TAMPEREEN TEKNILLINEN YLIOPISTO

ANTONIO GARCÍA HORTELANO

WATER HEATING SYSTEM
-APPLICATION IN AN APARTMENT BUILDING-

Master of Science Thesis

Examiner: Professor Timo Kalema
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ABSTRACT

TAMPERE UNIVERSITY OF TECHNOLOGY
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Energy consumption is one of the great measures of progress and welfare of society . The concept of "energy crisis" occurs when energy sources from which people supply are exhausted. An economic model like the actual, whose operation depends on continued growth, also requires a growing demand for energy. As sources of fossil and nuclear energy are finite, it is inevitable that at some point the application cannot be supplied and the whole system collapses, unless new methods of energy are discovered and developed: these would be alternative or renewable energies.

For all these reasons, large companies and major institutions are devoting efforts in research and development the so-called renewable energy which are naturally replenished. Although some of these energies are still under development, others can be found today in fields such as transport fuels, power generation, heating systems...

One of these energies is the solar energy. Solar energy, radiant light and heat from the sun, has been harnessed by humans since ancient times using a range of ever-evolving technologies. One of the most common uses of solar energy is the solar thermal energy a technology for harnessing solar energy for thermal energy (heat) using solar thermal collectors.

Along this document, the use of solar thermal energy will be studied and analyzed in a specific case; a heating water system in an apartment building in Madrid (Spain). To this end, based on the amount of energy needed to meet consumption, the system will be sized, including the number of collectors, connections, the heat exchangers and all the elements needed.

PREFACE

I wrote this Master of Science thesis in my Erasmus stay in Tampere, Finland. I did it working during ten months in the whole academic year 2010-2011.

I would like to thank my examiner Prof. Timo Kalema for his guidance and feedback.

Finally, I would like to thank my friends, both old and new, and my family for their support over the years.

Antonio García Hortelano
Tampere 14.06.2011

TABLE OF CONTENTS

TERMS AND DEFINITIONS

AMBIENTAL PARAMETERS

Solar radiation: the energy from the sun in the form of electromagnetic waves.

Direct solar radiation: solar radiation incident on a plane, from a small solid angle centered on the solar disk.

Hemispheric solar radiation: solar radiation incident on a flat surface, received from a solid angle of 2.sr (the hemisphere located above the surface). It is necessary to specify the inclination and azimuth of the receiving surface.

Diffuse solar radiation: solar radiation hemispheric less direct solar radiation.

Global solar radiation: hemispheric solar radiation received at horizontal surface.

Solar irradiance (H): radiant power incident per unit area on a given plane [W/ m^2] .

Direct solar irradiance: the quotient of the radiant flux received on a given surface, from a small solid angle centered on the solar disk, and the area of that surface. If the plane is perpendicular to the axis of the solid angle, the solar irradiance received is called directly normal [W/ m^2].

Diffuse solar irradiance: solar irradiance of diffuse radiation on a flat receiving surface. It is necessary to specify the inclination and azimuth of the receiving surface.

Reflected solar irradiance: the radiation per time and area that reaches a surface from the reflection of solar radiation on the ground and other objects.

Irradiation (I): incident energy per unit area on a given plane, obtained by integration of irradiance over a time interval, usually an hour or a day [MJ/m^2 or kWh/m^2].

Ambient air: air (both indoor and outdoor) that involves thermal energy storage, a solar collector or any object.

Peak sun hours (PSH): it is the equivalent number of hours per day when solar irradiance averages 1 kW/m^2. For example, six peak sun hours means that the energy received during total daylight hours equals the energy that would have been received had the irradiance for six hours been 1 kW/m^2.

INSTALLATION PARAMETERS

Open installations: the primary circuit is connected permanently to the atmosphere.

Indoor installations: the primary circuit has no direct communication with the atmosphere.

Direct system installations: the heat transfer fluid is the consumption water that passes through the sensors.

Indirect system installations: the heat transfer fluid is kept in a separate circuit, no communication with the load circuit.

Thermosyphon installations: the heat transfer fluid circulates by free convection.

Forced circulation installation: installation equipped with devices that cause the forced circulation of the working fluid.

Primary circuit: circuit which includes the collectors and pipes that connect them, where fluid collects solar energy and transmit it.

Secondary circuit: circuit which collects the energy transferred from the primary circuit to the consumption water.

Consumption circuit: circuit which water consumption flow by.

