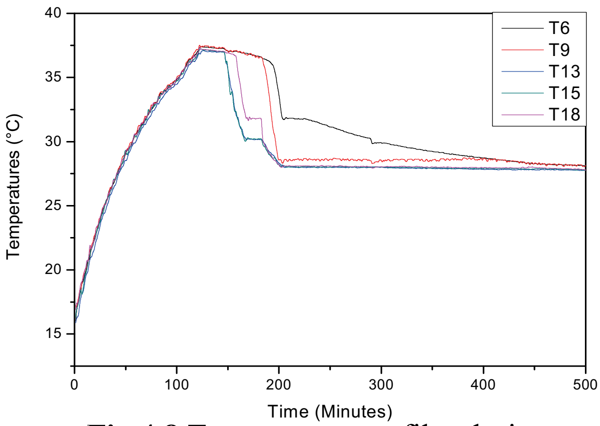


Fig.4.8 Temperature profiles during charge and discharge

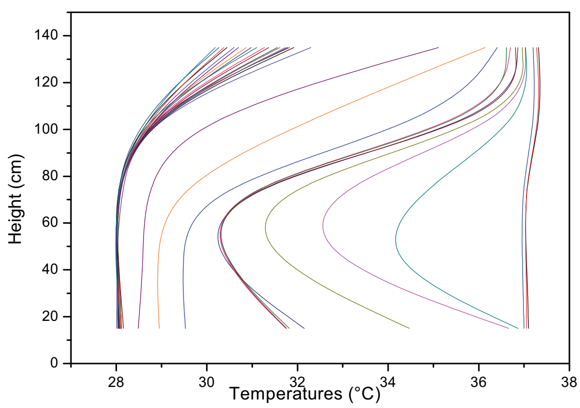


Fig.4.9 Temporary temperature evolution in tank during the two hot water extractions

In this second case, (Fig.4.8 and Fig.4.9) the temperature at the top of the tank decreases by 7°C after the first drawing off and 9°C after the second, whereas the temperatures in the lower layers decrease from 37°C to approximately 28°C, with the appearance of a small degree of stratification due to the two drawings off. After this dynamic phase and after a while it will be a balance within the tank due to the mixture by convection of water and the temperature is about 29°C.

The confrontation of numerical results with experimental data showed that the two exponentials model correctly reproduces the storage tank thermo-hydraulic behaviour for various stresses. Furthermore, we can see an absence of the stratification phenomenon within the tank.

4.4. Exchangers

The storage tank and its heat exchanger are the essential elements that contribute to optimize the performance of low flow solar water heaters. The exchangers used were made of copper with a diameter of 15 to 20 mm and 1m in length. This type of system significantly improves the thermal stratification in the storage tank. For that, we undertook other tests on a new configuration of tank, equipped with an exchanger (Fig.4.10), and thermocouples measuring nine fluid temperatures at different heights of the tank (Fig.4.11).

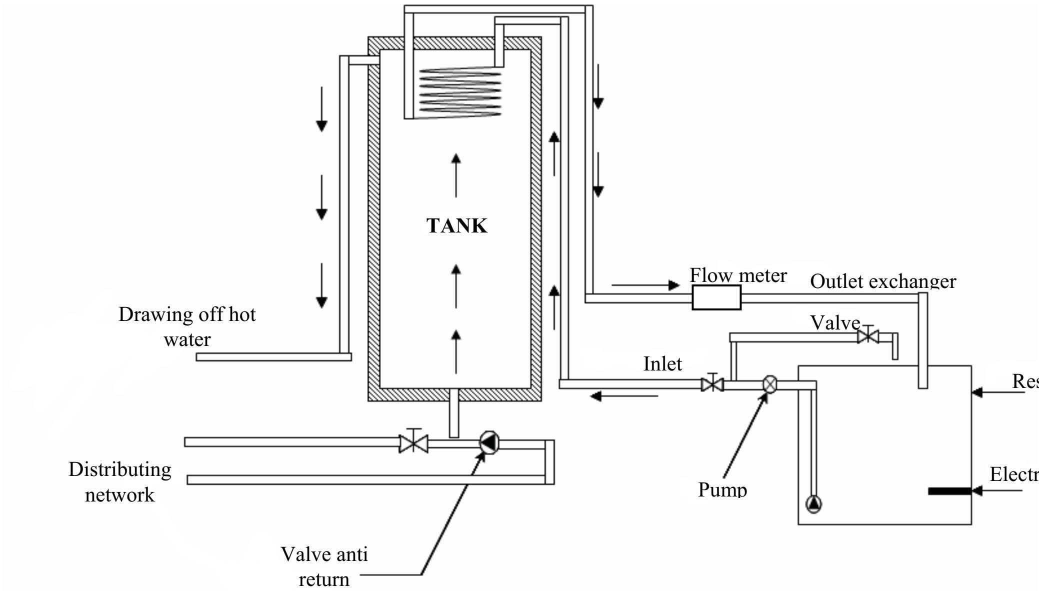


Fig.4.10 System synoptic scheme with the heat exchanger at top of the tank

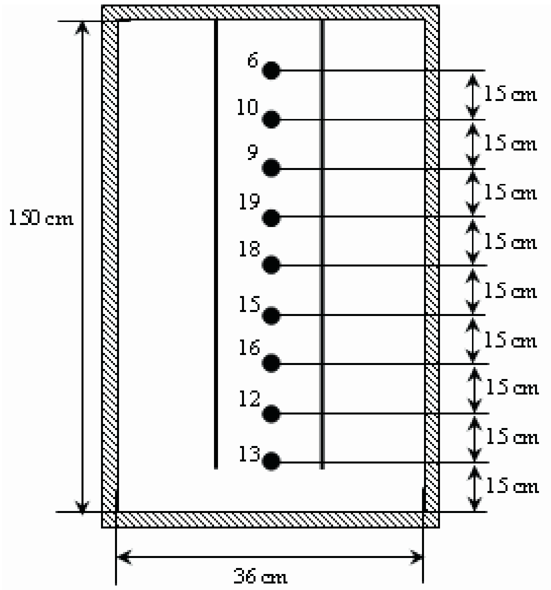


Fig.4.11 Location of thermocouples for temperature measurements in the tank

In this test a comparison was made between different positions of the exchanger. Tables 4.5, 4.6 and 4.7 show the characteristics of each test.

Table 4.5: Test N°4 characteristics (exchanger in the top of tank)

Date	11/05/2008
Test duration	80 min
Time step	1 min
Temperature constraints	50 ± 3°C
First loop, flow constraints	6.2 ± 0.1 l/min

Table 4.6: Test N°5 characteristics (exchanger in middle of tank)

Date	11/05/2008
Test duration	110 min
Time step	1 min
Temperature constraints	50 ± 3°C
First loop, flow constraints	5.8 ± 0.1 l/min

Table 4.7: Test N°6 characteristics (exchanger in the bottom of tank)

Date	11/04/2008
Test duration	3 hours
Time step	1 min
Temperature constraints	50 ± 3°C
First loop, flow constraints	5.6 ± 0.1 l/min

Figs 4.12, 4.13 and 4.14 show temporary temperatures inside the tank and exchanger inlet/outlet temperature for various positions of the heat exchanger.

By comparing the three configurations, we note that (Fig.4.12) a volume of 33 litres of the fluid reached the temperature of 45°C in 80 min. For the second configuration (Fig.4.13), 76 litres reaches 45°C in 95 min, with a high degree of stratification. Regarding Fig.4.14, the whole volume of the tank warmed up in almost three hours. As the stratification is a significant phenomenon for storage, it is preferable to put the exchanger either at the middle or in the top of tank.

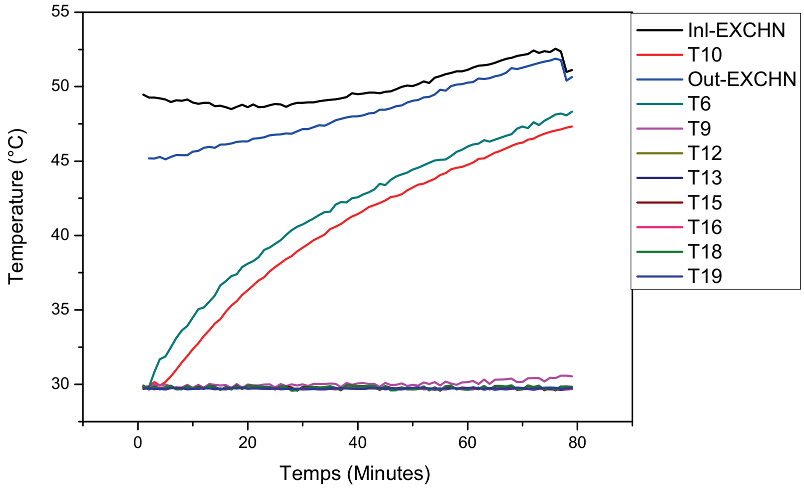


Fig.4.12: Temperatures profile in tank with exchanger at the top

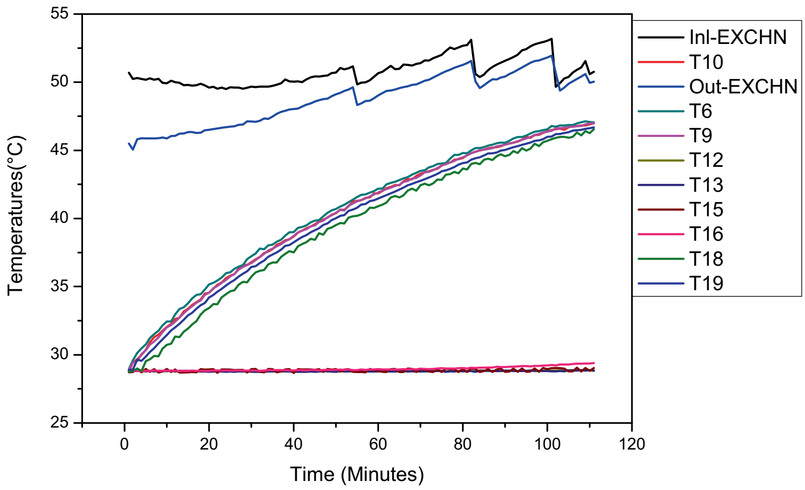


Fig.4.13: Temperatures profile in tank with exchanger at the centre

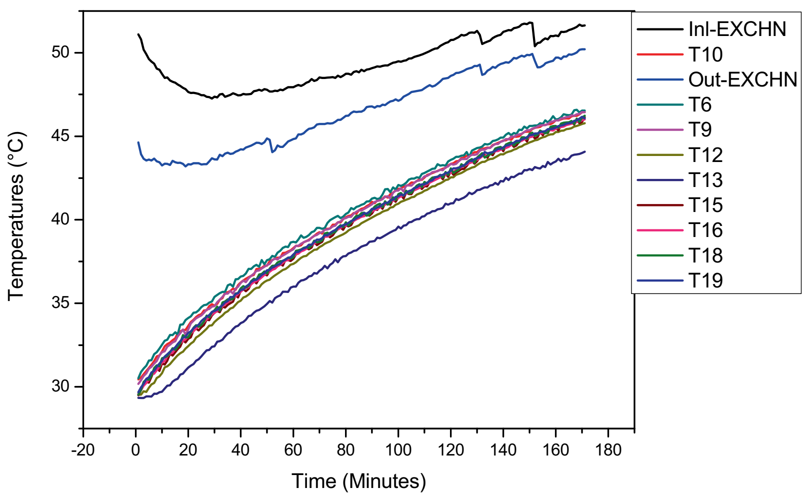


Fig.4.14: Temperatures profile in tank with exchanger at the bottom

4.5. Conclusion

The tanks most commonly used for hot water storage have a cylindrical form. The first series of tests was carried out in order to highlight the system thermal behaviour, when it is subjected to various controlled loads.

The resolution method of the two exponentials model was validated, compared to the theoretical model.

The thermal performances were determined and the most effective configurations compared to the thermal storage efficiency were identified. It shows, not unexpectedly, that the highest efficiency is obtained when the hot fluid is injected into the top and the cold fluid is drawn from the bottom of tank.

Regarding the influence of the exchanger position, the stratification in the middle of the tank is very well reproduced with better performances for storage.

It is necessary to take into account all these phenomena in solar applications especially the coupling between the solar collector and storage tank. To reach this objective it is interesting to combine the storage tank with a concentrating solar collector using an optical fiber.

5. System for hot water production and storage

Solar energy has for many centuries been harvested by various methods. Different types of solar collectors, normally producing hot water at temperatures between 30°C to 90°C, are commonly used. Large areas of mirrors are used to concentrate the solar heat to very high temperatures in order to drive boilers for production of electricity. The other main development is solar cells (photo-voltaic) for direct production of electricity. More recent development concerns the collection and transmission of visible light through optical fibers for direct illumination (André and Schade, 2002).

Optical fibers play an important part in the fields of both long distance telecommunications and short link networks (LAN's, etc.). More recently, designers have successfully applied less expensive, more mechanically robust fibers to various problems of illumination, such as safety lighting, background lighting, and medical lighting, among others. Typically, optical fiber for communications purposes used glass fiber with very small core diameters. On the other hand, more recent development work has concentrated on improving the light-propagating properties of larger core diameter fiber made out of plastic (William and Charles, 2002).

Fig.5.1: A SOLUX collector without the protecting acrylic dome [André and Schade (2002)].

In the current study hot water production is achieved by a solar (light) collector (Fig.5.1) from which the visible light is transported though an optical fiber (Fig.5.2) to a water tank where the light dissipates into heat.

Since light collectors have the advantage of low attenuation, and no losses in the conversion from light to heated water (Cariou et al, 1985) it is an energy efficient way to produce hot water. The efficiency of conventional flat plate solar collectors means that approximately 60% of incident solar energy is collected and transformed into 60°C water.

The objective of this study was to design a new light collector system for hot water production with considerably higher efficiency than that of conventional systems.

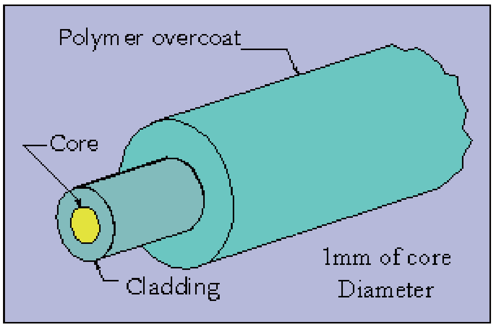


Fig.5.2: Construction of optical fiber [ATIS Telecom Glossary, 2007].

The idea of carrying concentrated solar energy via optical fibers was first suggested in 1980 by a group of French researchers (Cariou et al., 1982). Kandilli and Ulgen (2007), who reviewed performed work in this field, showed that experimental studies began more recently as summarized below:

- Khatri et al. (1993) discussed a solar energy collection system in which optical fibers are used to transport energy from a single-stage and a double-stage compound parabolic concentrator (CPC).

- Liang et al. (1997) emphasized the high flux solar energy transmission by a flexible optical fiber bundle and the research on the associated CPC could greatly expand the existing field of solar energy concentrator applications.

- Gordon (2001) showed that optical fibers used to transport sunlight exhibit considerable light leakage within their nominal numerical aperture and this leakage depends on (i) the incidence angle, (ii) the optical properties of the core and the cladding and (iii) the fiber length.

- Kato and Nakamura (1976) show that the technology for optical fiber transmission offers a high quality production with a large diameter of the core.

- Feuermann and Gordon (1998) in studying solar surgery used remote fiber-optic irradiation with highly concentrated sunlight in lieu of lasers.

- Jaramillo et al. (1999) developed a theoretical thermal study of optical fibers transmitting concentrated solar energy.

- Kribus et al. (2000) presented a study on the potential use of optical fibers for solar thermal power generation. The main performance characteristics (numerical aperture and attenuation) and typical costs of currently available fibers were discussed.

5.1. Prototype concentrator design

The effective incoming radiation to the aperture plane is the beam radiation. An incident beam of solar radiation is a cone with an angular width of 0.53° (Duffie and Beckman 1991). The focal length is a determining factor in image size, and the aperture is the determining factor in total energy. It is assumed that the beam radiation is normal to the aperture and the reflection is specular and perfect.

Each optical fiber has a pure transparent inner core and a thin transparent outer cladding. The total internal reflection allows us to guide the sunlight through the fiber. The fiber core has an index of refraction n_1, which is greater than that of the cladding n_2. The ratio of the core index and cladding index determines the acceptance/admission angle of radiation θ_{max} at which total internal reflection occurs:

$$NA = \sin\theta_{\max} = (n_1^2 - n_2^2)^2 \qquad (7)$$

Here, NA is the numerical aperture. On the other hand, the energy rate Q_p hitting a flat receiver of a paraboloidal concentrator, where the optical fiber inlet is placed, is given by

$$Q_p = \pi f^2 (\sin^2\phi_r - \sin^2\phi_s)\rho_m G_b \qquad (8)$$

Where f is the focal length, ρ_m is the reflectance of the surface, ϕ_r is the rim angle of the paraboloidal dish, ϕ_s is the shading angle because of the receptor size and Gb is the solar beam irradiance (Jaramillo and Del Rio 2002). It is important to mention that the rim angle ϕ_r should be equal to or smaller than the optical fiber admission angle.

In optimum conditions, to ensure that the whole radiation gets into the fiber, the maximum rim angle of the paraboloidal dish must be

$$\phi_r = \theta_{\max} \tag{9}$$

This corresponds to the maximum admission angle of the optical fiber.

On the other hand, from the definition of decibel losses per unit length, the energy rate Q_0 at the end of the optical fiber can be expressed as:

$$Q_0 = Q_p 10^{-L\, dB_{loss}/10} \tag{10}$$

5.2. Outline of Studied system,

The studied light collector system for hot water production, which is outlined in Fig.5.3, contains:

- Light concentrator represented by a parabolic mirror that has the following geometrical characteristics: Diameter (D) = 1.5 m.
- Optical fiber SPCH 1000/1035/1400Z from Fiberguide company:
    - Length = 3 m,
    - Silica core with a diameter of 1 mm,
    - Average attenuation $\tau = 6.2*10^{-3}$ dB/m on all solar spectrum,
    - Numerical aperture θf=20°.
- Cylindrical storage tank:
    - Height = 1.5m ,
    - Volume = 0,15 m^3,

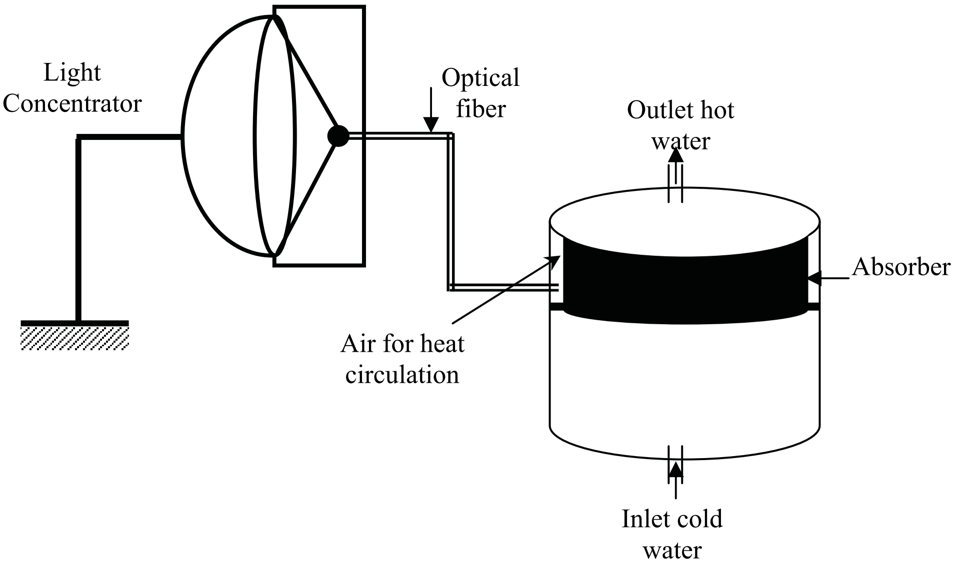


Fig.5.3: Light collector for hot water production and storage

Table.5.1 Parameters of parabolic dish

Parameters		Parameters	
D_a (aperture diameter (m))	1.5	C_{max} (maximum ratio of geometrical concentration)	1858
D_r (receptor diameter (m))	0.035	NA (Numerical aperture)	0.4
f (focal length (m))	0.88	g_{max} (combined expression of the view factor and concentration ratio)	8309
f_0(optimal focal length (m))	0.9	d_{min} (minimal focal spot diameter (m))	0.020
Φ_r (paraboloidal dish rim angle(°))	46	d_{max} (maximum focal spot diameter (m))	0.034
Φ_s (paraboloidal dish shading angle(°))	0	ρ_m (reflectivity of the paraboloidal dish)	0.85
θ_{max} (admission/acceptance angle(°))	45	η_{of} (efficiency of the fiber-optic bundle (%))	99.7
$\delta/2$ (dispersion angle)	0.4	η_o (optimum optical efficiency (%))	61.7
dB_{loss} (attenuation of optical fiber (dB/km))	$6.2*10^{-3}$	η_s (overall system efficiency (%))	61
L (fiber-optic bundle length (m))	2		

Table 5.1 presents measured and calculated parameters of the parabolic dish. The optical fiber used is the type SPCH 1000/1035/1400Z from Fiberguide Company. It has a length of 3m and its diameter is 0.001m. The maximum admission angle of optical fiber was determined by Eq. (7). The attenuation of the optical fiber is indicated as $6.2*10^{-3}$dB/m by the manufacturer data. The ideal transmission efficiency of the optical fiber was found to be 99.7%. For maximum concentration ratio, the rim angle was found to be 1858 and used.

In this case and using θ_{max} the view factor Fs was found to be 0.728. Consequently, optical efficiency of the whole system was calculated as 61.7% in optimum condition. Overall system efficiency was found to be 61%. These losses are due to the absorptivity and reflectivity of the dish, sun tracking, to the transport by optical fibers and by pipes contact with the atmosphere.

The size of the image has uniform flux as given by d_{min}. As indicated in Table 5.1, d_{min} was calculated as 2.0 cm. If the aim is to receive all flux concentrated on the focal point, it should be considered maximum image size. For the existing system, d_{max} was calculated as 3.4 cm by considering dispersion effects.

The cylindrical storage tank is composed of four parts as shown in Fig.5.4.

- **Part 1 :** External wall of the tank made of galvanized steel(H=1.50m; D=0.60m),
- **Part 2 :** 5cm of thermal insulating foam,
- **Part 3** *:* Internal wall of tank made of galvanized steel (H=1.45m; D=0.50m),
- **Part 4** *:* Absorber of galvanized steel with (H=0.50m; D=0.46m).