Precast solar energy system: a solar energy system whose purposes is preparing hot water. The system consists of either an integrated set or a uniform set of components and configurations. It is normally used when the conditions that uniform and commercially, the system is sold as a single product. Serial connected auxiliary power systems are not considered part of the system.

Compact system: precast solar energy system whose elements are assembled as a single unit; although, physically, they can be differentiated.

Party system: precast solar energy system whose elements (collector and accumulator) are physically separated from each other.

Integrated system: precast solar energy system whose main elements (collector and accumulator) are a single component and it is not possible to distinguish them.

COLLECTORS

Solar thermal collector: A device designed to absorb solar radiation and transmit thermal energy to a fluid that flows through it.

Solar collector with liquid fluid: solar collector that uses liquid as working fluid.

Solar collector with air: solar collector that uses air as working fluid.

Flat solar collector: solar collector absorber without concentration devices which is a flat surface.

Collector without cover: solar collector without cover over the absorber.

Concentration sensor: solar sensor that uses reflectors, lenses or other optical elements to redirect and concentrate the solar radiation in a small surface.

Vacuum sensor: sensor with vacuum in the space between absorber and cover.

Vacuum tube collector: vacuum sensor that uses a transparent tube (usually glass) with vacuum between tube wall and the absorber.

Cover: A transparent (or translucent) element that covers the absorber to reduce heat loses and protects it.

Absorber: a sensor component whose function is to absorb solar radiant energy and transfer it as heat to a fluid.

Absorbing plate: absorber whose surface is a plane surface.

Opening: surface where unconcentrated solar radiation is admitted to the sensor.

Opening area: maximum plane projection of the collector surface that is transparent to unconcentrated incident solar radiation.

Total area: maximum area projected by the sensor, excluding any form of support and the tubes exposed.

Heat transfer fluid or working fluid: the fluid is responsible for collecting and transmitting the energy captured by the absorber.

Frame: is the element that forms the outer surface of the collector; it holds the cover and contains and protects the other components of the collector.

Insulating materials: those materials with low thermal conductivity, whose employment in the solar collector are designed to reduce heat losses through the back and sides.

Cover gasket: is an element whose function is to ensure the tightness of the cover-frame linkage.

Stationary temperature in the collector: It is the maximum fluid temperature when the sensor is exposed to high levels of radiation and temperature and wind speed is null, there is no movement in the collector and reached stationary conditions.

OTHER DEVICES

Heat exchanger: a device which produces energy transfer from the primary circuit to the secondary circuit.

Solar deposit or solar tank: tank which collects the water heated by solar energy.

Expansion tank: a device which can absorb changes in volume and pressure in a closed circuit caused by changes in temperature of the circulating fluid. It can be opened or closed when needed.

Circulation pumps: electromechanical device that produces the forced circulation of fluid through the circuit.

Air purger: a device that allows the exit of air accumulated in the circuit. It can be manual or automatic.

Safety valve: a device that limits the maximum pressure in the circuit.

Non return valve: a device that prevents fluid flow in one direction.

Temperature differential controller: electronic device that commands various electrical components of the system (pumps, valves, etc.) depending mainly on the temperatures in any part of the system.

Safety thermostat: a device used to detect the maximum allowable temperature of the heat transfer fluid in any point of the system.

Frost Controller: a device that prevents the freezing of the working fluid.

1. INTRODUCTION

During the following project, it will be studied and designed a water heating system using solar energy in a flat building in Madrid (Spain).

The goal of the project is threefold; firstly and foremost, it is the technologic goal where the system, all its components and its working principle will be calculated and designed. The second goal is the economic study of the system and its potential profitability. And finally, the third goal is the analysis of the environmental benefits that would occur with the construction of the heating system.

With these objectives is necessary to know and calculate the following parameters:

- Meteorological data (outside temperatures and solar radiation).

- Consumption and domestic hot water required.

- Current heating installation (energy source used, heating and storage systems and heat exchangers).

- Proposal solar installation (solar collectors, solar primary circuit, heat exchangers, secondary circuit, and storage systems).

- Location of the solar installation.

- Energy balance (total energy demand, monthly and annual basis and the calculation of the contributions of solar origin that can be achieved).

- Economic balance (solar installation cost, annual savings and repayment terms).

- Proposed scheme with the solar energy system.

2. THEORICAL BACKGROUND

During last decades, energy consumption has increased significantly. Despite advances in efficiency and sustainability, of all the energy harnessed since the industrial revolution, more than half has been consumed in the last two decades [F2.1]. This fact is primarily the result of global increases in the standard of living and of the increase in world population.