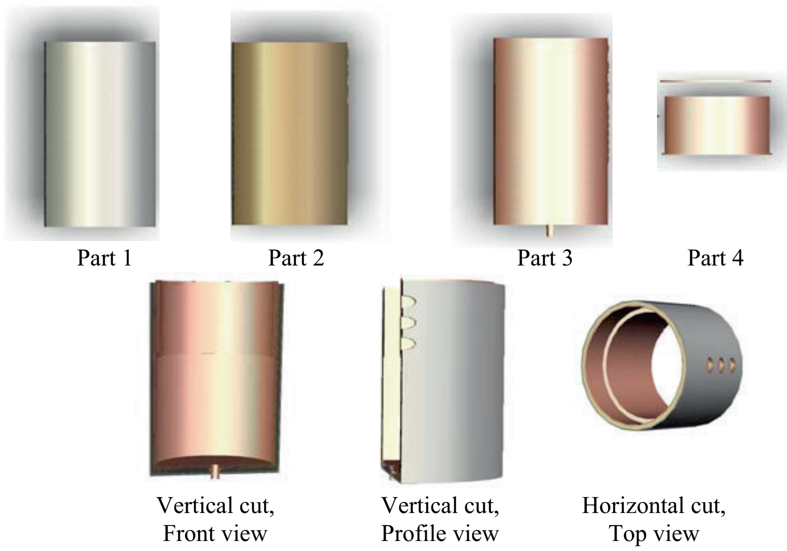


Fig.5.4 Description of how the cylindrical tank is assembled.

5.3. Experimental study of the prototype

The prototype design was based much more on the materials available on the market. Flow measurements were initially made by a flow meter, but after it was destroyed it was replaced by the much more accurate old system “bucket and stopwatch”.

After designing and assembling the two modules (concentrator + storage tank) (Fig.5.5), we undertook an experimental study.

Fig.5.5 Photos of the designed system and sunlight in the optical fiber outlet

Practical considerations militate against large and small dish rim angles. At small rim angles, relatively high efficiency can be realized at peak concentration; but for a given fiber size, smaller diameter dishes must be used. Dish focal length is large relative to the diameter, which creates a practical problem in terms of module depth (Paper VI).

To estimate realistic system performance, we reduce the theoretical limits by the following factors:

1. An absorptive loss in the dish mirror of 5%; (Feuermann and Gordon, 1999).

2. Fresnel reflective losses at the module glazing (3% with an AL coating); (Feuermann and Gordon, 1999).

3. Fresnel reflective losses at the fiber ends (a total of 5% with AL coatings); (Feuermann and Gordon, 1999).

4. Ray rejection in the secondary concentrator (in the range of 1 to 4% depending on the dish rim angle, and about 2% at ϕ=45°) (Welford and Winston, 1989);

5. Absorption in a remote second stage of about 2 to 4%, depending on ϕ (absorptive losses in a dielectric proximate second stage are negligible) (Welford and Winston, 1989);

5.4. Results and discussion

Adrar city is located in south-west of Algeria, at 1600 km from Algiers capital, with a latitude of 27.88°, a longitude of -0.88° and an altitude of 270 m. It is characterized by particular climate conditions, i.e. hot and dry summers, cold and harsh winters.

Any use of solar radiations must take into account the local and regional climatic conditions. It is widely known that all the observed atmospheric processes are the consequences of the received solar radiation. Thus, the measurement of the solar radiation presents special interest for the environmental researchers (Santamouris and Asimakopoulos 2001).

As shown in Fig.5.6, the great increase for whole radiations is located between February and March; and a slight stability for the global collector slope between March and October.

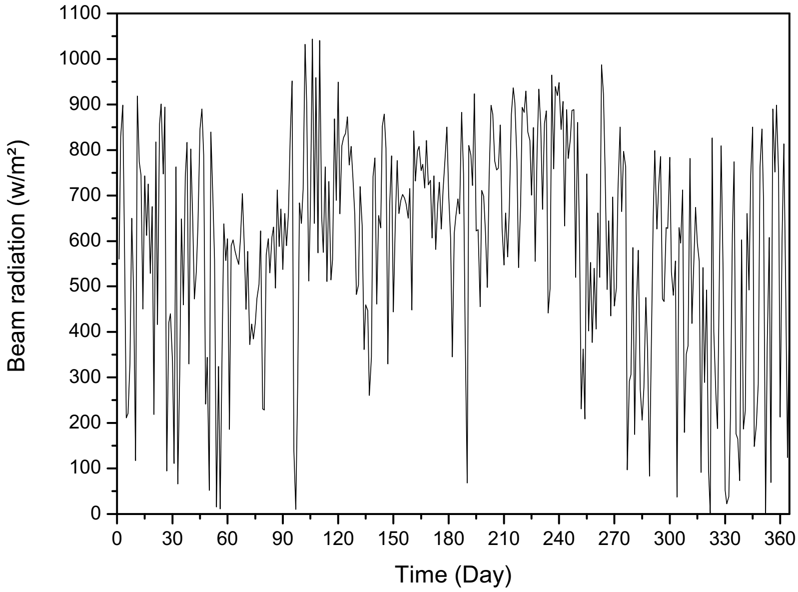


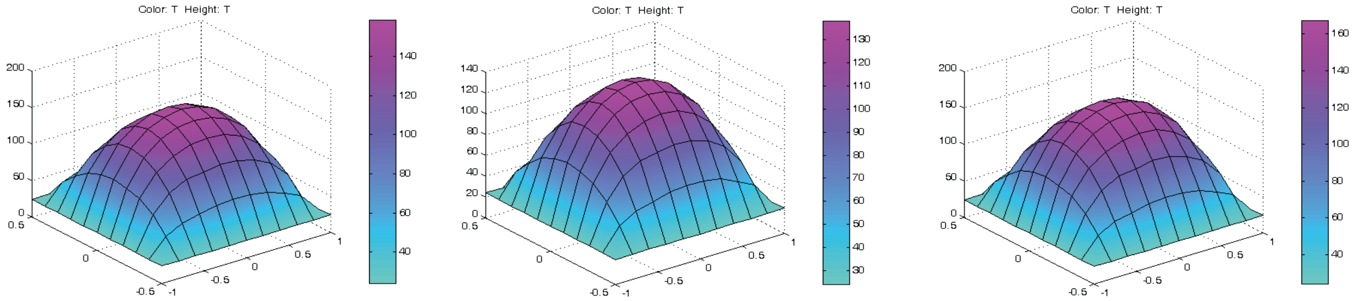
Fig.5.6 Calculated hourly solar beam radiation (W/m²)

Hourly global irradiance data of 2004 obtained from meteorological stations were used to predict the hourly normal beam radiation reaching the aperture plane of the paraboloidal dish. Global irradiance data were separated to beam and diffuse component by using the Clear Sky Model [Meteonorm 6.1, 2008].

The calculated hourly beam radiation was used to evaluate the monthly average hourly output power obtained from the paraboloidal dishes.

These data were presented to estimate the output power of the system. Taking into account all the factors that reduce the system performance, Eq.(8) and Eq.(10) give the annual mean of the output power from optical fiber .

After the determination of the output power quantity transported by the optical fiber to the storage tank, where it is absorbed then transmitted by the selective surface, the temperature in Fig.5.7 can reach 170°C at the equinoxes and 150°C at the solstice.



Summer solstice Winter solstice Equinoxes

Fig.5.7 Temperatures reached on a selective surface at the solstices and the equinoxes

In performed tests the water flow rate through the accumulator tank was 0.6 l/min. Fig.5.8 shows the inlet and outlet temperature on 22 January 2009. In the light of these results, we noted the existence of a gap in the beginning of the manipulation of over 10 °C which became 50 °C at noon. It took approximately four hours to fill the storage tank with a flow rate of 0.6 l/min to obtain an average daily temperature of 36.5 °C. The decreases observed in Fig.5.8 for the outlet temperature T_s resulted from the manual sun tracking system.

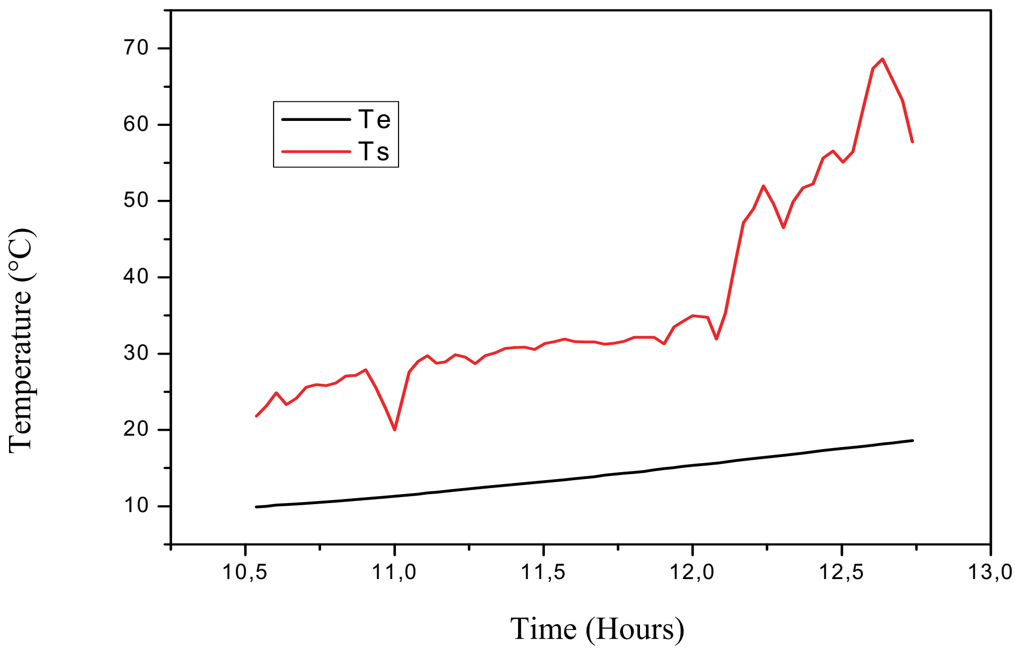


Fig.5.8 Measured water inlet and outlet temperatures, 22 Jan 2009.

5.5. Concluding remarks

The major finding of the current analysis is that the use of optical fibers in solar thermal concentrating systems for heating is technically feasible, under specific circumstances. In the first phase of our solar fiber-optic paraboloidal dish program the main conclusions of the performed study are:

1. A solar fiber-optic heating system was designed, constructed and tested
2. Collection, concentration and transmission of sunlight were demonstrated.
3. The mathematical model of paraboloidal dish system was described.
4. The measured output power from the dish was evaluated against radiation data from the local meteorological station supplied by the Renewable Energy Research Unit (URER/MS) in Adrar.
5. Optimizing the receiver and dish size is important in order to maximize the system efficiency.
6. The calculated optical efficiency of the system was 62% at optimum conditions.
7. The calculated mean power from the paraboloidal dish was 885 kW which for an aperture area of 1.7 m^2 means 520 kW/m^2.
8. The heat was stored in a 0.15 m^3 water storage tank.
9. The temperature of outlet water from the storage tank reached 70^oC.

During the experimentation we encountered some difficulties e.g.

- Dust accumulation on the mirror and at the fiber tip.
- The parabolic mirror was made from a satellite dish, the surface of which was clothed by a foil tape. Its aluminium plies reduced the reflectivity that meant extra energy loss.
- The optical fiber was aligned at the concentrator focus by placing a cone (Length = 0.15m Diameter = 0.075 m) in the parabolic focus, mouthing the mirror. It was problematic to attach the fiber exactly at the focus.
- The manual measurements meant fewer measurements and more errors.
- Instead of using the desired optical fiber with relatively large diameter and high numerical aperture (NA), two optical fibers with a NA=0.4 manufactured by FiberGuide Company, USA, was used.

The current study showed that concentrated solar energy can be transferred as light by optical fibers to a selective surface (absorber) inside a storage tank, where the light is converted to heat. Such systems should have great potential in a wide range of solar energy applications.

6. Using Fouggara for Heating and Cooling Buildings in Sahara

The objective of the current study was to investigate the feasibility of the ancient Fouggara system for heating, cooling and ventilation. The heating cooling demand of a building in Adrar, Algeria, was estimated and the function of the Fouggara was based on real operation data from a Fouggara in the Sahara desert.

Adrar is located in the south-west of Algeria (Fig.6.1), 1600 km from Algiers, with a latitude of 27.81°, a longitude of -0.18° and an altitude of 279 m. It is characterized by very hot and dry summers up to ~50^{o}C and cold winters down to ~0^{o}C (Fig.6.1 and Fig.6.2).

It is difficult to define the building comfort. Thermal comfort is therefore a function of how the building is used and the ambient temperature. The width of 'comfort zone' depends on the balance between both parameters.

Fig.6.1 Location of Adrar site (Microsoft ® Encarta 2008)

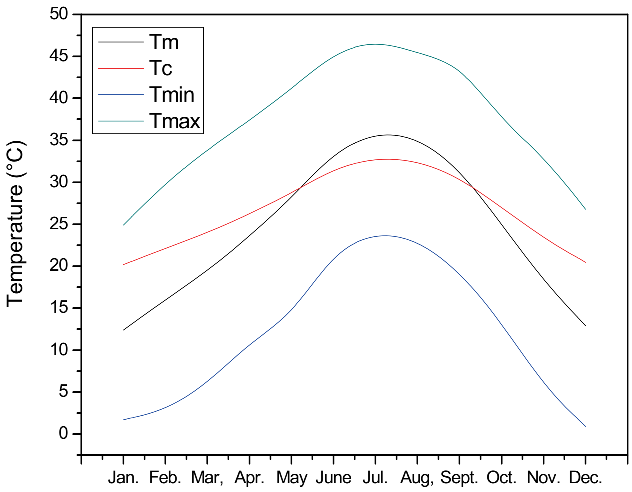


Fig.6.2 Monthly temperature variations in Adrar.

From these results (Fig.6.2) we can see that the Adrar site is characterized by an air temperature ranging from 0°C to 47°C. This climate means that cooling is required during 8 months while the heating period is 2-3 months.

The heating and cooling demand in northern Africa varies considerably in the different countries. The greatest cooling demand is to the East and in the Sahara desert, while the greatest heating demand is in the North West. In Algeria the heating demand is greater than the cooling demand.

Since the comfort temperature is based on monthly mean air temperature it is not affected by diurnal temperature variations. The indoor temperature variations are fundamental for the occupants comfort since the variations are essential for the energy demand and to maintain comfort.

Based in results presented in Fig.6.2, the monthly heating and cooling degree days (DD_h and DD_c) were calculated see Table 6.1.

Table 6.1: Monthly heating and cooling degree days.

Month	T_c	Ambient average temperatures	DD_h	DD_c
January	20.2	12.4	-241.8	
February	22.1	16.0	-172.0	
March	24.0	19.4	-142.0	
April	26.2	23.6	-80.0	
May	28.7	28.1	-17.8	
June	31.7	33.6		58.7
July	33.0	36.0		95.0
August	33.0	35.4		86.3
September	30.6	31.6		31.1
October	27.0	25.0	-62.0	
November	23.3	18.2	-153.8	
December	20.5	12.9	-234.5	

Table 6.1 shows that the Adrar site requires heating during October to May, with the total heating degree days DD_h = 1102. Cooling is needed from June to September based on comfort temperature as the base temperature. The total number of cooling degree days is DD_c = 271.

By using the building described in Chapter 3, the total energy requirements for heating and cooling of the test building are given below.

- Annual requirements for heating for a comfort temperature, October to May :
  - DDh = 1102; where C = 7.7 MWh.

- Annual requirements for cooling for a calculated comfort temperature T_c , June to September:
  - DDc = 271; where C = 1.9 MWh.

6.1. Heating and cooling using the ground

Depending on weather, time of day and season, outside air undergoes great variations of temperature and humidity. In contrast, the ground temperature at 10-15 meters below the ground surface is constantly equal to the annual mean air temperature. The air-ground exchanger also called Canadian well or Provence well takes advantage of this inertia by making contact between the outside air heat and the ground. Its primary purpose is the pre-conditioning of heat and humidity of ventilation air in buildings.

Specifically, the air-ground heat exchanger consists of tubes buried a few meters under the ground, near or directly under the building to ventilate. The air from the outside, moved by a ventilator, runs through the tubes before being blown into the building (Fig.6.3).

During the passage through the ground tubes, the air flow is warmed or cooled by exchanging heat with the ground. Thus, the air can also be dehumidified when cooled. The ground-air heat exchanger thus reduces thermal variations and air humidity, which corresponds to pre-conditioned air. This system uses very little driving energy, so, it is almost passive.

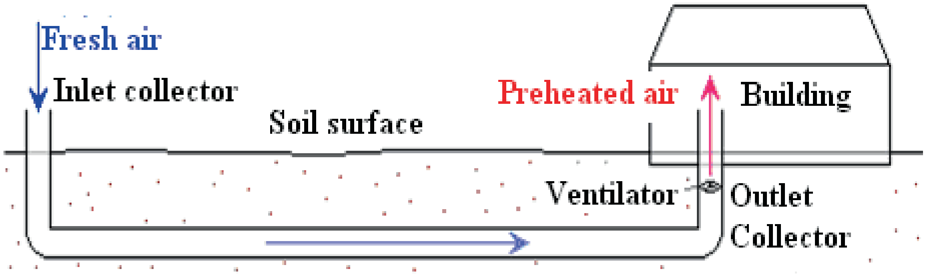


Fig.6.3 Simplified diagram of a Canadian well-pipe
(Example of operation preheating)

Air-ground exchangers are available on the market. The performance of this type of integrated equipment is benign for sustainable building development

The Canadian well-pipe system was chosen as the ventilation system of reference for this study. However, several studies address the technology of air-ground heat exchangers, either as a main subject [Mihalakakou et al. (1995a and 1995b); Benkert et al.(1997); Bojić et al. (1997); Benkert and Heidt (2000); Kunetz and Lefebvre (2001); Hollmuller (2002); Gehlin (2002); De Paepe and Janssens (2003); Dibowski (2003) and (2005), Zweifel (2004); Al Ajmi et al. (2005); Ghosal and Tiwari (2006); Badescu and Sicre (2003) and (2007), or as a specific study on cooling of buildings.

Such systems are used for cooling of buildings and industrial applications e.g. greenhouses (Gauthier et al. 1997; Hollmuller and Lachal 2001, Ghosal and Tiwari op. cit. 2006).

6.2. Fouggara: An ancient system for underground transport of water

The Fouggara is known under various names; "Kariz" in Afghanistan, "Qanat" in Iran, "Fouggara" in Algeria and "Khettara" in Morocco. The water flow through the Fouggara is gravity driven. At the mouth of the underground channels it reaches the surface where it is used for irrigation and as drinking water, see Fig.6.4a and Fig.6.4b.

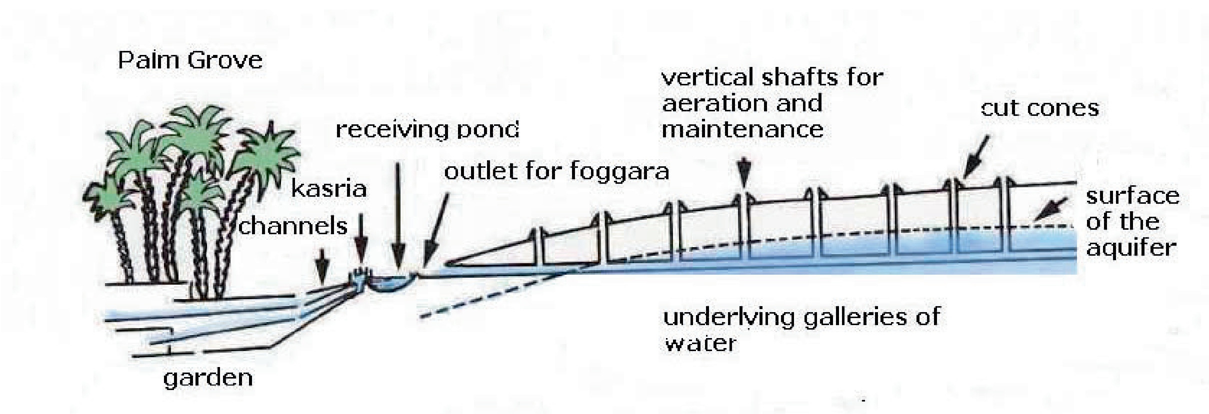


Fig.6.4a Outline of the Fouggara principle. (El FAiz and Ruf 2006)

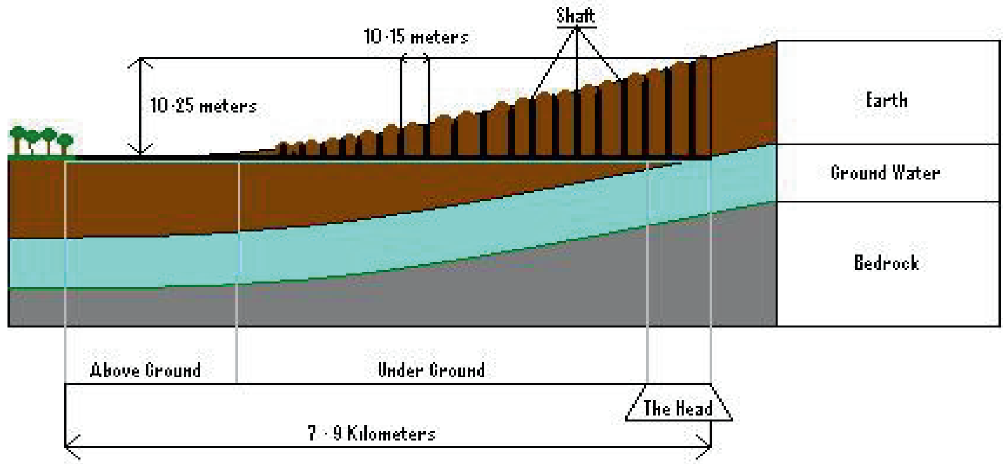


Fig.6.4b Vertical section along the Fouggara. (Khettara)

This Fouggara technique is very old and was first used in Iran. It was introduced in Algeria by Arabs at the time of their conquest of Algeria about the year 700. This technique was adopted by farmers of the area for the following reasons:

- The construction of the Fouggara (Khettara) does not require any financial expenditure, but simply an effort of work.
- Underground transportation of water minimises the water loss due to evaporation and prevents the accumulation of the sediment caused by sandstorms (El Faiz and Ruf 2006).

6.3. Characteristics

The Fouggara is made up of:

- The source where the water seeps into the channel from a ground water source,
- An underground channel which brings the water to its intended destination,
- An on-ground main channel which leads to a network of channels feeding the water to particular areas or fields for irrigation.

It is characterised by;

- The length of Fouggara varies from a few hundred meters to tens of kilometres.
- The Fouggara is gravity driven as the water source is situated at a higher elevation then the area to be irrigated.
- The underground channel is wide enough to allow the clearing out and cleaning by hand.
- The depth of the shaft at the source varies from 10 to 25 meters.
- The distance between two consecutive shafts depends in the stability of the terrain and varies between 10 and 15 meters.
- The water flow rate varies from 0.002 to 0.020 m^3/s
- It is located at a depth where the temperature is constantly at the annual mean air temperature (i.e. 21^{o}C in Adrar), (ABH Sahara).