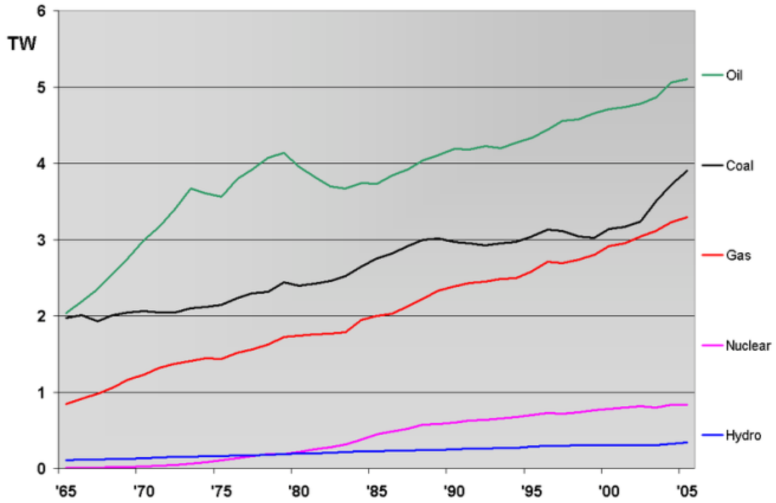


Figure 2.1: Rates of world energy usage in terawatts (TW) 1965-2005

However, the natural resources of the traditional ways of energy (e.g. coal, gas, oil) are finite and some of them are supposed to be extinguished in the current century. In addition, the excessive consumption of conventional energy brings problems such as gradual pollution, global warming and the drilling of the ozone layer. For these reasons, the so-called renewable energies are becoming more important. Some of these energies (first generation technologies), as biomass or geothermal, have been used for more than one century but in these cases, the natural resources are also finite. Some other ways of energy (second generation technologies) are market-ready and are being deployed at the present time; they include solar energy, wind energy or bioenergy. There is also a third generation technology that includes biomass gasification or ocean energy but it still requires a hard research and development effort in order to make it economically sustainable.

As mentioned, one of the so-called second generation technologies is the solar energy. It refers primarily to the use of solar radiation for practical ends. However, all renewable energies, other than geothermal and tidal, derive their energy from the sun.

The Sun is almost perfectly spherical and consists of hot plasma interwoven with magnetic fields. It has a diameter of about 1,392,000 km, about 109 times that of Earth and the mean distance of the Sun from the Earth is approximately 149.6 million kilometers, though the distance varies as the Earth moves from perihelion in January to aphelion in July. **[1]**

The Sun radiant energy is distributed according to a fictitious sphere whose center is the Sun and whose radius grows at the same speed as the radiation itself. Therefore, the intensity at a point of the spherical surface, the solar energy distributed over a growing area, will be much smaller the larger the radius of such sphere.

The Sun behaves almost like a black body which emits energy according to Planck's law at a temperature of 6000 K. Solar radiation ranges from infrared to ultraviolet. The scale that measures the solar radiation that reaches the Earth is the irradiance, which measures the energy per unit time and area [W / m^2]. The estimated value of this intensity at a distance that is our planet from the Sun is 1367 W/m^2 (solar constant). As mentioned, the orbit of the Earth is not circular but elliptical, so the solar constant varies slightly depending on the point of the orbit the Earth is. **[1]**

Not all the Sun radiation reaches the Earth's surface [F2.2], because the shorter ultraviolet wavelengths are absorbed by gases in the atmosphere, mainly by ozone. On the other hand, although the radiation moves in a straight line, the photons undergo diffusion and dispersion when they reach the atmosphere. This diffused radiation also reaches the surface coming from the whole sky due to the mentioned changes of direction by the atmosphere. This radiation is known as diffuse radiation. For the sun energy case, it is necessary to consider the sum of diffuse radiation and direct radiation, forming the total radiation. The diffuse radiation is about one third of the total radiation is received during the year.