In Algeria the Fouggara is extended in all the Sahara and in Adrar there are 900 Fouggaras, more than 2/3 of them are not in operation. Today, the land area irrigated by Fouggaras is estimated at 8500 ha even though only 1/3 of them are in operation. (Fig.6.5). (Bouzidi, 1999)

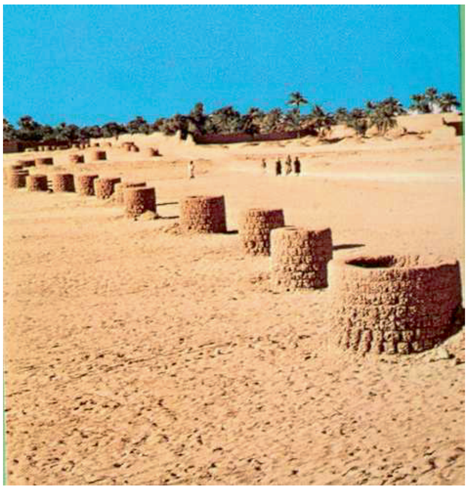


Fig.6.5 Many wells represent the Fouggara

The existing Fouggaras, active or inactive, mean available underground duct systems that could be used as a source for heating and cooling. In the remote Saharan villages of the Saharawi people (Fig.6.6), which lack resources for expensive drilling, available Fouggaras offer an inexpensive alternative. Such a system would be made simply by pumping air through the Fouggara and conducting this air to the air conditioned building. The ambient air then approaches the temperature of the ground (~21°C) during the passage through the ground before used for ventilation, heating, and cooling of buildings (Fig.6.7 and Paper VII).

Fig.6.6 Fouggara in Adrar.

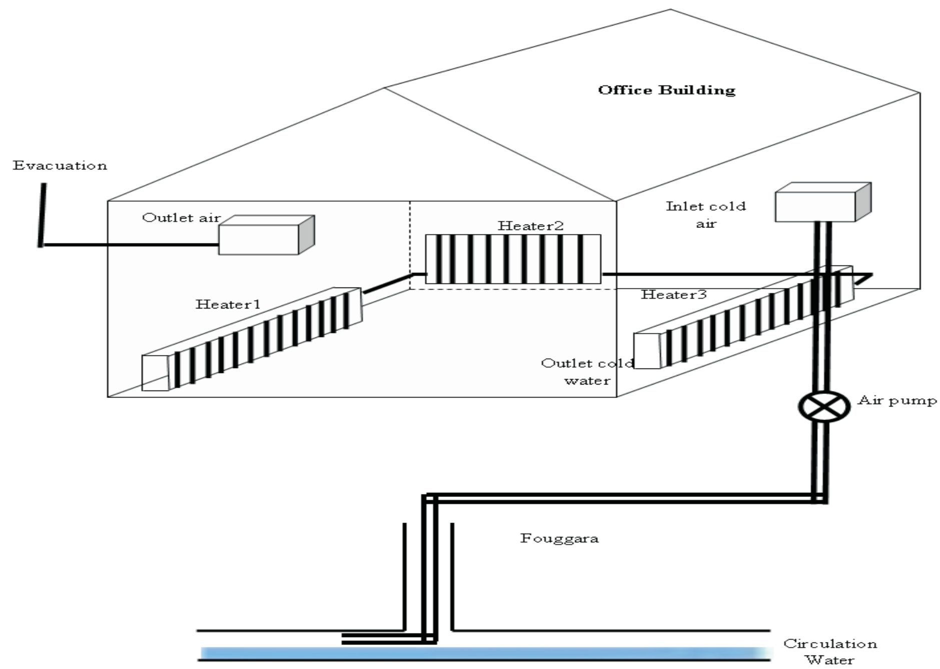


Fig.6.7 The ground-air conditioning system.

The suggested Fouggara air conditioning system, for cooling in the summer and heating in the winter of the URER test house consists of:

- One room characterised by a volume of 70 m3
- A pipe system connected to an air convector
- A pump with an air flow rate of 0.02 m3/h from the Fouggara.

Fig.6.8 shows the ambient temperature and the air conditioned indoor temperatures during the year, given that the difference between the ambient and indoor temperatures in Adrar is approximately ±20°C.

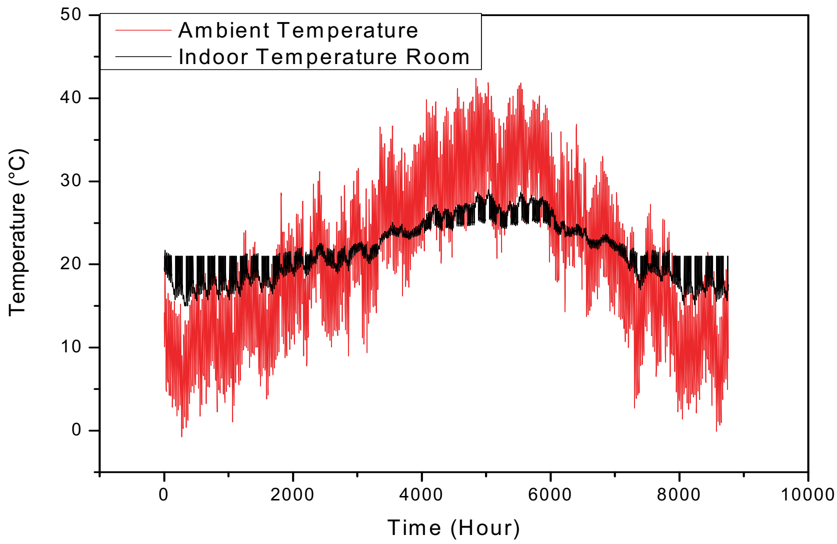


Fig.6.8 Ambient and Indoor temperature variation

6.4. Conclusion

In this chapter we presented an ancient system called Fouggara for heating and cooling of a building in the desert. It is characterised by a constant annual temperature of 21°C in Adrar. This site is also characterised by very hot temperatures in summer which can exceed 40°C and can reach 0°C in winter. For this reason the novel idea is to use this energy source of Fouggara (not in operation) for heating and cooling of buildings in Adrar. It involves pumping the air from the source at a temperature of 21°C to the building for heating in winter and cooling in summer.
From this study a very important conclusion can be drawn. Using the ancient system of Fouggara would contribute to reducing and eliminating the energy demand for heating and cooling buildings in the Sahara desert.

7. General Conclusion

The overall objective was to study and develop energy-efficient systems for domestic hot water production in Algerian similar climates. The considered energy supply system contains a high-temperature solar collector combined with a storage tank. An ancient irrigation system (Fouggara) was also studied for the possible purpose of air conditioning. The work was divided into four parts:

- Characteristic energy load for single-family house in Algeria
- Thermal stratification in the buffer tank
- Solar collector and optical fibers for direct solar radiation
- Fouggara for heating and cooling buildings in Sahara

In the first stage, we defined the bioclimatic house with the various existing sources of positive energy. In order to contribute more information on buildings a study was undertaken on a house located in Tlemcen. The calculated energy demands for heating and/or cooling were:

- Energy demand for heating: 7375 kWh.
- Energy demand for cooling: 589 kWh.

Moreover, we introduced the concept of positive energy which fits fully in the current context of energy management, renewable energy use, heat storage, and sustainable development. Thus, it appeared interesting to model the storage tank by stressing the various technologies used in these systems as well as the research carried out in this field.

Initially, we studied the influence of intrinsic parameters (inlet temperatures, flow rate and the exchanger position). In the light of the obtained results, the confrontation of numerical results with the experimental data for storage tank proved the coherence of a two exponentials model compared to the theoretical model. It also shows that the highest efficiency is obtained when hot fluid is injected into the top and cold fluid is drawn from the bottom of tank.

The performance of a solar system, containing a parabolic mirror, an optical fiber for heat transport and water tank for heat storage were also studied:

- The idea of transporting concentrated solar energy (light) via optical fibers has been a source for numerous studies conducted on different areas since the 1980s.
- The use of optical fibers for (short distance) energy transportation is efficient.
- The maximum water temperature reached was about 70°C.
- The comparison between our system and a conventional system of CSTB, the developed system is more advantageous and reliable.

The gains produced vary with the climate and the type of materials used.

In order to utilise naturally stored heat or cold from the ground, seasonal temperature variations are required. The ground temperature, which is equal to the annual mean air temperature, is warmer than the air temperature during winter and colder during summer. In the north of Libya the mean temperature difference between the coldest and warmest month is about 20°C. In Algeria this difference is even greater, which is favourable.

The heating and cooling demand in northern Africa varies considerably in the different countries, with the greatest cooling demand in the Sahara and the greatest heating demand in the north-west.

The annual energy cooling demand of the test building situated in Adrar gave:

- Heating demand for comfort temperature,
  - October to May: DD = 1102; where C = 7.7 MWh.
- Cooling demand for comfort temperature,
  - June to September: DD = 271; where C = 1.9 MWh.

The study of the ancient Fouggara system aimed to investigate its possible use as a source for heating, cooling and ventilation of buildings.

The Fouggara consists of:

- The source where the water seeps into the channel from a ground water source,
- An underground channel which brings the water to its intended destination,
- An over ground channel which leads to a network of channels feeding the water to particular areas or fields for irrigation.

Since the Fouggara is at a depth below the ground surface where the seasonal temperature changes do not reach, the Fouggara temperature is close to the annual mean air temperature. In Adrar in the south of Algeria the mean annual temperature is 21°C. Pumping air through the Fouggara means that the air temperature will approach this temperature.

- This novel idea of using an ancient construction as a source for heating, cooling and ventilation has not yet been tested.
- It would be attractive to realise such a test on the test building in Adrar.
- However, before performing such a test simulations of how the operation affects the air and ground temperatures should be made.

7.1. Further research

The principal future prospects for this research work revolve around:

- The use of optical fibers with large diameter with a large numerical aperture or a whole of optical fibers.
- Improvement of the optical fiber/storage tank system.
- To code the model and introduce it into the TRNSYS library.
- To look further into the phenomena of solar radiation transport and the concentration in order to optimize the hot water production.
- Using the Fouggara.

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PAPER I

Stratification dans les ballons d'eau chaude sanitaire : Comparaison de modèles existants

Proceedings, 12èmes Journées Internationales de Thermique (JITH), (2005), pp 219-222, Tanger, Maroc, 2005,

STRATIFICATION DANS LES BALLONS D'EAU CHAUDE SANITAIRE : COMPARAISON DE MODELES EXISTANTS

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Résumé – les modèles représentant la stratification thermique dans les ballons d'eau chaude traitent dans leur majorité le cas d'un ballon vertical alimenté en eau froide par le bas et muni d'un puisage d'eau chaude par le haut. La stratification y est modélisée en subdivisant le ballon en tranches horizontales suffisamment fines pour des raisons de stabilité numérique. Pour chaque tranche, sont établies les équations de bilan de masse et d'énergie sachant que l'écoulement est de type piston et se fait de bas en haut. La résolution du système global obtenu fournit les températures caractérisant chaque couche de ballon en fonction du puisage d'eau, des énergies de chauffage (électricité la nuit et appoint solaire ou gaz le jour), de l'emplacement des points d'apport énergétique…

Ce type de modèle est très restrictif dans le cas où le ballon est couplé à un autre ballon de préchauffage ou à un échangeur externe nécessitant un écoulement de haut en bas provenant du côté échangeur.

La présente communication consiste en une étude bibliographique sur l'état de l'art en matière de modélisation des ballons d'eau chaude sanitaire.

La première partie de cette étude présentera différents modèles : multicouches, zonaux, de stockage brassé ou de couches à volumes variables.

La deuxième partie présente une comparaison entre les résultats de 2 modèles de ballon de type multicouche implantés dans les environnements TRNSYS et Ca-Sis dans des configurations classiques de circuits d'eau chaude couplés au solaire.

Mots Clés :

Modélisation, stockage, ballon d'ECS, stratification, simulation, température.

Introduction

La préparation de l'eau chaude sanitaire constitue une des applications privilégiées de l'énergie solaire dans le bâtiment. Ceci tient à la nature du besoin : température de l'eau chaude relativement peu élevée (entre 50 et 60°C), faible variation des besoins au cours de l'année, contrairement au chauffage. Outre les capteurs, le composant essentiel d'un système de production d'eau chaude sanitaire est le ballon de stockage. Pour des raisons d'efficacité, la partie solaire se trouve en partie basse du ballon et l'appoint en partie haute (figure 1).

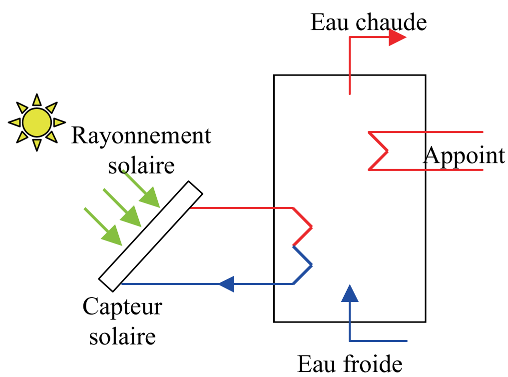


***Figure1 :** Schéma d'un système de production d'eau chaude sanitaire*

L'énergie exploitable du ballon est augmentée si on améliore la stratification des températures. En effet, cette dernière permet d'obtenir une température dans le haut du ballon la plus élevée possible, ce qui a pour conséquence de diminuer l'énergie d'appoint. De plus, une température plus basse dans le bas du ballon améliore le rendement du capteur [1]. Afin d'optimiser un tel système il est nécessaire de modéliser chaque composant qui le constitue. Pour ce qui concerne les ballons d'eau chaude sanitaire, il existe actuellement dans la littérature, 4 grands types de modèles :

- Modèle de stockage brassé
- Modèle à volumes variables
- Modèle zonal
- Modèle multicouches

Après avoir présenté ces modèles dans la suite, deux d'entre eux sont retenus pour comparer leur résultats dans des configurations classiques. Il s'agit des modèles multicouches implantés respectivement dans TRNSYS [2] et CA-SIS [3].

Les différents modèles de ballons

Différents auteurs ont mis au point plusieurs modèles de ballons d'eau chaude sanitaire s'adaptant à leurs besoins comme dans TRNSYS (Types 4, 38, 60 par exemple), dans Ca-Sis (type 81) ou dans CLIM2000.

- Modèle de stockage brassé : Ce modèle considère que l'eau du ballon est caractérisée par une seule température : hypothèse d'un mélange parfait et instantané. De ce fait, la stratification n'est pas représentée.

- Modèle à volumes variables (Type 38) : Ce modèle simule la stratification thermique dans le ballon en considérant des couches à volumes variables dont l'épaisseur s'adapte au pas de temps de calcul ainsi qu'au débit de puisage et débit du fluide. L'avantage de ce modèle est d'utiliser, sans avoir à agir sur le pas de temps, un maillage fin dans les zones de forte stratification et un maillage grossier en regroupant, selon le mode choisi par l'utilisateur, des mailles voisines avec un écart de température n'excédant pas 0,5°C. L'inconvénient de ce modèle est qu'il est limité à deux entrées et deux sorties d'eau [1].

- Modèle zonal (Modèle de Lana Kenjo [4]) : L'hypothèse de base d'un modèle zonal est la division du volume de l'enceinte étudiée en N zones caractérisées par un nœud de température chacune. Le système compte N-1 équations de bilan massique et N équations de bilan énergétique indépendantes avec comme inconnues N températures et autant de débits que de frontières entre zones. En général, le nombre d'inconnues dépasse largement le nombre d'équations de bilan. Les équations de quantité de mouvement ne sont pas prises en compte dans ce type de modèle, c'est pourquoi il est nécessaire d'émettre des hypothèses supplémentaires pour la fermeture du problème [2]. La méthode générale pour cela consiste à imposer des scénarios d'écoulement du fluide. La figure 2 présente les échanges entre les différentes zones pour un cas général où la température de la paroi est supérieure à celle du fluide, à différents niveaux de l'axe vertical. En introduisant l'énergie échangée obtenue dans le bilan énergétique de chaque couche dans l'équation donnant l'énergie échangée par un élément de paroi, nous pouvons calculer le débit venant de chaque élément central vers la zone pariétale selon la relation :

$$\dot{m}(i)=\frac{S(i).H(i)}{C_p.\theta(i)}-\dot{m}_1(i).\frac{[T_c(i)-T_c(i-1)]}{[T_p(i)-T_a(i)].\theta(i)}$$

Avec :

- H(i) est le coefficient de convection de l'élément i (W/m^2 °C)
- S(i) la surface de la paroi associée (m^2)
- T_a la température dans la zone centrale (°C)
- T_c la température de la couche limite (°C)
- T_p la température de la paroi associée à la zone pariétale (°C)
- θ(i) représente la température moyenne adimensionnée de la couche i :

$$\theta(i)=\frac{[T_c(i)-T_a(i)]}{[T_p(i)-T_a(i)]}$$

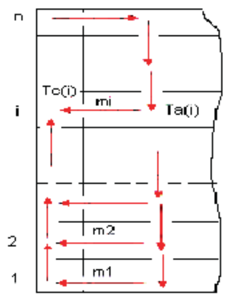


Figure 2 *: schéma des échanges entre zones d'un modèle zonal*

- Modèle multicouches (Types 4, 60 dans TRNSYS et 81 dans Ca-Sis) : L'hypothèse retenue pour ce modèle est de diviser le ballon de stockage en N couches de volumes égaux et d'établir un bilan énergétique pour chaque couche. Ce dernier est réalisé de la façon suivante pour la i^{ème} couche correspondant à la figure 3.

$$(M_i C_p)\frac{dT}{dt}=\phi_{env}+\phi_{cond}+\phi_{échangeur}+\phi_{aux}+\phi_{injecté}+\phi_{flue}$$

Avec :

- M_i est la masse du volume i [kg]
- C_p la chaleur massique du fluide [J/kg.K]
- $\phi_{env}=(U+\Delta U)\,A_s\,(T_{env}-T_i)$ Flux convectif échangé avec l'ambiance [W]
- $\phi_{cond}=\frac{(K+\Delta K)\,A_s}{\Delta x}(T_{i-1}-T_i)$ Flux par conduction entre chaque couche [W]
- $\phi_{échangeur}=UA_{hx}DTLM$ Flux convectif lié à l'échangeur interne [W]
- ϕ_{aux} : Flux injecté par la résistance électrique d'appoint = Q_{aux} [W]
- $\phi_{injecté}=\dot{m}_{1in}\,C_p\,(T_{1in}-T_i)$

  $avec\ \dot{m}_{1in}=\dot{m}_{1out}\ ou\ \phi_{injecté}=\dot{m}_{1up}\,C_p\,(T_{i+1}-T_i)$ Flux lié à une source chaude ou froide [W]
- $\phi_{flue}=U\,A_{flue}(T_{flue}-T_i)$ Flux convectif échangé avec une éventuelle cheminée [W]

L'augmentation du nombre de nœuds diminue le mélange interne, ce qui permet donc de définir une stratification du ballon plus au moins importante. Le phénomène de convection qui entre en jeu entre les couches est pris en compte dans le Type 81 développé par EDF dans l'environnement Ca-Sis.

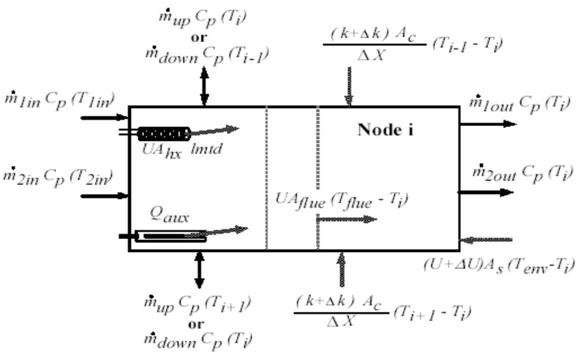


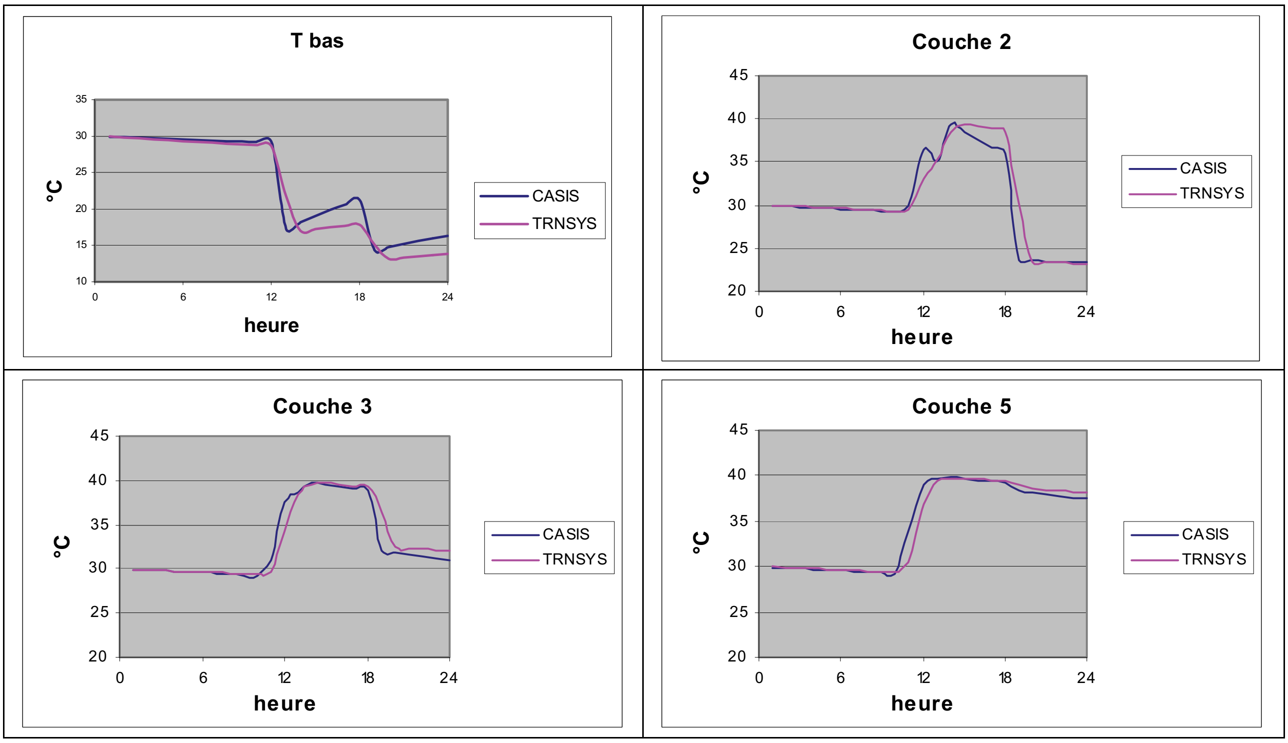
***Figure 3** : représentation du bilan du nœud i dans le modèle multicouche du Type 60 de TRNSYS*

Comparaison des résultats issus du modèle de ballon installé dans Ca-Sis et du Type 60 de TRNSYS

Pour la comparaison des 2 modèles multicouches, l'un prenant en compte la convection et l'autre pas, un exemple simple est utilisé sans faire intervenir d'autre modèle extérieur comme celui du capteur. Cela risquerait d'introduire d'autres différences dans les résultats. Cet exemple utilise un ballon cylindrique de 1000 litres, de 2 m de hauteur, de coefficient de transfert thermique égal à 1.5 W/m².K, sans appoint électrique ou gaz ni échangeur interne. Le nombre de couches considéré est égal à 10. Un appoint solaire permet, à certains moments de la journée de faire rentrer de l'eau dans le ballon à 40°C (de 10 h à 14 h avec un débit de 500 l/h) : l'entrée se fait à une hauteur de 1.7 m (couche 9) et le retour au capteur à une hauteur de 0.3 m (couche 2). Un profil de puisage simple d'un débit de 100 l/h, eau froide entrant à 10 °C, a été généré : de 12 h à 13 h (pendant la circulation du circuit solaire) et de 18 h à 19 h, quand le circuit solaire est arrêté). La température initiale du ballon a été choisie égale à 30°C. L'eau froide pénètre par le bas (couche 1) et l'eau chaude ressort par le haut (couche 10).

Les résultats comparatifs sont donnés en températures pour les 10 couches. Les courbes de température montrent une très bonne concordance au cours des 24h de simulation.

Certains paramètres ne se retrouvent pas exactement dans les deux modes de simulation, et c'est ce qui peut expliquer les légères différences : il est défini un coefficient de convection haut et bas dans Ca-sis que l'on ne trouve pas dans le type 60 de TRNSYS qui prend en compte la conductivité de l'eau et un paramètre de conductivité additionnelle pris par défaut (ce que ne fait pas le modèle de Ca-Sis).



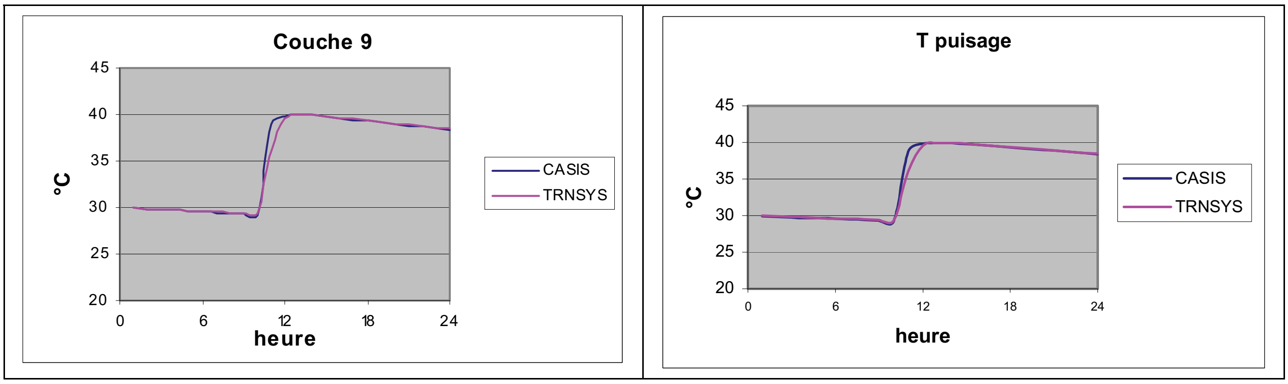


Figure 4 *: évolution de la température en différentes couches du ballon au cours d'une journée. Comparaison des Types 81(CASIS) et 60 (TRNSYS)*

CONCLUSION

Il existe une grande variété de modèles de ballons plus ou moins sophistiqués. Beaucoup d'entre eux ont été développés dans l'environnement TRNSYS et se différentient par les b ilans énergétiques mais au ssi le nombre plus ou moins grand de ports entrée/sortie, de nombre de c ouches maximales ou l e nombre de systèmes auxiliaires d'appoints.

La stratification thermique à l'in térieur du ballon conduit à des résultats sur la quantité d'énergie soutirée plus ou moins grande et donc à une e fficacité plus ou moins grande du ballon.

Des études comparatives basées sur des expérimentations sont encore nécessaires pour adapter au mieux les coefficients de convection haut et bas de chaque couche afin d'obtenir la meilleure description possible des échanges dans le modèle implémenté dans Ca-Sis par exemple.

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PAPER II

Diurnal and nocturnal temperatures modelling of the site of Tlemcen

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Phys. Chem. News 27 (2006) 59-64 *PCN*

DIURNAL AND NOCTURNAL TEMPERATURES MODELLING OF THE SITE OF TLEMCEN

MODELISATION DES TEMPERATURES DIURNES ET NOCTURNES DU SITE DE TLEMCEN

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Abstract

The hourly and monthly middle temperature knowledge is necessary for an adequate dimensionnement of the thermal photo generators, of the agricultural greenhouses and for the air-conditioning of the site. Our work consists in a modelling of these temperatures on the site of TLEMCEN. The treatment of the gradual measure results on several years permitted us the diurnal and nocturnal middle temperature modelling and their variations during the day of the year. This study permits us an optimization of energy captured and the reduction of losses for the storage of energy. This study is necessary for a dimensionnement adapted to cost reduces the generator. The technique of the fittage used permitted us the modelling of these temperatures and the dimensionnement of a generator photo thermal.

Keywords : Temperature; Photo thermique generator; Modelisation; optimisation; Thermique energy; Energy storage; Dimensionnement.

Résumé

La connaissance des températures moyennes horaires et mensuelles est nécessaire pour un dimensionnement adéquat des générateurs photo thermiques, des serres agricoles et pour la climatisation d'un site. Notre travail consiste en une modélisation de ces températures sur le site de TLEMCEN. Le traitement des résultats de mesure échelonnés sur plusieurs années nous a permis la modélisation des températures moyennes diurnes et nocturnes et leurs variations au cours de la journée de l'année. Cette étude nous permet une optimisation d'énergie captée et la réduction des pertes pour le stockage d'énergie. Cette étude est nécessaire pour un dimensionnement adapté à coût réduit du générateur. La technique du fittage utilisée nous a permis la modélisation de ces températures et le dimensionnement d'un générateur photo thermique.

Mots Clés : Température; Générateur photo thermique; Modélisation; Optimisation; Energie thermiques; Stockage d' énergies; Dimensionnement.

1. Introduction

Le climat est un état original de l'atmosphère pour la latitude de l'endroit considéré. Les données météorologiques permettent de préciser par une étude des moyennes de grandeurs mesurées sur de longues périodes et en des lieux différents, l'importance de ces grandeurs sur les prévisions à envisagées.[1]

Un site est caractérisé généralement par les variations de température et d'intensité du rayonnement solaire incident. Ces paramètres sont enregistrés chaque jour en continu ou à intervalles fixes. Ils constituent une banque de données qui permettent d'entreprendre des études statistiques. Pour être capable de contrôler la température de l'air et l'humidité relative est encore un problème pour les processus du séchage conventionnels, chauffage des habitations et pour dimensionnement adéquat des capteurs solaires. Dans cet article nous nous limitons aux températures ambiantes et à l'intensité du rayonnement.[2]

La connaissance de l'intensité du rayonnement incident et des températures d'un site, permettent un bon dimensionnement du générateur photo thermique et du système de stockage d'énergie.[3]

2. Energie et température instantanée

Toute utilisation d'énergie solaire présente par ailleurs une dépendance vis à vis des caractéristiques du climat local et régional [4]. La figure (1) représente les résultats d'un programme de simulation permettant le calcul de l 'énergie instantané maximale aux équinoxes et aux solstices.

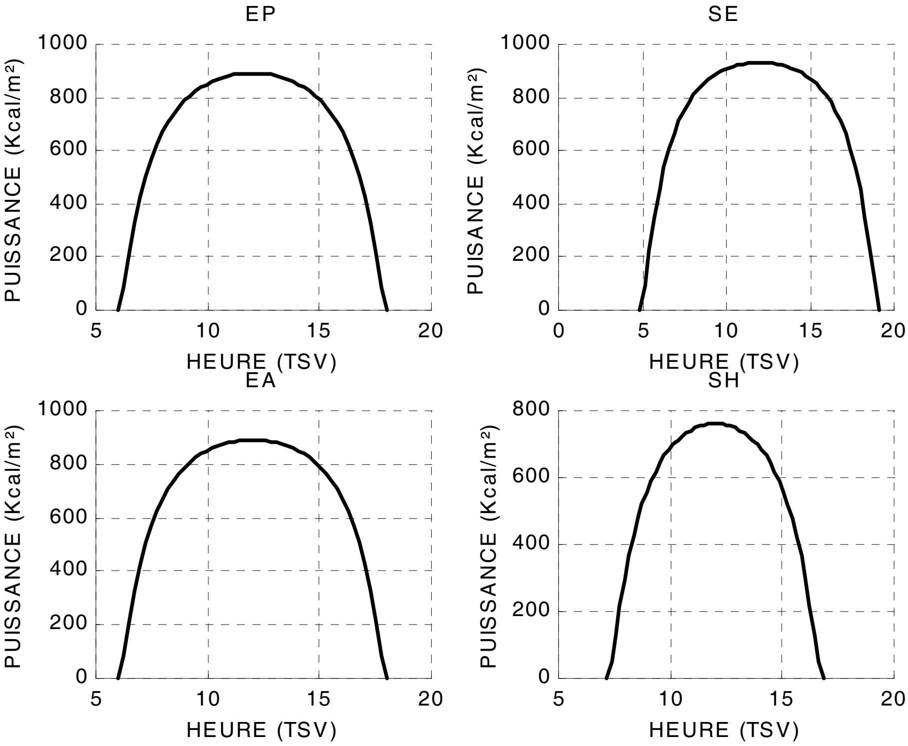


Figure 1 : Energie thermique instantanée aux équinoxes et aux solstices (Site de TLEMCEN – Ciel Bleu clair).

Afin de modéliser les variations de température du site de TLEMCEN, nous présentons le résultat d'une méthodologie de traitement des données de l'office national de la météo (résultats de trois années de mesure 1999-2000-2001) [5].

Les variations de température que nous présentons excluent les phénomènes anormaux comme les incendies temporelles ou les anomalies climatiques. A travers les courbes de variations de températures ambiantes diurnes et nocturnes, nous constatons qu'en général le site de TLEMCEN rencontre une variation de température de jour très importante avant midi TSV et une décroissance légère après le midi TSV. Cependant de grandes variations sont à signalées durant les périodes de printemps et d'automne.

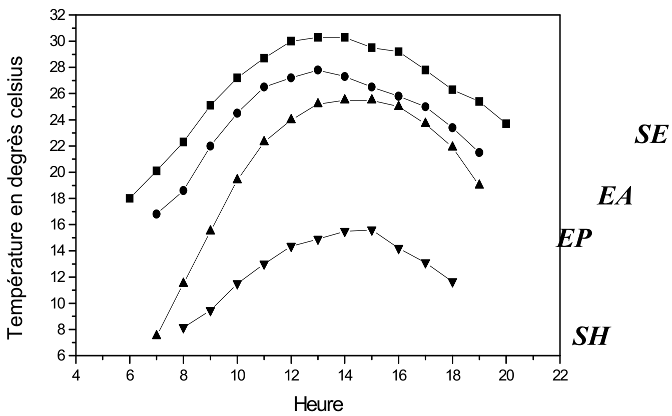


Figure 2 : Température instantanée diurne aux équinoxes et aux solstices au voisinage de midi TL.

Les variations des températures ambiantes diurnes et nocturnes sont très importantes sur le site de TLEMCEN durant les périodes de pri ntemps et d'été. Elles ren contrent une lég ère stagnation durant la période hivernale.

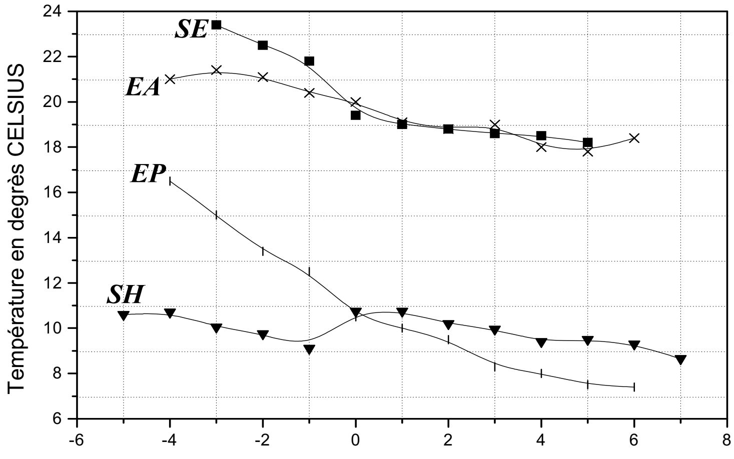


Figure 3 : Température instantanée nocturnes aux équinoxes et aux solstices au voisinage de minuit TL.

Le site de Tlemcen est caractérisé par une montée de température diurnes le matin et par u ne décroissance de cette dernière l'après m idi. Alors qu'une décroissance continue est enregistrée pour les températures nocturnes.

3. Energie et température journalière

La figure (4) représente la variation de l'énergie thermique journalière incidente sur un capteur qui suit le so leil durant toute l'année. L'optimisation d'énergie photo therm ique captée et la réduction des pertes de st ockage, nécessite la connaissance des températures ambiantes et des températures au sol. A travers les résultats de mesure L'écart entre les températures maximales et m inimales du site de Tlemcen ne dépasse pas 10°C quelque soit la saison de l'année.

La connaissance des températures ambiantes moyennes journalières joue un rôle très important pour l'étude de l 'isolation des capt eurs photo thermiques. Ces températures permettent une estimation des pertes dans le stockage de l'énergie thermique [6].

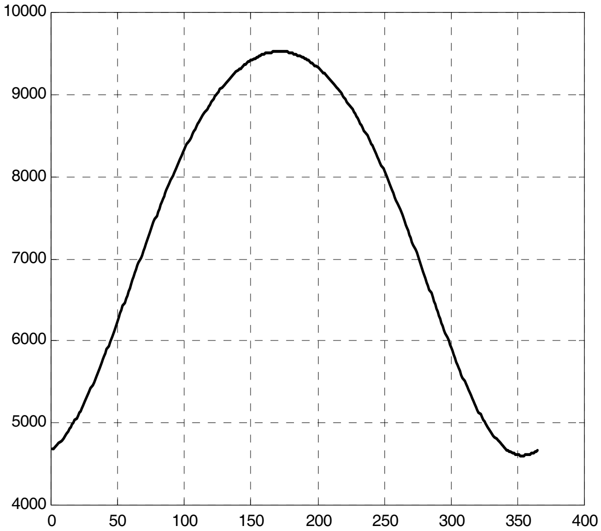


Figure 4 : Energie thermique journalière maximale reçue par un capteur qui suit le soleil.

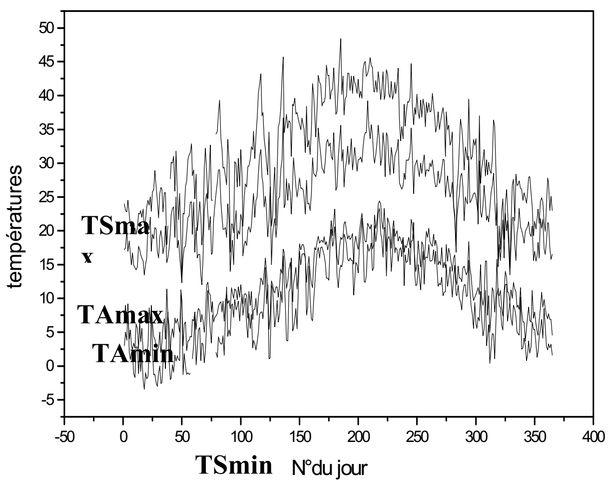


Figure 5 : Variation des Températures journalières (Max et Min) ambiantes et au sol

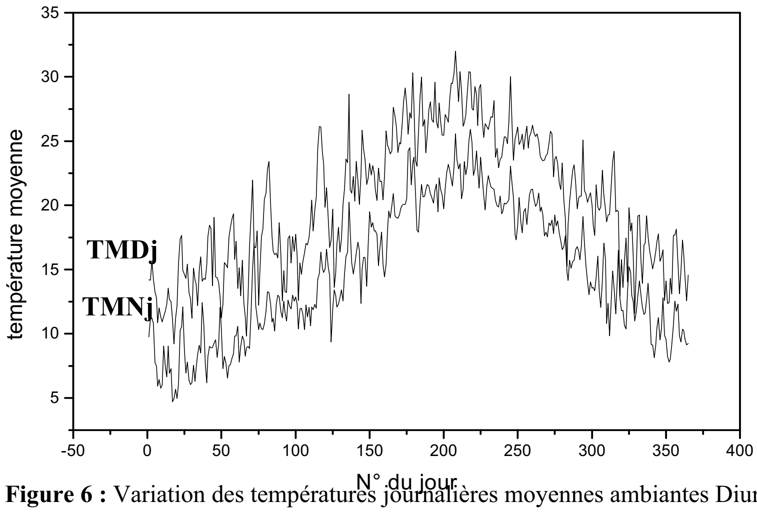


Figure 6 : Variation des températures journalières moyennes ambiantes Diurnes et Nocturnes

4. Température mensuelle

Les températures mensuelles (maximales et minimales) d'un site permettent un dimensionnement économique des systèmes photo thermiques [7].

	Températures ambiantes		Températures au sol	
Mois	**Maximales**	**minimales**	**Maximales**	**Minimales**
Janvier	17.26	4.66	23.03	0.9
Février	19.2	5.57	25.88	0.98
Mars	20.97	8.38	28	4.72
Avril	21.78	10.33	31.28	7.04
Mai	24.88	12.92	33.65	9.27
Juin	29	17	38.6	14.76
Juillet	31.89	19.09	42.24	16.29
Août	30.66	20.04	39.92	17.02
Septembre	29.26	17.44	37.84	14.69
Octobre	25.02	13.81	31.17	10.56
Novembre	21.01	10.71	25.82	7
Décembre	19.03	6.94	24.33	3.97

Tableau 1 : Variation mensuelle des températures maximales et minimales du site de Tlemcen.

La connaissance des températures moyennes mensuelles (maximales et minimales) ambiantes et au sol permettent le dimensionnement adapté d'un générateur photo thermique.

On constate que quelque soit la période de l'année les températures au sol maximales sont supérieures aux températures ambiantes et inversement pour les températures minimales

-Pour les températures ambiantes l'écart maximal de température (13.63°C) est au mois de Février, tandis que l'écart minimal (10.3°C) est au mois de Novembre.

- Pour les températures au sol l'écart maximal de température (25.95°C) est au mois de Juillet, tandis que l'écart minimal (18.82°C) est au mois de Novembre.

5. Modélisation

La modélisation des températures moyennes mensuelles (maximales et minimales) ambiantes et au sol du site de Tl emcen ont donné des allures Lorentzienne de la forme :

$$T = T_0 + \frac{2A}{\pi}\frac{C}{4(M-B)^2+C^2} \qquad (1)$$

avec

A,B,C sont des constantes caractéristiques du site.
T_0 constante caractérisant la grandeur utilisée
M Variable représentant le mois de l'année
T Variable représentant la grandeur utilisée

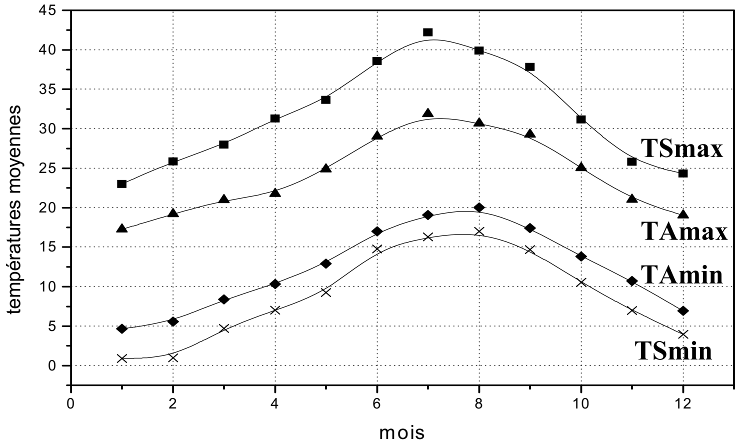


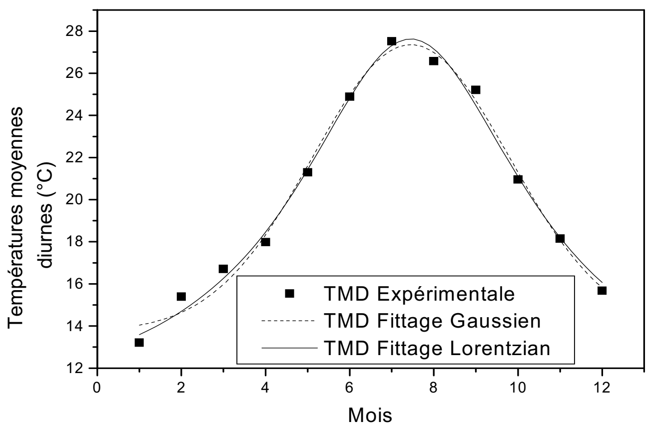
Figure 7 : Variation des températures moyennes ambiantes et au sol maximales et minimales.

Grandeurs	T_0	A	B	C
TSmax	17.62653	250.71388	7.11533	6.57942
TSmin	-5.03868	262.28394	7.55456	7.47958
TAmax	14.06422	175.40342	7.41058	6.22901
TAmin	-0.37205	234.65486	7.55346	7.32958

Tableau 2 : Constantes caractéristiques du site de Tlemcen.

Le traitement des donnés expérimentales des températures moyennes mensuelles diurnes et nocturnes du site de Tlemcen :

*** Pour les températures moyennes mensuelles diurnes :**



Modèle Lorentzien

$$TMD = 9.8 + \frac{801.46}{4\left(M - 7.45\right)^2 + 38.81} \quad (2)$$

Modèle Gaussien

$$TMD = 13.73 + 13.705 \exp^{\frac{-(M-7.45)^2}{11.04}} \quad (3)$$

- **Pour les températures moyennes mensuelles nocturnes :**

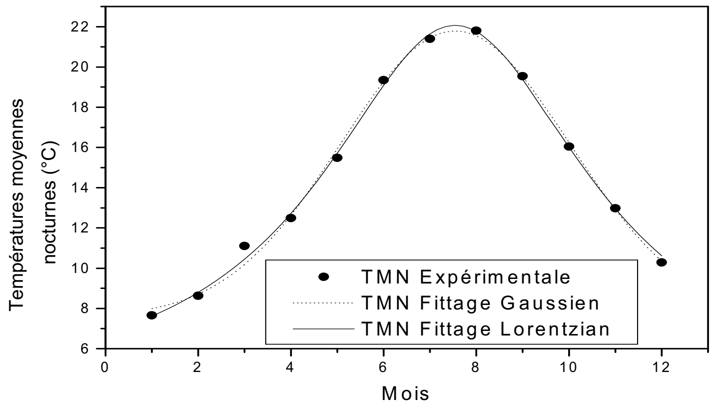


Figure 9 : Variation des températures moyennes mensuelles nocturnes à TLEMCEN.

Modèle Laurentzien

$$TMN = 3.38 + \frac{937.51}{4\left(M - 7.54\right)^2 + 50.27} \quad (4)$$

Modèle Gaussien

$$TMN = 7.56 + 14.22 \exp^{\frac{-(M-7.54)^2}{12.2}} \quad (5)$$

La modélisation des températures moyennes mensuelles diurnes et nocturnes du site de Tlemcen a do nné une allure Lorentzienne plus précise en pé riode d'été et une allure Gaussienne précise en période hivernale:

La connaissance des températures mensuelles moyennes diurnes et nocturnes sont nécessaires pour un dimensionnement adéquat et pour les prévision de stockage.