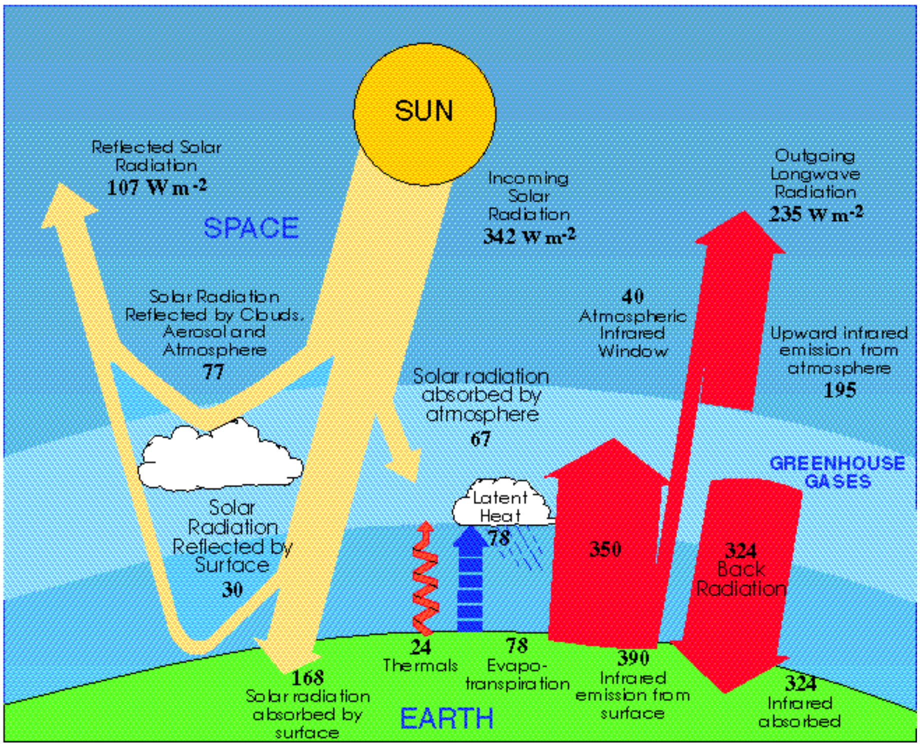


Figure 2.2: Average power diagram

Irradiation, E, is the amount of radiant energy that reaches a given area at a given time. Radiant intensity, I, is the incident energy per unit time and area. The relationship between them, therefore, is I = E /(S · t), where S is the surface where the radiation incides.

On the other hand, to design solar facilities, data on a horizontal surface should be converted in data on an in data on an inclined surface for both the direct component of radiation and the diffuse component.

Assuming that the sun shines uniformly, so the distribution throughout the day is equal, and adopting a standard curve of radiation to the direct irradiance, that relates the direct normal irradiance with solar zenith height, it is possible estimate the relationship between monthly direct radiation on a sloping surface and the corresponding monthly average direct radiation on a horizontal plane **[Appendix A]**.

The total radiation on an inclined surface including the albedo (the fraction of incident energy diffused by a luminous body) is:

$$I_{tot} = I_{0(\theta_n)} + I_{dif} = \eta_D \cdot I_{0(h)} + I_{dif}$$
$$= \eta_D I_{0(h)} + I_{dif(h)} \cdot \frac{1+\cos\theta_n}{2} + (I_0 + I_{dif})_h \cdot \frac{1-\cos\theta_n}{2} \cdot \rho \quad [2.1]$$

- $I_{0(\theta_n)}$ is the radiation on an inclined surface at an angle θ_n.
- I_{dif} is the diffused radiation from the sky to a inclined surface at an angle θ_n.
- η_D is the relationship η_D between the radiation on the inclined surface and the radiation on the horizontal surface $I_{0(h)}$.
- $I_{0(h)}$ is the radiation on an horizontal surface.
- θ_n is the angle that the surface is inclined.
- ρ is the reflectivity.

2.1 Solar Technology

The solar energy goal is to maximize the physical effects of radiation, using the proper capture devices to obtain energy in the form that is required for every need.

Currently, the solar energy is used, among other ones, in the next applications: water heating, electric power generation, cooling, distillation, desalination, solar ovens, evaporating, air conditioning, drying, transporting… All these applications have been developed but not all of them can be used for commercial purposes due to the high cost involved compared with the use of conventional energies.

Among the commercial uses of solar energy, the most important are electric power generation and water heating.

Sun energy can be transformed into electricity, either directly using photovoltaic, or indirectly using concentrated solar power. Photovoltaics (PV) convert solar radiation into direct current electricity using semiconductors that exhibit the photovoltaic effect. Photovoltaic power generation employs solar panels composed of cells that contain a photovoltaic material (monocrystalline silicon, amorphous silicon or cadmium telluride). Concentrated solar power (CSP) systems use lenses or mirrors and tracking systems to focus a large area of sunlight into a small beam. The concentrated heat is then used as a heat source for a conventional power plant.