6. Conclusion

La modélisation des grandeurs caractéristiques d'un site est nécessaire pour un dimensionnement économique du générateur photo thermique. Notre contribution est d'introduire les températures d'un site dans l e dimensionnement adéquat de ces générateurs. La méthode de modélisation appliqué dans ce trav ail est les fittag es Laurentzien et gaussien. La t echnique de fittage basée sur les valeurs mensuelles moyennes de chaque année, et des moyennes de trois années a donné des résultats prometteurs quant aux applications thermiques. L'importance d'introduire les te mpératures d'un site dans la modélisation est nécessaire pour les prévisions à envisagées(climatisation, stockage...).

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PAPER III

Etude d'un système de production et stockage d'eau chaude sanitaire pour le site de Tlemcen

Comples Heliothechnique, 36B, (2007), pp 3-7,

REVUE INTERNATIONALE D'HELIOTECHNIQUE *ENERGIE-ENVIRONNEMENT*- 36B (2007) 03-07
http: \\www.comples.org
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ETUDE D'UN SYSTEME DE PRODUCTION ET STOCKAGE D'EAU CHAUDE SANITAIRE POUR LE SITE DE TLEMCEN

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Résumé

La fonction thermique du bâtiment est l'une des fonctions importantes de celui-ci : il faut qu'au sein du bâtiment soit entretenue une ambiance qui satisfasse l'homme du point de vue confort thermique. Que ce soit pour des besoins de climatisation en pays chauds ou saisons chaudes ou pour des besoins de chauffage en pays froids ou saisons froides. La préparation de l'eau chaude sanitaire constitue une des applications privilégiées de l'énergie solaire dans le bâtiment pour plusieurs raisons. La première tient à la nature du besoin. Les températures sont peu élevées : eau froide à température proche de l'ambiance ; et l'eau chaude entre 50 et 60°C. Une caractéristique intéressante de la préparation de l'eau sanitaire est la faible variation des besoins au court de l'année, contrairement au chauffage.

Mots clés *– Conception, simulation, énergies solaires, concentrateur, chauffage de l'eau sanitaire.*

Introduction

L'idée de transporter l'énergie solaire concentrée à commencée en 1980 par un groupe de chercheurs français [1]. Actuellement la technologie des fibres optiques disponibles pour cette transmission offre une haute qualité de production avec un gros diamètre du cœur de ces fibres [2].
Dans ce travail, nous utilisons des conducteurs de lumière qui ont l'avantage de faibles atténuations, avoir des pertes minimes dans la conversion [3].
Le transport de chaleur sous forme d'onde électromagnétique s'effectue avec une réduction considérable de pertes que le transport de chaleur avec fluide caloporteur.
Afin de réduire les pertes avant de chaleur d'un capteur photo thermique et d'augmenter considérablement la température à l'intérieur du ballon.
Afin de réduire les pertes thermiques de fluide caloporteur d'un capteur photo thermique.
Afin d'utiliser la conversion photo thermique à un endroit bien précis.
Notre travail consiste en une nouvelle conception d'un système de production et stockage d'eau chaude sanitaire à pertes réduite et à rendement élevé.
Le système est parfaitement isolé du milieu extérieur et alimenté par fibre optique en son sommet, nous a donné des résultats prometteuses par rapport au capteur plan photo thermique classique.
L'énergie thermique disponible à partir d'un tel capteur solaire est équivalente à l'énergie incidente diminuée des pertes. Les pertes d'énergie thermique sont dues à la réflexion, la réémission du rayonnement, la convection et la conduction.

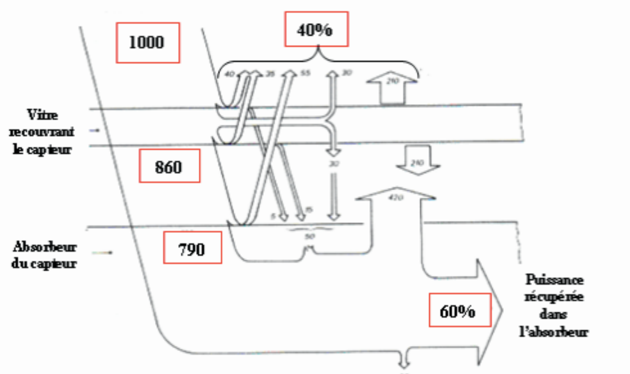


Fig.1. Répartition de l'énergie incidente sur un capteur classique

Nous constatons d'après la figure (1) que pour un flux de 1000W/m^2, au niveau de la vitre presque 14% de ce rayonnement qui est perdu par réflexion des faces avant et arrière de la vitre. Et arrivant jusqu'à l'absorbeur presque 21% qui est perdu et enfin la puissance récupérée est de l'ordre de 60% c'est-à-dire que 40% du rayonnement global incident qui est perdu.

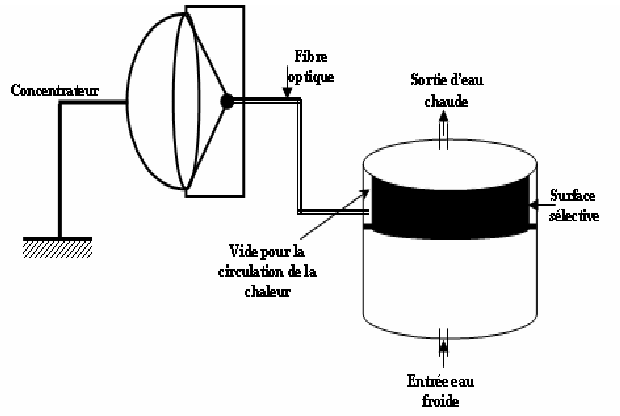


Figure2 : Système de production et stockage d'eau chaude sanitaire

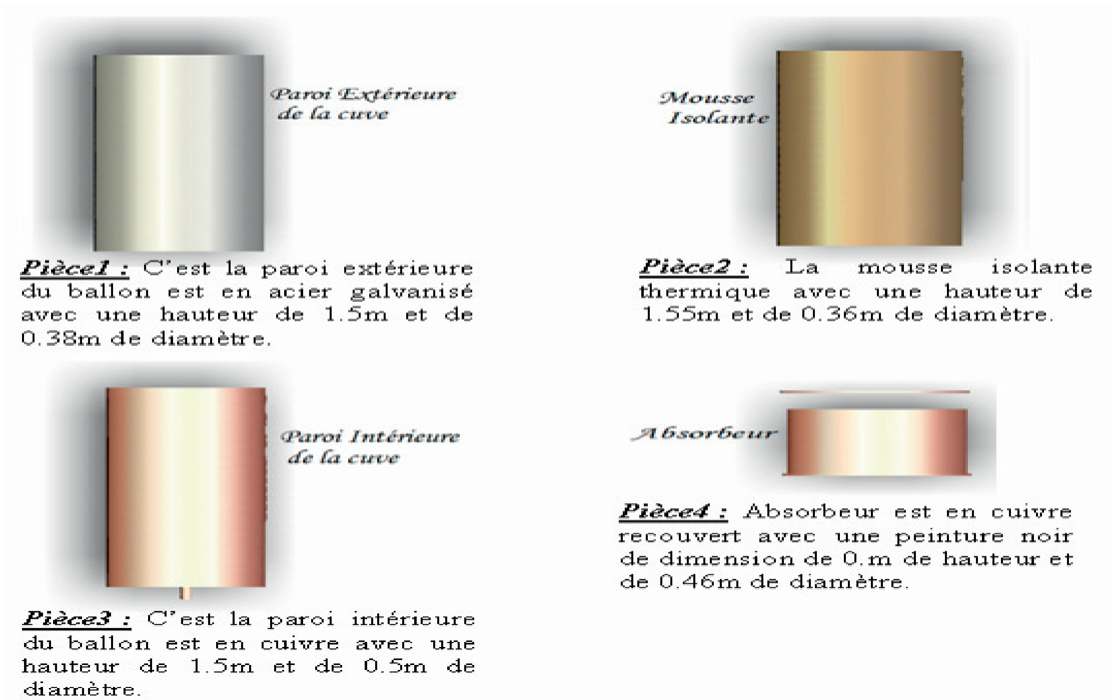
La question qu'on se pose est : peut on éliminer ces pertes ? mais malheureusement la réponse est non mais nous pouvons les réduires dont le but de cette etude est la conception d'un nouveau système de capteur photo thermique à perte réduite et à rendement élevé (Figure2),

Il est constitué :

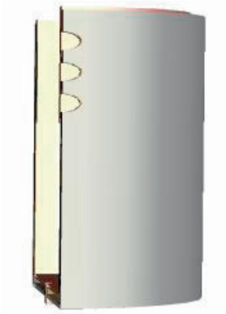
* D'un concentrateur qui est un miroir parabolique qui a les caractéristiques géométriques suivantes : R=366mm ; la distance focale f=R/2.

* Une fibre optique QSF 1000 avec une longueur de l = 5m composée d'un cœur de quartz et de silice possède un cœur de 1 mm de diamètre, fabriqué en Silice de synthèse, d'atténuation moyenne τ égale à 2.10^{-2} dB/m sur l'ensemble du spectre solaire, une ouverture numérique de $\theta_F = 20°$,

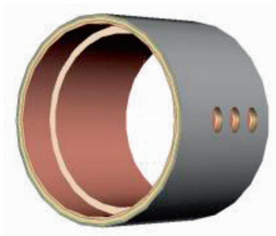
* Un ballon de stockage cylindrique avec une hauteur de 1.5m et d'un volume de 150litres, constitué de quatre pièces (Figure3).



Coupe verticale,
Vue d'en face



Coupe verticale,
Vue de profil



Coupe horizontale,
Vue d'en haut

Estimation d'énergie transportée par fibre optique

L'énergie solaire est concentrée sur une tâche de dimension égale à la surface d'entrée de la fibre [4], [5].

Pour l'estimation de l'énergie thermique à l'entre et à la sortie de la fibre solaire, nous avons modélisé le rayonnement solaire puis nous avons mis au point un programme de simulation de la course du soleil.

1. L'énergie à l'entrée de la fibre

La variation de l'énergie journalière à l'entrée de la fibre optique en fonction des différentes ouvertures du concentrateur est représentée par la figure 3 :

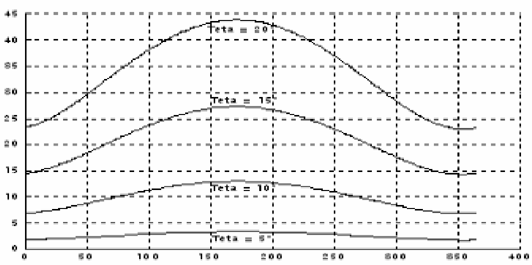


Fig. 3. Energie journalière à l'entrée de la fibre pour différentes ouvertures.

La figure 3 montre l'influence de l'ouverture du concentrateur sur l'estimation des énergies à l'entrée de la fibre optique. Pour une bonne adaptation du concentrateur, l'ouverture de ce dernier doit être égale à l'ouverture numérique de la fibre [6]. Une fibre adapté à l'ouverture du concentrateur transporte un maximum d'énergie solaire de 44 Wh/mm^2 au SE, et un minimum de 24 Wh/mm^2 au SH à Tlemcen et par ciel bleu clair.

2. L'énergie solaire de sortie de la fibre optique

La simulation de l'énergie instantanée de sortie d'une fibre solaire de 5 mètres de longueur adaptée au dispositif étudié et appliqué au site de Tlemcen pour un ciel bleu clair est représenté par la figure 4 ainsi que la course du soleil, et nous constatons que l'énergie de sortie après traversée d'une fibre optique de longueur 5m, au SH ne dépasse pas les 4 KWh/mm^2. Elle est maximale, sa

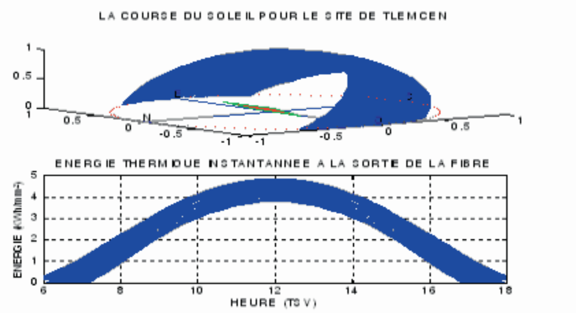


Fig.4 . Energie journalière de sortie d'une fibre de longueur l = 5m à Tlemcen.

valeur est 6 KWh/mm^2 au SE.

Après avoir déterminer la quantité de chaleur transportée par fibre optique jusqu'au ballon de stockage et sachant que toute cette dernière sera absorbé puis transmise par la surface sélective utilisée sous forme de radiations de grandes longueurs d'onde et dont la température peut atteindre les 16 7°C aux équinoxes et elle est de l'ordre de 150°C au solstice été (Figure5).

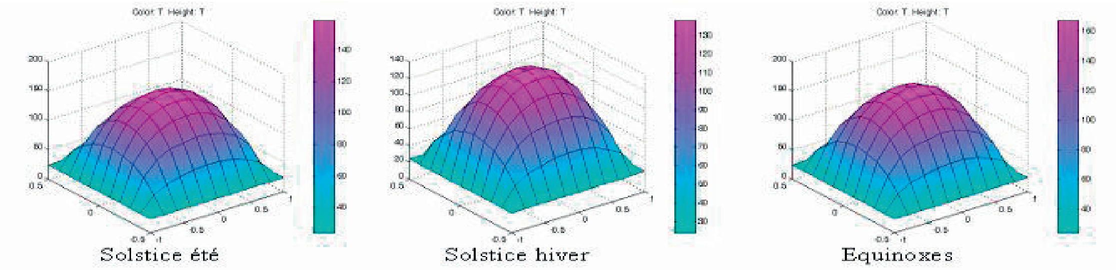


Figure5 : Températures atteintes sur la surface sélective aux solstices et aux équinoxes

D'après la figure (6) qui montrent le profil des températures calculées à l'intérieur du ballon pour différentes couches du ballon, dans la phase dynamique (t<5h) il s'établit dans l'enveloppe une stratification thermique due à l'échange thermique entre la surface sélective et le fluide du ballon. Au-delà de cette phase, les températures deviennent homogènes.

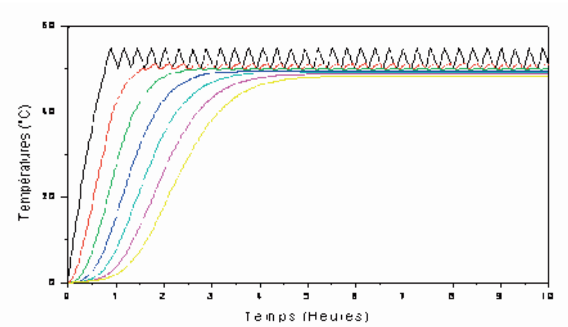


Figure 6 : Variation de la température pour différentes couches

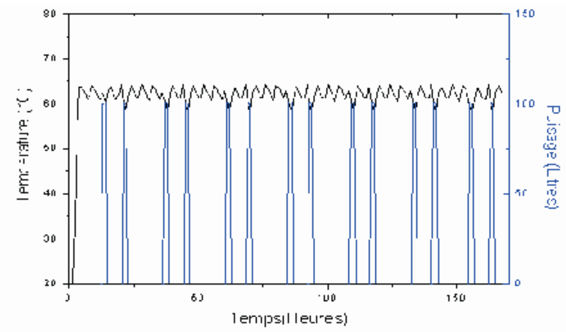


Figure 7 : Profil de la température moyenne avec puisage

Nous avons aussi simulé le profil des températures avec puisage de 100 litres (figure7), deux fois par jour et durant une semaine pour le ballon de stockage dans la partie supérieure. En effet, à travers les résultats, nous remarquons que la température est de l'ordre de 62°C et des qu'elle diminue de 5°C le chauffage d'appoint commence à faire sont effet.

Conclusion

L'énergie solaire concentrée peut être transportée avec des conducteurs de lumière, cette technique s'avère très prometteuse quant aux applications thermiques. Elle devrait permettre une utilisation de l'énergie solaire concentrée avec des rendements de puissance qui peuvent dépasser les 50%.

Notre travail consiste à élaborer un programme qui permet d'estimer l'énergie incidente au fil du soleil, sur un concentrateur parabolique. L'énergie concentrée est transférée puis transportée par une fibre optique vers une surface sélective située à l'intérieur d'un ballon de stockage ou l'échange de chaleur se fait entre la surface et le fluide du ballon.

Le programme de simulation réalisé détermine la puissance de sortie d'une fibre optique, ainsi que les températures à l'intérieure du ballon avec un puisage de 100 litres deux fois par jour durant une semaine.

Ainsi, notre contribution consiste en une nouvelle conception de capteur photo thermique à pertes réduites et à rendement élevé.

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PAPER IV

Requirements energy estimate of heating & cooling for a dwelling in the site of TLEMCEN

Revue des Energies Renouvelables, Numéro special, Mai 21-24, 2007, ICRESD_07, Tlemcen, ALGERIE,

Requirements energy estimate of heating & cooling for a dwelling in the site of TLEMCEN

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Abstract

In order to utilise naturally stored heat or cold from the ground, seasonal temperature variations are required. The ground temperature, which is equal to the annual mean temperature, is then always warmer than the air temperature during winter and colder during summer. In the north of Libya the mean temperature difference between the coldest and warmest month is about 20°C. In Algeria this difference is even greater, which is favourable.
The heating and cooling demand in northern Africa varies considerably in the different countries, with the greatest cooling demand in Egypt and the greatest heating demand in Algeria. In Algeria the heating demand is actually greater than the cooling demand, Fig 1.
In this work, we are interested more particularly in heating or cooling energy requirement estimation for a good dwelling optimisation in the site of Tlemcen.

***Keywords** – Temperature measurements, heat storage, heating and cooling demand,*

Introduction

The environmental and energy new preoccupations related to thermal comfort in buildings and the quality of air require on precise knowledge of temperatures and air movements inside buildings. In this context, we attempt to determine the energy needs for heating in winter and cooling in summer for a building in which the maintenance of an ambitious and acceptable interior temperature approaching: 15°C in winter and 25°C in summer.

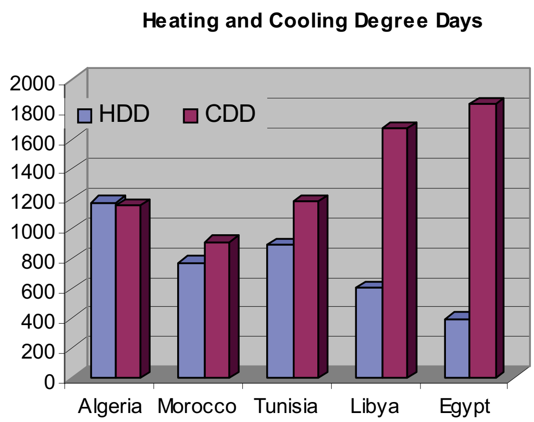

Fig 1. Heating and cooling degree days in North African countries.

In order to have a fuel consumption approximate estimation, which can be interesting to define the volume of storage of fuel or propane gas, it is necessary to know the annual requirements for heating. For this, several parameters are useful:

- Calorific losses of building (KW)
- The number of unified day degrees DJU
- The intermittence or factor of intermittence.
- Fatal losses.

Figure1 : Geographical situation of the Tlemcen city

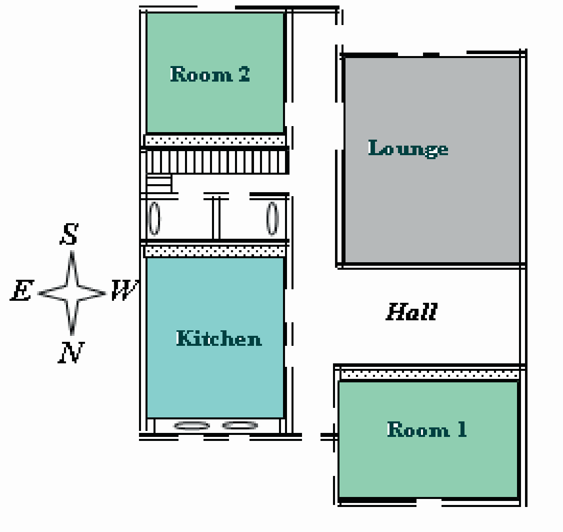

Figure2 : Sight of the building plan

The building

On the architectural plan the building is designed in the following way (Figure2) where the livable surface is of 165m^2. It is necessary to take here into account, on the one hand the livable volume V_h (m^3), and on the other hand the volumic coefficient of loss G taking account of the thickness of the walls, materials used and the number of the openings (Table1). **[1]**

Losses	Lounge			Hall		Room 1		Room 2		Kitchen		Passageway		Bathroom	
	k	S(surface)	KS	S	kS	S	kS	S	kS	S	kS	S	kS	S	kS
Walls	3.5	62.1	217.3	36.4	127.4	52	182	50.2	175.7	52.9	185.15	52.4	183.4	31	108.5
Windows	5.8	6.8	39.5	11.2	64.96	4	23.2	5.9	34.22	8.7	50.46	32	185.6	4.2	24.36
Floor	4	35	140	20	80	25	100	25	100	30	120	15	60	15	60
Roofing	4	35	140	20	80	25	100	25	100	30	120	15	60	15	60
Total lossess			536.36		352.36		405.2		409.92		475.61		489		252.86

Table1 : *Total losses*

Then the total sum of losses (for $\Delta T = 1°C$) of building is P = 2921.75W/°C.
Where the volumic coefficient G of loss:

$$G = \frac{P}{V_h} = 6.52\ (W/m^3 °C) \qquad (1)$$

After the total loss coefficient determination as well as the volumic coefficient G of loss and for simplifying the problem one considers primarily ***the daily ambient temperature Average and the temperature on the ground*** such as they appear in the readings taken by the services of national meteorology during three years of measurement. The knowledge of these temperatures is necessary for an adequate dimensioning of the thermal photo generators, agricultural greenhouses and for the heating and cooling of the site. In order to determine the number of degree day of the site of TLEMCEN, we present the results of a processing data methodology of the national office of the weather (three years results of measurement).**[2]**

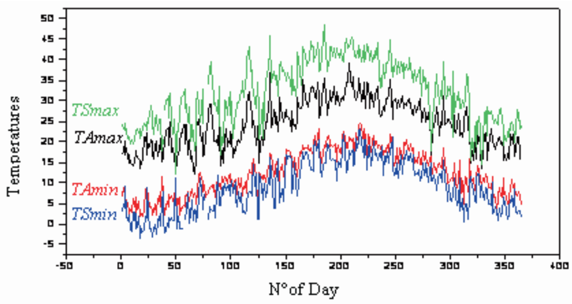

Figure 4 : Daily temperature variation (Maximal and minimal) ambient and in the ground for Tlemcen site

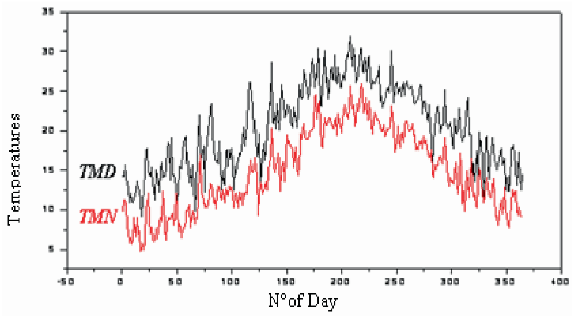

Figure 5 : Daily ambient temperature variation diurnal (TMD) and nocturnal (TMN) for Tlemcen site

According to the results of measurement shown in the figure (3), the difference between the maximum and minimal temperatures of the site of Tlemcen does not exceed 10°C whatever the season of the year.