2.2 Water Heating System Theory

Heating water is the application that best fits the use of solar energy due to the range of temperatures that are needed to achieve (40 - 60 °C) is the range with the greatest efficiency of solar energy collectors. It is also a need that must be satisfied during the twelve months of the year, so that investment in the system is paid off more quickly than in the case of seasonal applications, such as winter heating, or heating swimming pools in summer.

In low geographical latitudes (below 40 degrees) from 60 to 70% of the domestic hot water use with temperatures up to 60 °C can be provided by solar heating system **[6]**.

This technology use solar thermal collectors to collect heat by absorbing sun radiation [F2.3]. Sunlight strikes and heats a surface within a solar collector or a storage tank. Either a heat-transfer fluid or the actual potable water to be used flows through tubes attached to the absorber and picks up the heat from it.

The heated water is stored in a separate preheat tank or a conventional water heater tank until needed [F2.3]. If additional heat is needed, it is provided by electricity or fossil-fuel energy by the conventional water-heating system.

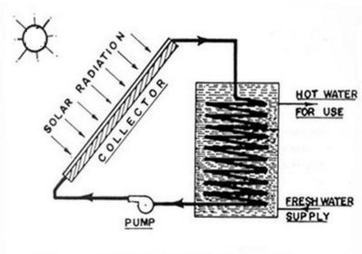


Figure 2.3: Solar water heating

All solar water heating system use the same basic method for capturing and transferring solar energy, but they do so with three specific technologies (low, medium and high temperature collector) using different collectors and systems. The distinctions are important due to different water heating needs in various locations. Different types of collectors and systems, materials and components are used depending on the expected operating temperature range.

Low temperature collectors are used primarily for heating systems and water heating. They are included in a system that can be designed with a wide range of variants but all of them are a combination of the same elements (mainly collector and accumulator), by forming a set or independently.

When developing a regular water heating system, firstly, it is necessary to provide the system a sufficient number of collectors to capture the energy, choosing the suitable location and inclination to exploit the maximum amount of solar energy available each month. It is also necessary to set the capture of that energy to transform it into useful energy. So, the temperature in the collectors and accumulator is continuously measured and compared; and automatic mechanisms in the primary circuit established fluid flow when necessary, depending on whether or not there is an increase energy earnings. To achieve this goal, it is necessary to introduce the concept of regulation difference, which will be discussed later.

Even though most of systems have a support energy system, it is important to consume firstly solar energy, so the storage system should work so as to promote the priority use of solar energy instead of the auxiliary for maximum saving of conventional energy and, therefore, economic. Sometimes this is not compatible with certain system designs in which the system will work improperly, causing a poor performance. So, it is necessary to develop the system choosing the proper devices to achieve the required goals (energetic, economic, pollution…).

2.3 Water Heating System Elements

In regular water heating system can be distinguished the following elements **[8]**:

2.3.1 Capture elements

It is responsible for capturing the incident solar energy and transforms it into heat energy. It consists of collectors, their fixing elements and other accessories.

Before reviewing the design and the different types of collectors, it is necessary to analyze their operation and how solar radiation is used within the collector, specifically in a flat plate collector.

A body exposed to the sun receives an energy flow (E), under whose effect the body gets hot. At the same time, the body loses energy (E_l) due to radiation, convection and conduction with the environment. When heat losses are equal to the incident energy flow, the body reaches the temperature of equilibrium (t^o_e) **[7][22]**:

$$E = E_l \qquad [2.2]$$

If some energy is drawn of the body to use it as useful energy (E_u), then the lost energy decreases and the balance is reached when:

$$E = E_l + E_u \qquad [2.3]$$

In this case, the body has become a solar thermal collector.

There are two options to increase E_u : either increase the incident energy or reduce heat losses. The first option involves improving the design and construction of the collector in order to reduce losses. In the second case the system is modified to concentrate the incident energy on a smaller surface; the smaller the area, the higher intensity. This system is called collectors of concentration.

Another factor to consider is that the greater the difference between the working temperature and environment temperature, the greater also the heat losses and, therefore, lower the useful energy that can be used. So the efficiency decreases when the working temperature increases. As a high efficiency is required, collectors should work at the lowest temperature that allows the system works, at least, at the minimum temperature required.