The knowledge of the daily ambient temperatures average takes a very significant role for the study of the insulation as well as the needs energy calculation for the heating in winter period or cooling in summer. **[3]**

TMD : The daily average temperature ambient diurnal.
TMN : The daily average temperature ambient nocturnal.

It is necessary to calculate the number of degree-days during all the year for our building which is located at Tlemcen with a volume of 448m^3. The following table presents:

Month	***Ambient average temperatures***	***Dj for heating***		***Dj for cooling***	
		15°C	***18°C***	***22°C***	***25°C***
January	10.43	-141.41	-234.41		
February	12.01	-83.54	-167.54		
March	13.9	-50.06	-127.88		
April	15.2	-24	-92.48		
May	17.89	-5.24	-33.82		
June	22.11			30.13	3.48
July	24.46			77.98	12.52
August	24.19			68.93	12.33
September	22.38			18.18	1.53
October	18.5	-12.91			
November	15.56	-19.13	-76.31		
December	12.95	-65.31	-156.54		

Table 2 : *Calculation of the monthly number of degrees – days (Dj)*

Although it is more difficult to take into account of the inhabitants thermal requirements in the building, do they agree not to use certain rooms in winter for the heating or summer for cooling? Taking all into account, one can then evaluate:

$$C = 24 * G * V_h * D_j \qquad (2)$$

- Annual requirements in heating :

- For a temperature of comfort of 15°C, October to May: D_j = 401.6 where C = 2815.33 KWh

-For a temperature of comfort of 18°C, November to May: D_j = 888.98 where C = 6232.02 KWh

- Annual requirements in cooling:

- For a temperature of comfort of 22°C, june to september : D_j = 195.22 where C = 1368.55 KWh

- For a temperature of comfort of 25°C, june to september : D_j = 29.86 where C = 209.32 KWh

Conclusion

This project goal is to have buildings with surplus energy balance to maintain a stable interior temperature. The present study concentrates on the energy needs calculation for a building situated in Tlemcen, and which gave:

- Maximum energy requirements for heating is about 6232.02 KWh.
- Maximum energy requirements for cooling is about 1368.55 KWh.

In order to lead to low yearly consumptions, an objective accessible in the shorter term is to set up solids bases on the envelope (walls, grounds and roofs). In particular, it will be interesting to introduce the positive energy which fits fully in the current context of control of energy, use of renewable energies and sustainable development.

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PAPER V

Experimental study of a domestic hot water storage tank thermal behaviour

Physical and Chemical News (PCN), Volume 46, (2009), pp 15-20.

Effstock2009, June 14-17 2009, Stockholm, Sweden,

March 2009

Phys. Chem. News 46 (2009) 15-20 PCN

EXPERIMENTAL STUDY OF A DOMESTIC HOT WATER STORAGE TANK THERMAL BEHAVIOUR

ETUDE EXPERIMENTALE DU COMPORTEMENT THERMIQUE D'UNE CUVE DE STOCKAGE D'EAU CHAUDE SANITAIRE

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Abstract

Many works were carried out on heat storage during the 20-30 last years, particularly on solar heat storage applications. Theoretical and experimental studies on internal behaviour storage were made and carried as well on laboratory installations as on a large scale. In order to understand the stratification phenomena, an experimental study of tank thermal behaviour of charge and discharge of domestic heat water storage is undertaken.

Keywords: Storage; Stratification; Temperature; Charge; Discharge; Thermal behaviour.

Résumé

Beaucoup de travaux ont été effectués sur le stockage de chaleur pendant les 20-30 dernières années, particulièrement sur le stockage de chaleur dans des applications solaires. Des études théoriques et expérimentales sur le comportement interne du fluide lors du stockage ont été faites et réalisées aussi bien sur des installations au laboratoire qu'à grande échelle. Afin de comprendre le phénomène de stratification, une étude expérimentale du comportement thermique de charge et de décharge d'une cuve de stockage d'eau chaude sanitaire (ECS) est entreprise.

Mots clés : Stockage; Stratification; Température; Charge; Décharge; Comportement thermique.

1. Introduction

Beaucoup de scientifiques ont reconnu que la stratification thermique est nécessaire pour obtenir de meilleures performances lors du stockage de chaleur dans des applications solaires. La stratification dans une cuve de stockage thermique dépend principalement du volume du réservoir, de la taille, de l'endroit, et de la conception des entrées et sorties du fluide. Il y a quatre " facteurs primaires de destratification, " qui contribuent à la perte et/ou à la dégradation de l'énergie stockée:

(a) Pertes de chaleur avec l'ambiance;

(b) Conduction de la chaleur de la couche chaude à la couche froide;

(c) Conduction verticale dans les parois de la cuve qui avec les pertes de chaleur induit des courants convecteurs (mélange);

(d) Mélange présenté pendant les cycles de charge et de décharge qui est généralement la cause principale de la destratification.

L'amélioration des performances des systèmes thermodynamiques implique le plus grand degrés de stratification qui est étudiée par des mesures des champs de températures et de vitesses dans le stockage d'eau chaude. La connaissance de la circulation d'un fluide tel que l'eau et des transferts de chaleur dans une enceinte actionnée par un système extérieur ou par écoulement naturel a fait l'objet de plusieurs études tant numériques qu'expérimentales.

Des études ont été faites sur les phénomènes qui affectent le degrés de stratification thermique [1], et sur le comportement thermique d'un chauffe – eau solaire à faible débit. Les résultats expérimentaux obtenus ont permis de mettre en évidence les phénomènes thermo-hydrauliques dont le ballon est le siège pour les différentes phases de son fonctionnement (injection du fluide à plusieurs niveaux, soutirage, chauffage par l'énergie d'appoint) [2].

Bouhdjar [3] a étudié numériquement le phénomène de stratification dans une cuve de stockage thermique, afin de déterminer les performances thermiques et les configurations les plus efficaces par rapport à l'efficacité de stockage thermique.

En ce qui concerne l'influence de la température d'entrée étudiée à travers l'influence du nombre de Richardson, une performance de stockage élevée est obtenue avec des températures d'entrée élevée et des températures minimales dans la cuve aussi basses que possible [4].

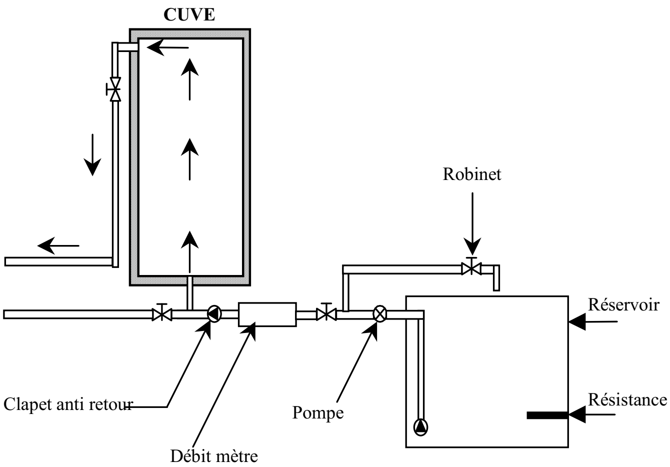
Une étude menée par Ouzzane [5], a permis de comprendre le phénomène de la thermo circulation à travers la distribution de la température dans le capteur ainsi que l'absence du phénomène de stratification. Dayan [6] a également optimisé les installations solaires domestiques. Son étude a montré l'influence du nombre de nœuds utilisé pour la simulation du ballon sur la couverture solaire.

Nous nous limitons dans cette étude à l'évaluation du comportement thermique d'un ballon de stockage d'eau chaude sanitaire.

2. Description du système

Le système comprend l'ensemble des éléments portés sur la Fig.1. Nous avons mesuré la température de l'eau en différents points de la cuve Fig.2. Ces températures sont mesurées à l'aide de 15 thermocouples de type K reliés à l'enregistreur Fluke Hydra Series II à 20 voies. Le débit d'eau circulant dans le système est mesuré à l'aide d'un débit mètre.

(a) : Système réalisé



(b) : Schéma descriptif du système

Figure 1 : Système.

Le système est constitué de:

- Un ballon de stockage d'une capacité de 150 litres, avec un coefficient de pertes UA de 2.75W/K, avec injection du fluide primaire par le bas de la cuve.
- Une pompe de 550 Wmax.
- Un réservoir de (60*60*45cm^3) 160l.
- Un débit mètre.
- Deux résistances de 5KW Maximum.

Afin de bien comprendre le comportement thermique de la cuve, nous avons effectué trois expériences (représentatives du fonctionnement des ballons d'ECS) sur un ballon traditionnel correspondant aux deux scénarios suivants :

- Charge du ballon grâce à un appoint électrique (Résistances)
- Charge + décharge du ballon par puisage d'ECS.

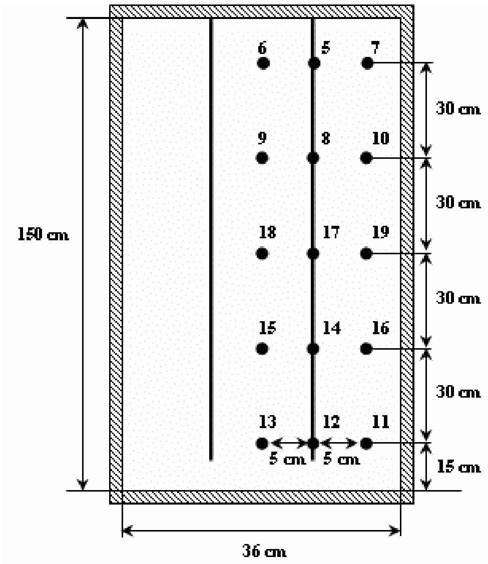


Figure 2 : Schéma représentant la distribution des thermocouples au sein de la cuve.

3. Résultats expérimentaux :

TEST 1 : charge du ballon avec appoint électrique

Avec une température d'entrée de 50 ± 3°C et un débit de 2.6 ± 0.1 l/Min, nous obtenons les profils de températures suivants :

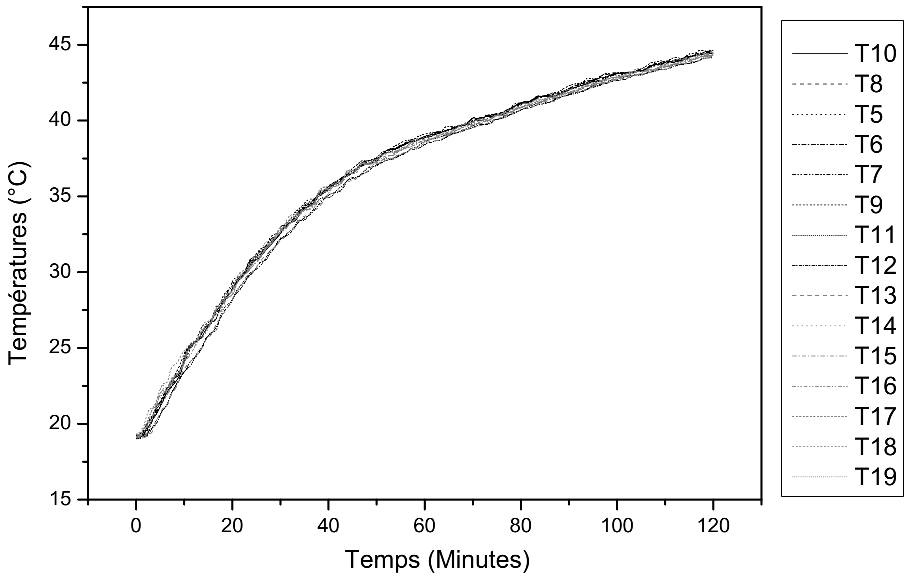


Figure 3 : Evolution des températures dans la cuve pour différentes positions lors de la charge.

Le premier scénario consiste à injecter le fluide par la partie inférieure du ballon. A la fin de la manipulation, la température dans la cuve est presque uniforme (45°C), (Fig.3). Nous constatons aussi que dans les Fig.4 et Fig.5, l'écart radial ne se fait pas sentir sauf pour les couches 2-3 et 4 de l'ordre de 1.7°C, qui est due à la vitesse d'écoulement du fluide entrant dans la cuve. En ce qui concerne l'écart axial (Fig.6 et Fig.7), il est de l'ordre de 1.5°C maximum quelque soit la distance avec l'axe. Cependant, nous remarquons l'absence de stratification dans l'enceinte.

TEST 2 : Charge + décharge

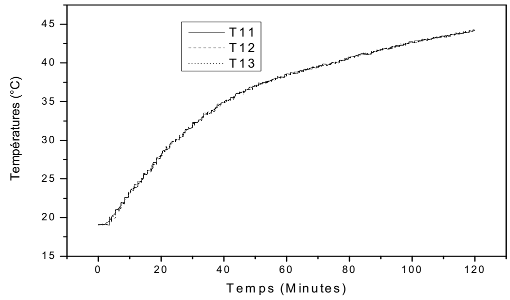
Température d'entrée de 40 ± 3°C

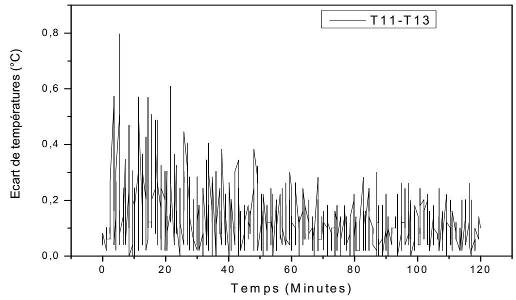
Puisage de 2 fois 2.8 l/Min durant 20 minutes

- Le Premier après 145Minutes
- Puis un deuxième, après un repos de 10 minutes.

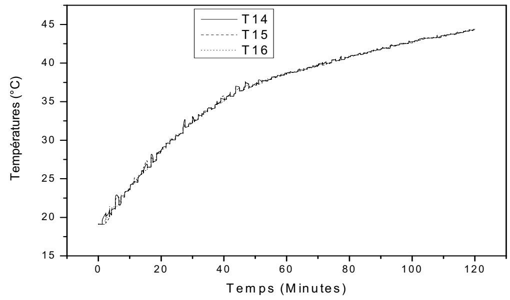
Le débit étant de 2.8 ± 0.1 l/Min

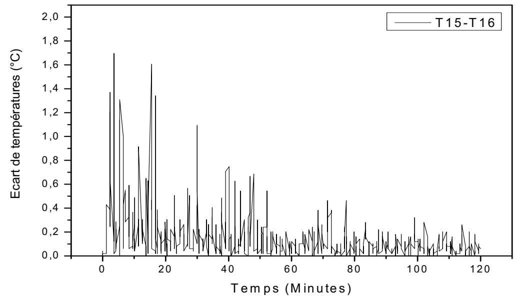
Couche 1



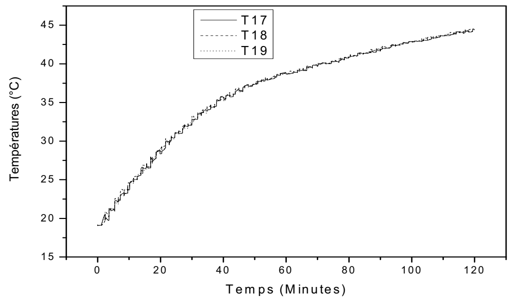


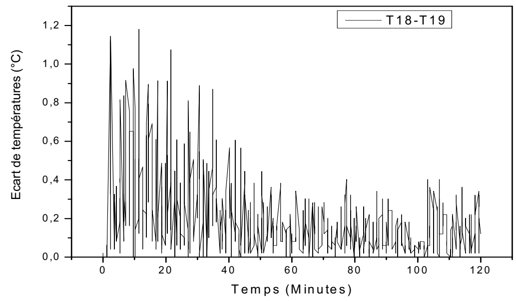
Couche 2



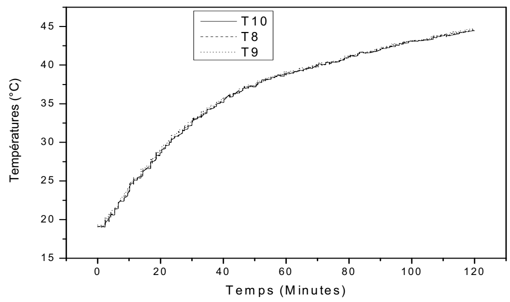


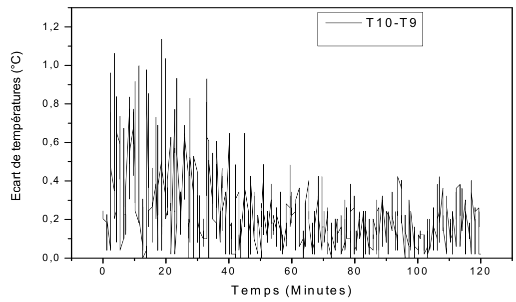
Couche 3



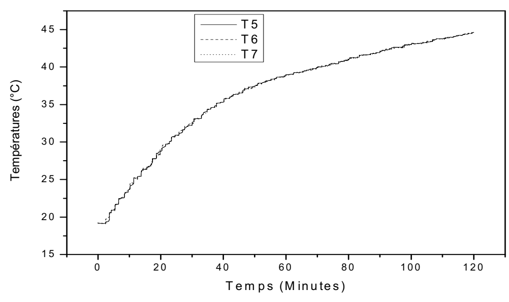


Couche 4





Couche 5



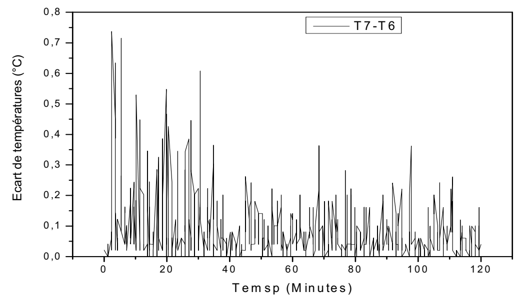
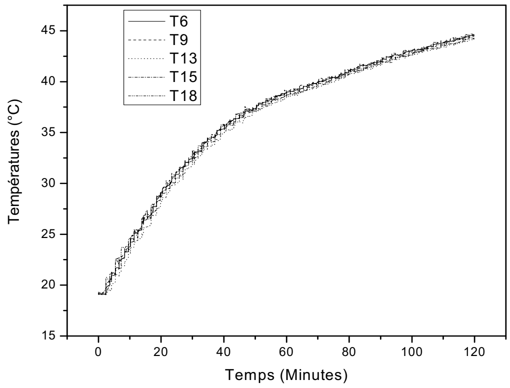
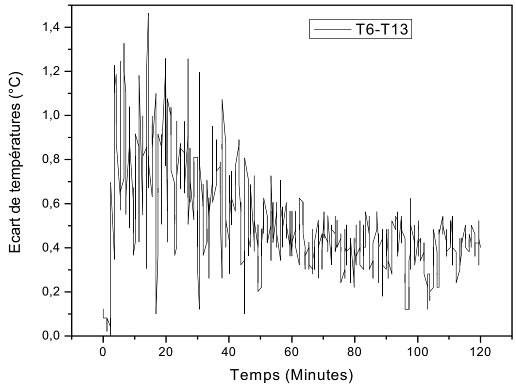


Figure 4 : Evolution radiale de la température dans la cuve pour différentes couches.

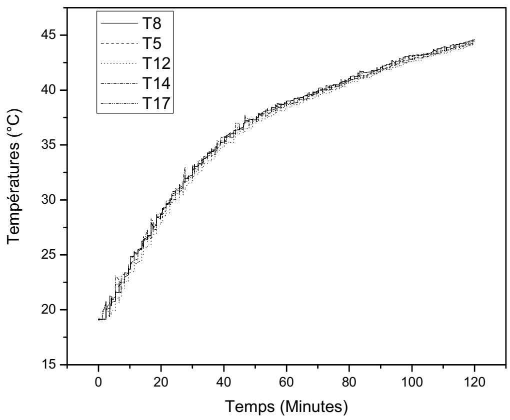
Figure 5 : Ecart radial des températures.

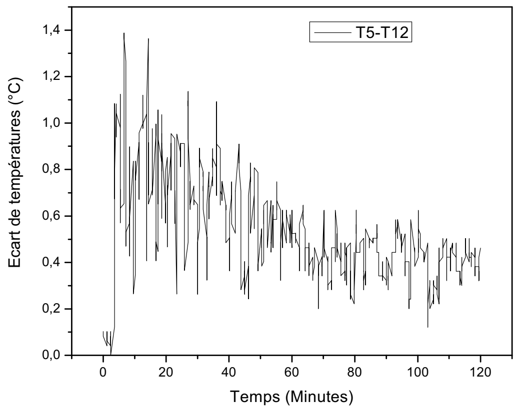
Sur l'axe



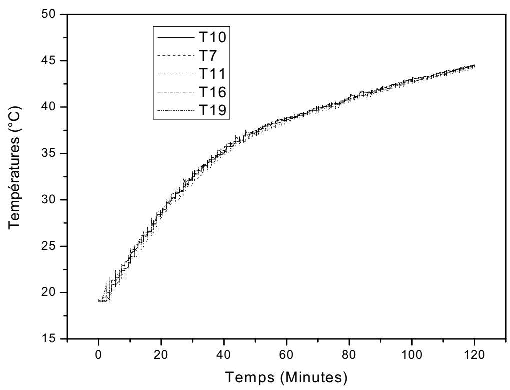


A une distance de 5cm de l'axe





A une distance de 10cm de l'axe



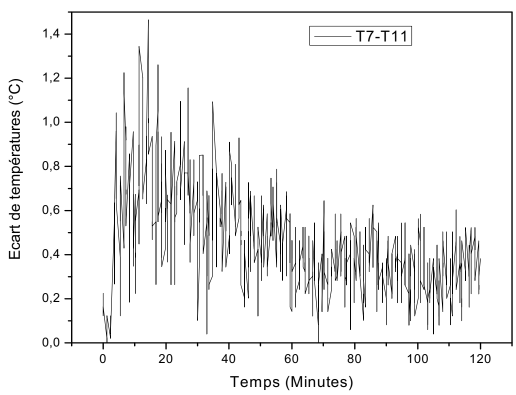


Figure 6 : Evolution axiale de la température dans la cuve pour différentes couches.

Figure 7 : Ecart axial des températures.

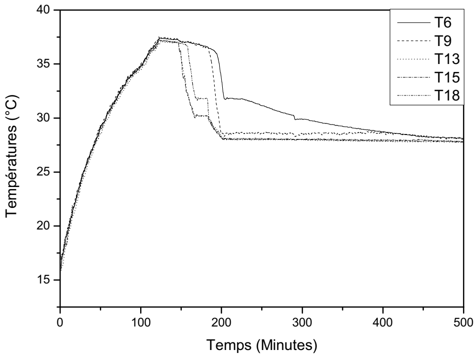


Figure 8 : Profils des températures durant la charge et décharge.

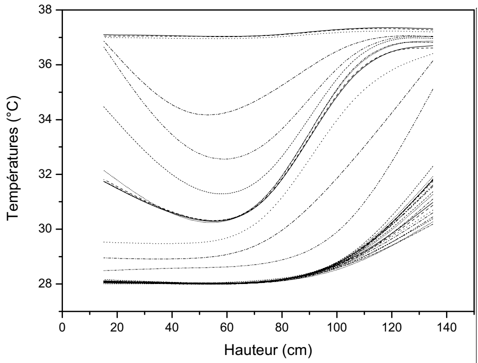


Figure 9 : Evolution temporelle des températures dans le ballon lors des deux puisages d'ECS.

Dans ce second cas, nous constatons que la température de la partie supérieure du ballon diminue de 7°C, alors que les températures des couches inférieures diminuent de 37°C à environ 28°C lors des deux puisages.

4. Conclusion

Les cuves de stockages les plus couramment utilisées dans le stockage d'eau chaude sanitaire sont de forme cylindrique. Afin de mettre en évidence le comportement thermique du système, lorsque celui-ci est soumis à diverses sollicitations bien contrôlées, plusieurs essais ont été réalisés sur la distribution de températures. Cette distribution cependant est fortement affectée par le champ de vitesse dans le stockage qui se compose de convection forcée induite par l'entrée et la convection naturelle due au transport de chaleur entre l'eau et les parois.

Nous avons pu observer le régime dynamique des températures pour un chauffage par appoint et lors d'un soutirage qui a montré l'effet piston induit par le remplissage du ballon.

L'injection du fluide par le bas de la cuve et l'effet de convection mixte induit par la différence de température, crée un mélange à l'intérieur du ballon, d'où l'homogénéité de température sur toute la hauteur. Le phénomène de stratification n'est jamais observé.

En perspective, il sera nécessaire de tenir compte de tous ces phénomènes lors de la modélisation du ballon de stockage d'ECS.

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PAPER VI

Concentration heating system with optical fiber supply

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Concentration Heating System with Optical Fiber Supply

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Abstract

This paper reports on an experimental realization and field testing of a recently proposed solar fiber optic mini dish light concentrator connected to a hot water accumulator. The prototype dish is 150 cm in diameter. In repeated test the collected and concentrated sunlight was transported in a one millimeter diameter optical fiber to a selective surface in the storage tank. This surface absorbs the radiation which remains trapped inside as it heat exchanges with tank fluid which temperature can reach 70°C.

Keywords: Conception, Simulation, Solar Energy, Optical fiber, Hot Water Accumulator;

1. Introduction

Solar energy has for many centuries been harvested by various methods. Different types of solar collectors, normally producing hot water at temperatures between 30°C to 90°C, are commonly used. Large areas of mirrors are used to concentrate the solar heat to very high temperatures in order to drive boilers for production of electricity. The other main development is solar cells (photo-voltaic) for direct production of electricity. More recent development concerns the collection and transmission of visible light through optical fibers for direct illumination [1].

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Optical fibers now play an important part in the fields of both long distance telecommunications and short link networks (LAN's, etc.). More recently, designers have successfully applied less expensive, more mechanically robust fibers to various problems of illumination, such as safety lighting, background lighting, and medical lighting, among others. Typically, optical fiber for communications purposes uses glass fiber with very small core diameters. On the other hand, more recent development work has concentrated on improving the light-propagating properties of larger core diameter fiber made out of plastic [2].

Fig.1a: A SOLUX collector without the protecting acrylic dome [1].

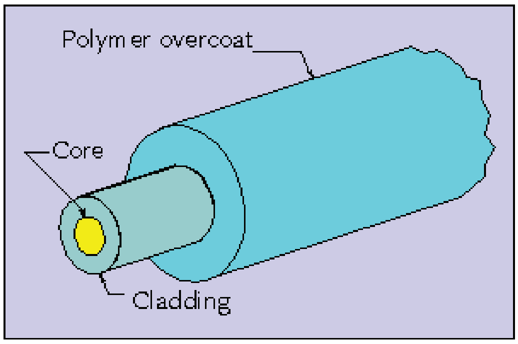

Fig.1b: Construction of optical fiber [6].

In current study hot water production is achieved by a solar (light) collector (Fig.1a) from which the visible light is transported though an optical fiber (Fig.1b) to a water tank where the light dissipates into heat. Since light collectors have the advantage of low attenuation, and no losses in the conversion from light to heated water [3], it is an energy efficient way to produce hot water. The efficiency of conventional flat plate solar collectors means that approximately 60% of incident solar energy is collected and transformed into 60^{o}C water.

The objective of this study was to design a new light collector system for hot water production with considerably higher efficiency. The idea of carrying concentrated solar energy via optical fibers was first suggested in 1980 by a group of French researchers [4], [5], which reviewed performed work in this field, showed that experimental studies began more recently as summarized below:

- Khatri et al. (1993) discussed a solar energy collection system in which optical fibers are used to transport energy from a single-stage and a double-stage compound parabolic concentrator (CPC); [7]

- Liang et al. (1997) emphasized the high flux solar energy transmission by a flexible optical fiber bundle and the research on the associated CPC could greatly expand the existing field of solar energy concentrator applications; [8]

- Gordon et al. (2002a, 2002b) showed that optical fibers used to transport sunlight exhibit considerable light leakage within their nominal numerical aperture and this leakage depends on (i) the incidence angle, (ii) the optical properties of the core and the cladding and (iii) the fiber length; [9], [10]

- Kato et al, (1976) shows that the technology for optical fiber transmission offers a high quality production with a large diameter of the core; [11]

- Feuermann et al. (1998) in studying solar surgery used remote fiber-optic irradiation with highly concentrated sunlight in lieu of lasers. [12]

- Jaramillo et al. (1999) developed a theoretical thermal study of optical fibers transmitting concentrated solar energy; [13]

- Kribus et al. (2000) presented a study on the potential use of optical fibers for solar thermal power generation. The main performance characteristics (numerical aperture and attenuation) and typical costs of currently available fibers were discussed. [14]

2. Prototype concentrator design

The effective incoming radiation to the aperture plane is the beam radiation. An incident beam of solar radiation is a cone with an angular width of 0.53° [15]. The focal length is a determining factor in image size, and the aperture is the determining factor in total energy. It is assumed that the beam radiation is normal to the aperture and the reflection is specular and perfect. Each optical fiber has a pure transparent inner core and a thin transparent outer cladding. The total internal reflection allows us to guide the sunlight through the fiber. The fiber core has an index of refraction n_1, which is greater than that of the cladding n_2. The ratio of the core index and cladding index determines the acceptance/admission angle of radiation θ_{max} at which total internal reflection occurs:

$$NA = \sin\theta_{max} = (n_1^2 - n_2^2)^2 \tag{1}$$

Here, NA is the numerical aperture. On the other hand, the energy rate Q_p hitting a flat receiver of a paraboloidal concentrator, where the optical fiber inlet is placed, is given by

$$Q_p = \pi f^2(\sin^2\phi_r - \sin^2\phi_s)\rho_m G_b \tag{2}$$

Where f is the focal length, ρ_m is the reflectance of the surface, ϕ_r is the rim angle of the paraboloidal dish, ϕ_s is the shading angle because of the receptor size and Gb is the solar beam irradiance [16]. It is important to mention that the rim angle ϕ_r should be equal to or smaller than the optical fiber admission angle.

In optimum conditions, to ensure that the whole radiation gets into the fiber, the maximum rim angle (Fig.2) of the paraboloidal dish must be

$$\phi_r = \theta_{max} \tag{3}$$

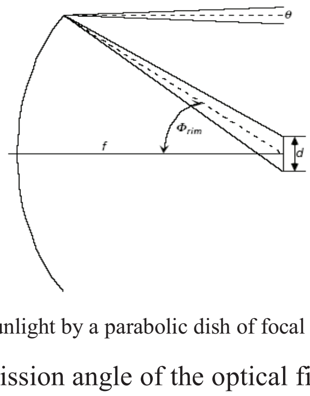

Fig. 2: Concentration of sunlight by a parabolic dish of focal length f and rim angle ϕ_r

This corresponds to the maximum admission angle of the optical fiber. The focal length f and the aperture diameter D_a of the paraboloidal dish are related by

$$\frac{f}{D_a} = \frac{1}{4\tan(\phi_r/2)} \tag{4}$$

For a flat receiver (at the focal plane of a paraboloidal concentrator) the receptor diameter D_r is given by

$$D_r = \frac{D_a \sin(0.267° + \delta/2)}{\sin\phi_r \cos(\phi_r + 0.267° + \delta/2)} \quad (5)$$

Where δ/2 is dispersion angle as a measure of the angular errors of the reflector surface and 0.267° is the half-angle of the incident beam cone of the solar radiation. It is important to indicate that the receptor diameter D_r should be equal to the diameter D_{0f} at the input section of optical fiber:

$D_{0f} = D_r$ (6)

Taking into account Eqs. (4) and (5), we can write the optimal focal length f_0 as

$$f_0 = \left(\frac{D_{0f}}{4\tan(\theta_{max}/2)}\right)\left(\frac{\sin\theta_{max}\cos(\theta_{max} + 0.267° + \delta/2)}{\sin(0.267° + \delta/2)}\right) \quad (7)$$

The energy rate at the inlet of the optical fiber Q_p can be written as

$$Q_p = A_{0f}\,\rho_m\,G_b\,F_s\,C_{max}, \quad (8)$$

$$F_s = \frac{\sin^2\theta_{max} - \sin^2\phi_s}{4\tan^2(\theta_{max}/2)}, \quad (9)$$

$$C_{max} = \frac{A_a}{A_{0f}} = \frac{\sin^2\theta_{max}\cos^2(\theta_{max} + 0.267° + \delta/2)}{\sin^2(0.267° + \delta/2)}, \quad (10)$$

Where A_{0f} is the area of the input optical fiber, A_a is the aperture area of the paraboloidal mirror, C_{max} is the maximum ratio of geometrical concentration and F_s is the view factor for a flat receiver of paraboloidal mirror and both depend on θ_{max}. A radiative flux Q_i as high as possible can be obtained by the maximum value for θ_{max} from the view factor F_s and the maximum geometrical concentration C_{max} [5]:

$$F_s C_{max} = g(\theta_{max}) = \left(\frac{\sin^2\theta_{max}}{4\tan^2(\theta_{max}/2)}\right)\left(\frac{\sin^2\theta_{max}\cos^2(\theta_{max} + 0.267°)}{\sin^2(0.267°)}\right), \quad (11)$$

g_{max} is the combined expression of the view factor and concentration ratio (dimensionless).

At this point, we consider a perfect/ideal paraboloidal mirror with zero dispersion (δ/2=0) and for the shading angle $\phi_s = 0.$ (12)

To determine the maximum value for the admission angle θ_{max} and to obtain the maximum energy rate at inlet optical fiber, it can be set as

$$\frac{dg(\theta_{max})}{d\theta_{max}} = 0, \quad 0 < \theta_{max} < 90,$$

We obtain $\theta_{max} \approx 40°$, and $g_{max}(40°) = 8638$ (Kandilli et al.,2009). On the other hand, from the definition of decibel losses per unit length, η_{of} optical efficiency of optical fiber and the energy rate Q_0 at the end of the optical fiber can be expressed as:

$$\eta_{of} = \frac{Q_0}{Q_p} = 10^{-L\,dB_{loss}/10} \quad (13)$$

Where L is optical fiber length and dB_{loss} is the optical fiber attenuation. The optical efficiency of the collector system for the optimum value can be given as

$$\eta_{opt} = \rho_m 10^{-L\,dB_{loss}/10} F_s(\theta_{\max}) \quad (14)$$

Where F_s is the view factor. Overall system efficiency can be expressed as

$$\eta_s = \frac{Q_0}{A_a G_b} \quad (15)$$

Maximum and minimum size of the image on the focal point given firstly by Gordon et al. was reconstructed by modifying the equations with dispersion angle. The diameter d_{min} of the approximately uniform-flux core region of the focal spot is

$$d_{\min} = 2D_a\left[\frac{1}{4\tan(\phi_r/2)}\right]\sin(0.267° + \delta/2) \quad (16)$$

The focal spot diameter d_{max} that accepts essentially all reflected rays is:

$$d_{\max} = \frac{D_a\left[1+\frac{1}{\tan^2(\phi_r/2)}\right]^2 \sin(0.267° + \delta/2)}{2\left[\frac{1}{\tan^2(\phi_r/2)}-1\right](\frac{1}{\tan(\phi_r/2)})} \quad (17)$$

When designing for maximum flux concentration from the dish, the diameter of the optical fiber should be selected as d_{min} and for a maximum efficiency design the fiber diameter will be closer to d_{max}. Optical fibers with large numerical aperture, large core diameter and small attenuation should be used for effective transmission of concentrated solar energy.

3. Outline of Studied system

The studied light collector system for hot water production, which is outlined in Fig.3, contains:

- Light concentrator represented by a parabolic mirror that has the following geometrical characteristics: Diameter (D) = 1.5 m.
- Optical fiber SPCH 1000/1035/1400Z from Fiberguide company with:
  - Length = 2 m,
  - Silica core with a diameter of 1 mm,
  - Average attenuation $\tau = 6.2*10^{-3}$ dB/m on all solar spectrum,
  - Numerical aperture θf=20°.
- Cylindrical storage tank composed with:
  - Height = 1.5m ,
  - Volume = 0,15 m^3,

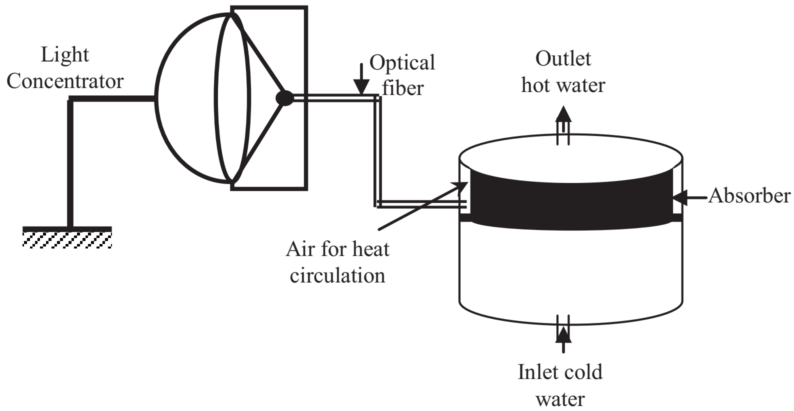

Fig.3: Light collector for hot water production and storage

Table1: Parameters of parabolic dish

Parameters		Parameters	
D_a (m)	1.5	C_{max}	1858
D_r (m)	0.035	NA	0.4
f (m)	0.88	g_{max}	8309
f_0(m)	0.9	d_{min} (m)	0.020
Φ_r (°)	46	d_{max} (m)	0.034
Φ_s (°)	0	ρ_m	0.85
θ_{max} (°)	45	η_{of} (%)	99.7
$\delta/2$	0.4	η_o (%)	61.7
dB_{loss} (dB/m)	$6.2*10^{-3}$	η_s (%)	61
L (m)	2		

Table 1 presents measured and calculated parameters of paraboloidal dish. The optical fiber used in the present study is the type SPCH 1000/1035/1400Z from Fiberguide Company. The length of optical fiber is 3m and the diameter is 0.001m. The maximum admission angle of optical fiber was determined by Eq. (1) and the rim angle of the dish by Eq. (4). The attenuation of the optical fiber is indicated as $6.2*10^{-3}$dB/m by the manufacturer data. The ideal transmission efficiency of the optical fiber was found to be 99.7%. For maximum concentration ratio, the rim angle was used in Eq. (10) and was found to be 1858.
In this case and using θ_{max} in Eq. (9) the view factor Fs was found to be 0.728. Consequently, optical efficiency of the whole system was calculated as 61.7% in optimum condition. Overall system efficiency was found to be 61% according to Eq. (15).

The size of the image has uniform flux as given by d_{min}. As indicated in Table 1, d_{min} was calculated as 2.0 cm according to Eq. (16). If the aim is to receive all flux concentrated on the focal point, it should be considered maximum image size. For the existing system, d_{max} was calculated as 3.4 cm by considering dispersion effects according to the Eq. (17).

The cylindrical storage tank is composed of four parts as shown in Fig.4.

- ***Part 1 :*** External wall of the tank made of galvanized steel(H=1.50m; D=0.60m)
- ***Part 2 :*** Thermal insulating foam with a height (H=1.55m; D=0.65m)
- ***Part 3 :*** Internal wall of tank made of galvanized steel (H=1.50m; D=0.50m)
- ***Part 4 :*** Absorber of galvanized steel with (H=0.50m; D=0.46m)

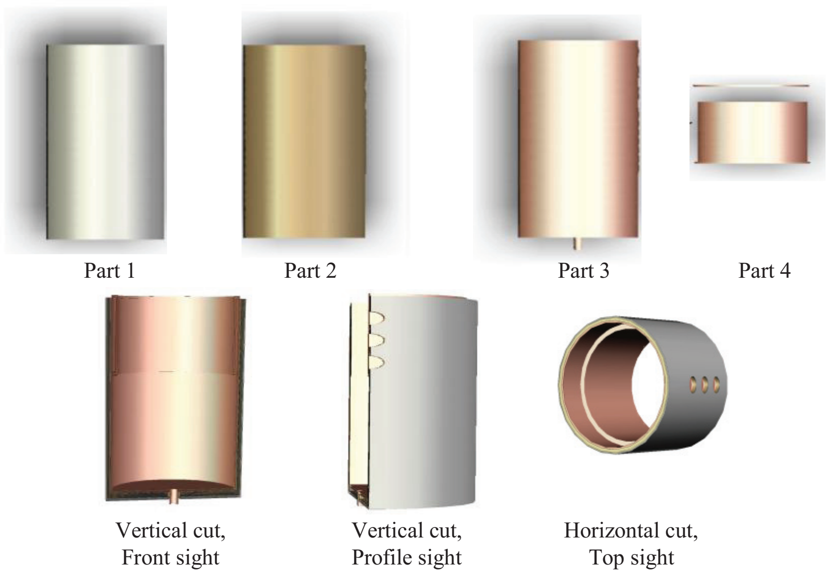

Fig.4: Description of how the cylindrical tank is assembled.

4. Experimental study of the prototype

After designing and assembling the two modules (concentrator + storage tank) (Fig.5), we undertook an experimental study.

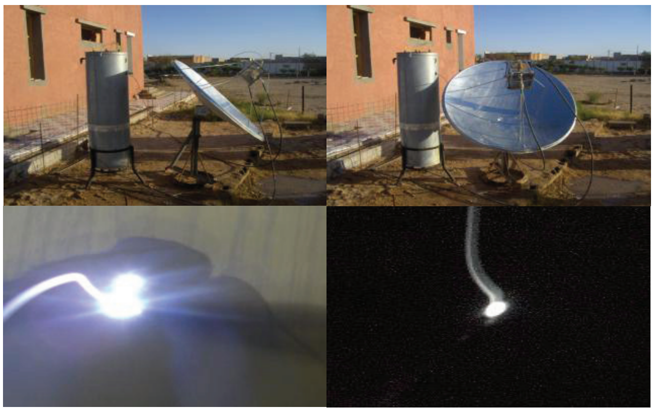

Fig.5: Photos of the designed system and sunlight in the optical fiber outlet

Practical considerations militate against large and small dish rim angles. At small rim angles, relatively high efficiency can be realized at peak concentration; but for a given fiber size, smaller diameter dishes must be used. Dish focal length is large relative to the diameter, which creates a practical problem in terms of module depth.

To estimate realistic system performance, we reduce the theoretical limits by the following factors:

1. An absorptive loss in the dish mirror of 5%;
2. Fresnel reflective losses at the module glazing (3% with an AL coating);
3. Fresnel reflective losses at the fiber ends (a total of 5% with AL coatings);
4. Ray rejection in the secondary concentrator (in the range of 1 to 4% depending on the dish rim angle, and about 2% at ϕ =45°) [17];

5. Absorption in a remote second stage of about 2 to 4%, depending on ϕ (absorptive losses in a dielectric proximate second stage are negligible) (Welford and Winston, 1989)[17];

5. Results and discussion

Adrar city is located in south-west of Algeria, at 1600 km from Algiers capital, with a latitude of 27,88°, a longitude of -0.88° and an altitude of 270 m. It is characterized by particular climate conditions, i.e. hot and dry summers, cold and rigorous winters.

Any use of solar radiations must take into account the local and regional climatic conditions. It is widely known that all the observed atmospheric processes are the consequences of the received solar radiation. Thus, the measurement of the solar radiation presents special interest for the environmental researchers [18].

As shown in Fig.6, the great increase for whole radiations is located between February and March; and the maximum is obtained in July, with a slight stability for the global collector slope between March and October.

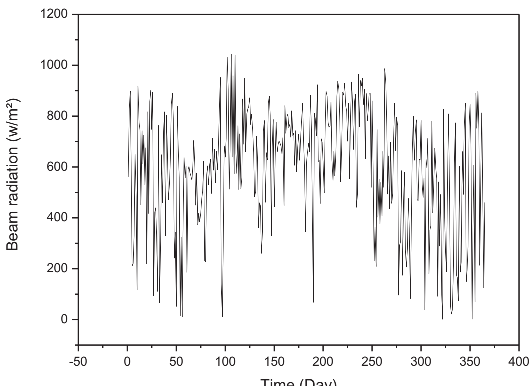

Fig.6: Hourly average solar beam radiation (W/m²)

Hourly global irradiance data of 2004 obtained meteorology stations were used to put forward the hourly normal beam radiation reaching the aperture plane of the paraboloidal dish. Global irradiance data were separated to beam and diffuse component by using the Clear Sky Model [Meteonorm]. The normal beam radiation calculated hourly was used to evaluate the monthly average hourly output power obtained from the paraboloidal dishes.

As shown in Fig.5, the great increase for whole radiations is located between February and March; and the maximum is obtained in July, with a slight stability for the global collector slope between March and October. These data were presented to estimate the output power of the system. And taking into account all the factors that reduce the system performance, we can determine with Eq.(8) and Eq.(13) the annual mean of the output power from optical fiber.

After the determination of the output power quantity transported by the optical fiber to the storage tank, where it is absorbed then transmitted by the selective surface, the temperature in Fig.7 can reach 170°C at the equinoxes and 150°C at the solstice.

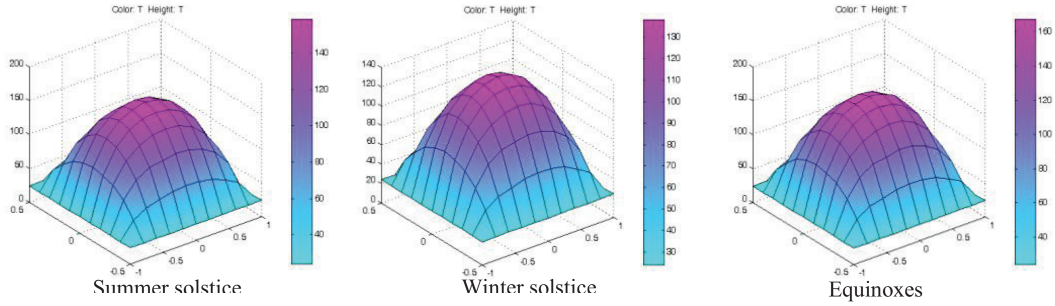

Fig.7: Temperatures reached on a selective surface at the solstices and the equinoxes

In performed tests the water flow rate through the accumulator tank was 0.6 l/min. Fig.7 shows the inlet and outlet temperature variation on 22 January 2009. Under the light of these results, we noted the existence of a gap in the beginning of the manipulation of over 10 °C which became 50 °C at noon. It took approximately four hours to fill the storage tank with a flow rate of 0.6 l/min to obtain an average daily temperature of 36.5 °C. The decreases observed in Fig.8 for the outlet temperature T_s are resulted by the manual sun tracking system.

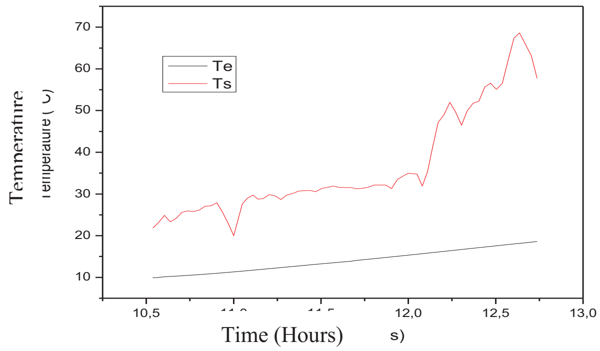

Fig.8: Measured water inlet and outlet temperatures, 22 Jan 2009.

6. Concluding remarks

The major finding of the current analysis is that the use of optical fibers in solar thermal concentrating systems for heating is technically feasible, under specific circumstances. In the first phase of our solar fiber-optic paraboloidal dish program the main conclusions of performed study are:

1. A solar fiber-optic heating system was designed, constructed and tested
2. Collection, concentration and transmission of sunlight were demonstrated.
3. The mathematical model of paraboloidal dish system was described.
4. The measured output power from the dish was evaluated against radiation data from the local meteorological station supplied by the Renewable Energy Research Unit (URER/MS) in Adrar.
5. Optimizing the receiver and dish sizes are important in order to maximize the system efficiency.
6. The calculated optical efficiency of the system was 62% at optimum conditions.
7. The calculated mean power from the paraboloidal dish was 885kW which for an aperture area of 1.7 m^2 means 520 kW/m^2.
8. The heat was stored in a 0.15 m^3 storage tank.
9. The temperature of outlet water from the storage tank reached 70°C.

During the experimentation we encountered some difficulties e.g.

- Dust accumulation on the mirror and at the fiber tip.
- The parabolic mirror was made from a satellite dish which surface was clothed by a foil tape. Its aluminum plies reduced the reflectivity that meant extra energy loss.
- The optical fiber was aligned at the concentrator focus by placing a cone (Length=0.15m Diameter=0.075m) in the parabolic focus, mouthing the mirror. It was problematic to attach the fiber exactly at the focus.
- The manual measurements meant fewer measurements and more errors.
- Instead of using the desired optical fiber with relatively large diameter and high numerical aperture (NA), two optical fibers with a NA=0.4 manufactured by FiberGuide Company, USA, was used.

Current study showed that concentrated solar energy can be transferred as light by optical fibers to a selective surface (absorber) inside a storage tank, where the light is converted to heat. Such systems should have great potential in a wide range of solar energy applications.

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PAPER VII

Using Fouggara for heating and cooling buildings in Sahara

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Using Fouggara for Heating and Cooling Buildings in Sahara

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Abstract

In order to utilise naturally stored heat or cold from the ground, seasonal temperature variations are required. The reason is that the ground temperature is then warmer than the air temperature during winter and colder during summer. The heating and cooling demand in North Africa varies considerably with the greatest cooling demand in the East and the greatest heating demand in the West. In parts of Algeria the mean temperature difference between the coldest and warmest month is greater than 20°C, which is favourable. In current work it was shown that the ancient Fouggara system, even today would be interesting for heating and cooling of buildings in the Sahara desert.

Keywords: Energy, Fouggara, heating and cooling, buildings, temperature;

1. Introduction

The development of sustainable buildings partly means to minimize required energy for heating in the winter and cooling in the summer. Increased thermal insulation and air tightness of the envelope reduce heat losses. However, the renewal of air is necessary to maintain good air quality. Air is also used to distribute heating and cooling to the building. Therefore, improving thermal performance of buildings require the control of air exchange.

The ventilation solutions in sustainable buildings provide better control of the air flow, both hygienic and thermal. [1]

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In this context, we attempt to determine the energy needs for heating and cooling for a building to maintain an indoor temperature between 18°C and 25°C.

The estimated annual fuel consumption, defines required storage volume of fuel or propane gas. For this, several parameters are useful:

- Heat losses from building (kW)
- Number of degree days DD
- Annual distribution of energy consumption
- Uncontrolled losses

The heating and cooling demand of any building also depends on its location. In this study the site is in the Adrar region in Algeria.

Objectives

The objective of current study was to investigate the feasibility of the ancient Fouggara system for heating, cooling and ventilation. The heating cooling demand of a building in Adrar, Algeria, was estimated and the function of the Fouggara was based in real operation data from a Fouggara in the Sahara desert.

2. Geographical characteristics

Adrar is located in south-west of Algeria (Fig.1), 1600 km from Algiers, with latitude of 27.81°, a longitude of -0.18° and an altitude of 279 m. It is characterized very hot and dry summers up to ~50°C and cold winters down to ~0°C (Fig.2a and Fig.2b).

It is difficult to define the building comfort. Thermal comfort is therefore a function of the building is used and the ambient temperature. The width of 'comfort zone' depends on the balance between both parameters.

Fig.1: Situation of Adrar site [2]

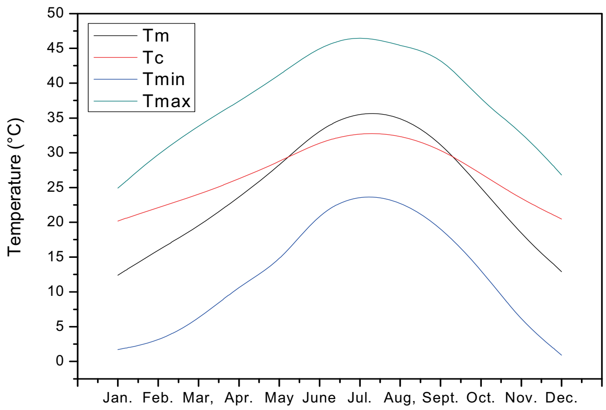

Fig.2 : Monthly temperatures variations

The relationship between comfort temperature and the outdoor temperature is used to design comfortable buildings (Fig.2). Here the indoor comfort temperature (T_c) is calculated from the outdoor temperature mean (T_m) and plotted on a monthly basis together with the monthly mean of the daily maximum (T_{max}), minimum (T_{min}) and mean outdoor temperature (T_m). [3]

Through these results we can see that the Adrar site is characterized by an air temperature ranging from -0.5°C to 47°C. This climate means a long cooling period of 8 months and the 2-3 months heating.

Figure 2 helps the designer to judge whether passive heating and/or cooling is possible. The relationship between the comfort temperature and the range of outdoor temperatures shows whether, for instance, night cooling is likely to be viable in keeping the building comfortable in summer and if passive solar heating will be sufficient in the winter, see Roaf et al, [4]

The heating and cooling demand in northern Africa varies considerably in the different countries. The greatest cooling demand is to the East and in the Sahara desert, while the greatest heating demand is in the North West. In Algeria the heating demand is greater than the cooling demand.

Since the comfort temperature is based on monthly mean air temperature it is not affected by diurnal temperature variations. The indoor temperature variations are fundamental for the occupants comfort since the variations are essential for the energy demand and to maintain comfort.

Based in results presented in Fig.2a and Fig.2b, the heating and cooling DD were calculated, table 1.

Table 1: Monthly degree days variation (DD)

Month	Tc	Ambient average temperatures	DD_h for heating	DD_c for cooling
January	20,2	12.4	-241.8	
February	22,1	16	-172	
March	24	19.4	-142	
April	26,2	23.6	-80	
May	28,7	28.1	-17.8	
June	31,7	33.6		58.7
July	33	36		95
August	33	35.4		86.3
September	30,6	31.6		31.1
October	27	25	-62	
November	23,3	18.2	-153.8	
December	20,5	12.9	-234,5	

Table 1 gives that the Adrar site requires heating during October to May, with the total heating degree days DD_h = 1102. Cooling is needed from June to September based on comfort temperature as the base temperature. The total number of cooling degree days is DD_c = 271.

3. Building

The studied building is part of URER (Renewable Energy Research Unit) in Adrar. A plan drawing of the building is shown in Fig.3. It is seen that the livable surface is of 165m^2 and the livable volume V_h = 448 m^3. The volumetric heat loss coefficient G, which characterizes the building's thermal quality, considers the thickness of the walls, materials used and the number of openings (Table.2), [5]

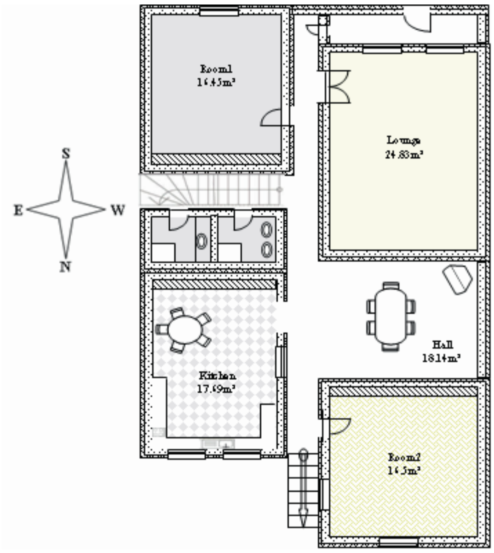

Fig.3: Plan drawing of building

Table 2: Total thermal losses from building

		Walls	Windows	Floor	Roofing	Total losses
Lounge	K	3.5	5.8	4	4	
	S (surface)	62.1	6.8	35	35	
	KS	217.3	39.5	140	140	536.4
Hall	S	36.4	11.2	20	20	
	KS	127.4	65	80	80	352.4
Room 1	S	52	4	25	25	
	KS	182	23.2	100	100	405.2
Room 2	S	50.2	5.9	25	25	
	KS	175.7	34.2	100	100	410
Kitchen	S	52.9	8.7	30	30	
	KS	185.1	50.5	120	120	475.7
Passageway	S	52.4	32	15	15	
	KS	183.4	185.6	60	60	489
Bathroom	S	31	4.2	15	15	
	KS	108.5	24.4	60	60	253
TOTAL						2922

Then the total sum of losses (for $\Delta T=1°C$) of the test building is P = 2922 W/°C, see table 2. The volumetric heat loss coefficient G is given by

$$G = \frac{P}{V_h} = 6.52\ (W / m^3 °C)$$

and the volumetric heat loss is then given by

$$C = G \cdot V_h \cdot 24 \cdot DD / 1000 \quad (\mathrm{kWh})$$

The total energy requirements for heating and cooling of the test building are given below.

- Annual requirements for heating for a comfort temperature, October to May :
  $DD_h = 1102$; where C = 7.7 MWh.

- Annual requirements for cooling for a calculated comfort temperature T_c, June to September:
  $DD_c = 271$; where C = 1.9 MWh.

4. Heating and cooling by using the ground

Depending on weather, time of day and season, outside air undergoes great variations of temperature and humidity. In contrast, the ground temperature at 10-15 meters below the ground surface is constantly at annual mean air temperature. The air-ground exchanger also called Canadian well or wells Provence - takes advantage of this inertia by contacting the outside air heat with the ground. Its primary purpose is the pre-conditioning of heat and humidity of ventilation air in buildings.

Specifically, the air-ground heat exchanger consists of tubes buried a few meters under the ground, near or directly under the building to ventilate. The air from the outside, moved by a ventilator, runs through the tubes before being blown into the building (Fig.4).

During the passage through the ground tubes, the air flow exchanges heat with the ground and can also deposit some water be condensation into the tubes. The ground-air heat exchanger thus reduces thermal variations and air humidity, which corresponds to pre-conditioned air. This system uses very little driving energy, so, it is almost passive.

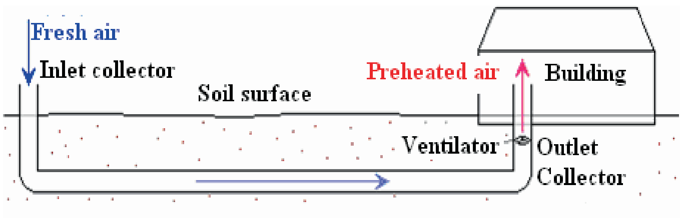

Fig.4: Simplified diagram of a Canadian well-pipe
(Example of operation preheating)

Air-ground exchangers are available on the market. The performance of this type of integrated equipment is benign for sustainable building development.

The Canadian will-pipe system was chosen as the ventilation system of reference for this study. However, several studies address the technology of air-ground heat exchangers, either as a main subject [Mihalakakou et al. 1995[6] et [25]; Benkert et al. 1997[7]; Bojić et al. 1997[8]; Benkert and Heidt 2000[9]; Kunetz and Lefebvre 2001[10]; Hollmuller 2002[11]; Gehlin 2002[12]; De Paepe and Janssens 2003[13]; Dibowski 2003[14], Zweifel 2004[15]; Al Ajmi et al. 2005[16]; Dibowski 2005a[17], 2005b[18]; Ghosal and Tiwari 2006[19]; Badescu 2007[20] and [21], or as a specific study on cooling of buildings.

Such systems are used for cooling of buildings and industrial applications e.g. greenhouses (Gauthier et al. 1997[22]; Hollmuller and Lachal 2001[23], Ghosal and Tiwari op. cit. [19]).

5. Fouggara: An ancient system for underground transport of water

The Fouggara is known under various names; "Kariz" in Afghanistan, "Qanat" in Iran, "Fouggara" in Algeria and "Khettara" in Morocco. The water flow through the Fouggara is gravity driven. At the mouth of the underground channels it reaches the surface where it is used for irrigation and as drinking water, see Fig.5a.

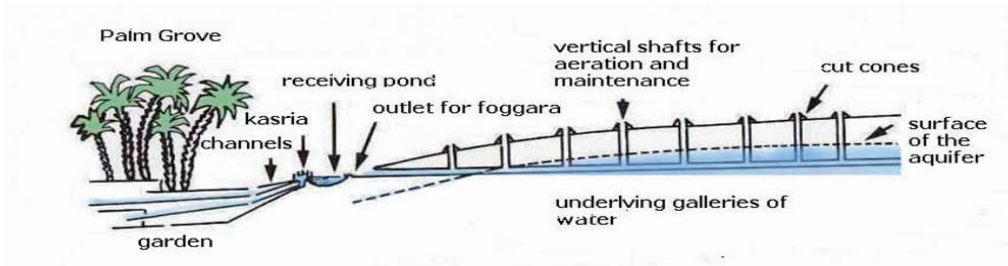

Fig.5a: Outline of the Fouggara principle. [27]

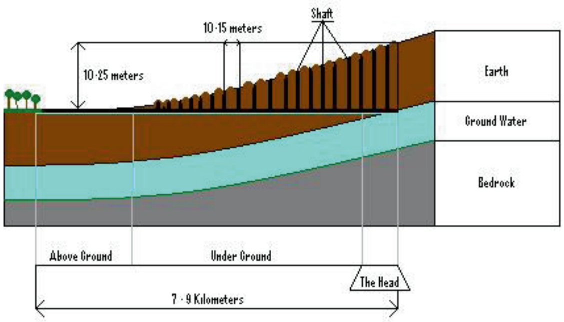

Fig.5b: Vertical section along the Fouggara. [28]

This Fouggara technique is very old and was first used in Iran. It was introduced in Algeria by Arabs at the time of their conquests of Algeria. This technique was adopted by farmers of the area for the following reasons:

- The construction of the Fouggara (Khettara) does not require a financial expenditure, but simply an effort of work.
- Underground transportation of water minimises the water loss due to evaporation and prevents the accumulation of the sediment caused by sandstorms. [26]

5.1. Characteristics

The Fouggara is made up of:

- The source where the water seeps into the channel from a ground water source,
- An underground channel which brings the water to its intended destination,
- An over ground channel which leads to a network of channels feeding the water to particular areas or fields for irrigation.

It is characterised by;

- The length of Fouggara varies from a few hundred meters to tens of kilometres.
- The Fouggara is gravity driven as the water source is situated at a higher elevation then the area to be irrigated.
- The underground channel is wide enough to allow the clearing out and cleaning by hand.
- The depth of the shaft at the source varies from 10 to 25 meters.
- The distance between two consecutive shafts depends in the stability of the terrain and varies between 10 and 15 meters.
- The water flow rate varies from 0.002 to 0.020 m^3/s
- It is located at a depth where the temperature is constantly at annual mean air temperature (i.e. 21°C in Adrar).

In Algeria the Fouggara is extended over 56000 ha in Naama and in Adrar there are 900 Fouggaras. Today, the land area irrigated by Fouggaras is estimated at 8500 ha though 1/3 of them are in operation. (Fig.6). [24]

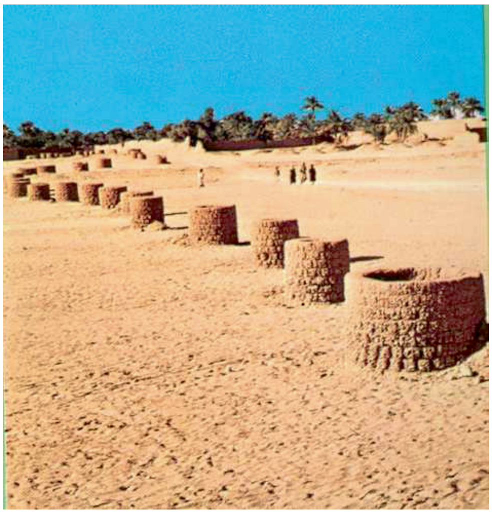

Fig.6: Many wells represent the Fouggara

The existing Fouggaras, active or inactive, means available underground duct systems that could be used as a source for heating and cooling. In the remote Saharan villages of the Saharan people (Fig.7), which lack resources for expensive drilling, available Fouggaras offer an inexpensive alternative. Such a system would be made simply by pumping air through the Fouggara and conducting this air to the air conditioned building. The ambient air then approaches the temperature of the ground (~21°C) during the passage through the ground before used for ventilation, heating, and cooling of buildings (Fig.8).

Fig.7: Fouggara in Adrar.

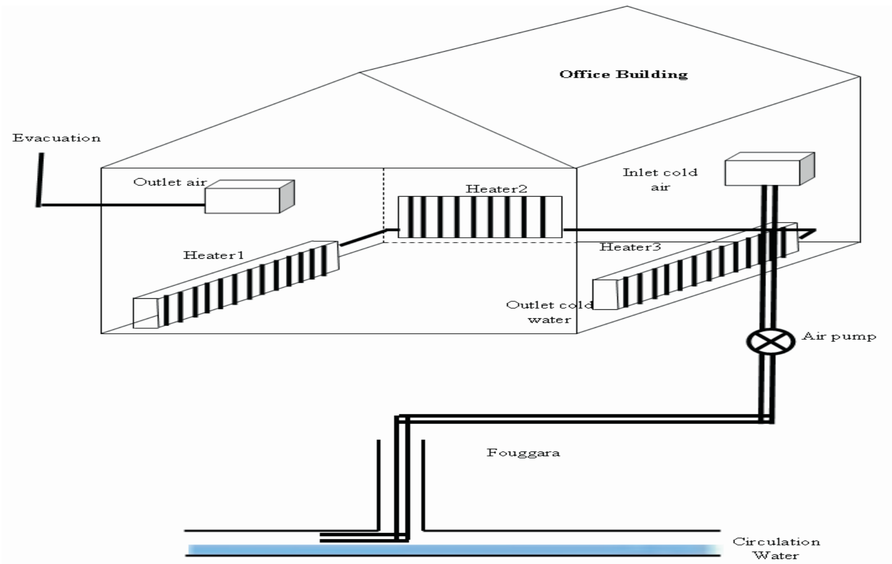

Fig.8 : The ground-air conditioning system. system

The suggested Fouggara air conditioning system, for cooling in the summer and heating in the winter, of the URER test house consists of:

- One room characterized by a volume of 70 m^3
- A pipe system connected to an air convector
- A pump with an air flow rate of 0.02 m^3/h from the Fouggara.

Fig.9 shows the ambient temperature and the air conditioned indoor temperatures during the year, give that the difference between the ambient and indoor temperatures in Adrar is approximately ±20°C.

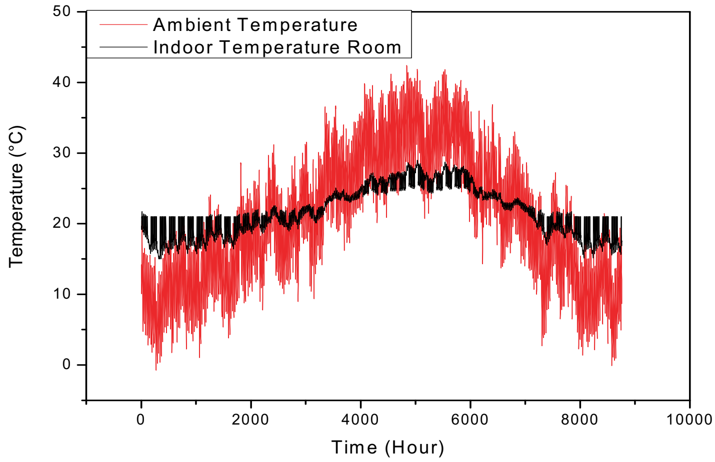

Fig.9: Ambient and Indoor temperature variation

6. Conclusion

The aim was to study the ancient Fouggara system and investigate its possible use as a source for heating, cooling and ventilation of buildings.

The annual energy cooling demand of the test building situated in Adrar gave:

- Heating demand for comfort temperature,
  - October to May: DD_h = 1102; where C = 7.7 MWh.
- Cooling demand for comfort temperature,
  - June to September: DD_c = 271; where C = 1.9 MWh.

The Fouggara consists of:

- The source where the water seeps into the channel from a ground water source,
- An underground channel which brings the water to its intended destination,
- An over ground channel which leads to a network of channels feeding the water to particular areas or fields for irrigation.

Since the Fouggara is at a depth below ground surface where the seasonal temperature changes do not reach, the Fouggara temperature is close to the annual mean air temperature. In Adrar in the south of Algeria the mean annual temperature is 21°C. Pumping air through the Fouggara means that the air temperature will approach this temperature.

- This novel idea of using an ancient construction as a source for heating, cooling and ventilation has not yet been tested.
- It would be attractive to realize such a test on the test building in Adrar.
- However, before performing such test, simulations of how the operation affects the air and ground temperatures should be made.

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