When some radiation reaches any body, it can be either totally or partially absorbed, reflected or can pass through the body. The radiation that is absorbed by the body causes it to warm up and it emits radiation with a wavelength that depends on the temperature that the body is.

As most solar radiation is between 0.3 and 2.4 μm; it could pass through a transparent glass that is usually an element of a flat plate collector. Although, in a transparent glass, a small part of radiation is reflected and another small part is absorbed, it is consider as opaque body. After passing through the glass, the radiation reaches the surface of the absorber, which is heated and emits radiation as with a wavelength roughly between 4.5 and 7.2 μm.

The radiation emitted by the absorber is reflected in a small percentage of the inner surface of the glass, and the rest will be absorbed by itself, so its temperature rises and it begins to emit radiation that is equally divided to the outside and inside the collector, thus contributing to an increase in the temperature of the absorber surface, this phenomenon is called greenhouse effect [F2.4].

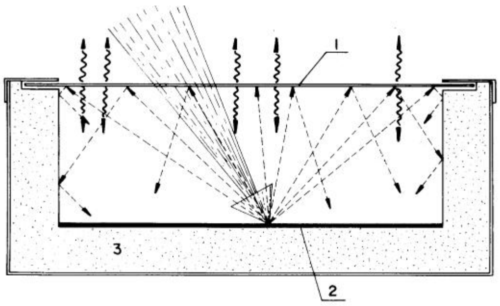


Figure 2.4: Illustration of the greenhouse effect within the collector and its elements

The external plastic cover is not only the cause of the greenhouse effect but reduces heat transfer by convection between the absorber and the external environment; reducing the losses considerably.

If there was not any fluid inside the absorber, its temperature would increase gradually and losses by conduction, convection and radiation would also rise, because they increase when the temperature increase. Finally, as previous mentioned, the absorber would reach the equilibrium temperature.

On the other hand, if a heat transfer fluid circulates through the interior of the collector, entering through a hole and going out through another, the temperature of the fluid would increase due to the stored energy in the absorber. If the fluid is at stationary conditions, it would reach at new equilibrium temperature that is called dynamic equilibrium temperature. This temperature is obviously lower than the static equilibrium one.

The maximum temperature that a collector can reach is the static equilibrium temperature. It is very important when designing the system because it will be the temperature of the system when it is not working, and because the theoretical maximum use temperature is always lower than the temperature static equilibrium.

After analyzing the performance of the collector, it is important to analyze the power balance that occurs in during its operation.

For studying the operation of a collector, it is important to consider it as stationary collector that receives solar radiation uniformly and constantly and a heat transfer fluid that flow through the collector entering a temperature and leaving it at a higher temperature, due to the absorption of some energy when it circulates through the conduits of the absorber. Therefore, the power balance of the collector is:

$$Q_T = Q_U + Q_L \qquad [2.4]$$

- Q_T is the total incident power.
- Q_U is the useful power, the power that is transfer to the heat transfer fluid.
- Q_L is the power lost by dissipation.

Total incident power (Q_T) is the radiation that reaches the surface, but if there is a transparent cover, its transmittance (τ) must be taken into account, because only a part of the radiation pass through it, and also the absorption coefficient (α) of the absorber plate.

$$Q_T = I \cdot S \cdot \tau \cdot \alpha \qquad [2.5]$$

- I is the total incident radiation on the collector per area (W/m^2).
- S is the collector area (m^2).
- τ is the transmittance of the transparent cover.
- α is the observance of the absorber plate.

The calculation of the lost power by dissipation to the outside is more complex because it is produced simultaneously by the conduction, convection, and radiation losses. To simplify this fact, it is used the so-called overall heat transfer coefficient, U, which is measured experimentally and its value is given by the producer. The lost power is also proportional to the difference between the average temperature of the absorber plate and the environment.

$$Q_L = A \cdot U \cdot (t_c^o - t_a^o) \qquad [2.6]$$

- A is the area of the collector (m^2).
- U is the overall loss coefficient ($W/m^2 \cdot °C$).
- t_c^o is the average temperature of the absorber plate (°C).
- t_a^o is the ambiental temperature (°C).

Therefore, the initial power balance equation is as follows:

$$Q_U = A \cdot [I \cdot (\tau \cdot \alpha)_{eff} - U \cdot (t_c^o - t_a^o)] \qquad [2.7]$$

The average temperature of the absorber plate cannot be calculated in a simple way, it has to be measured it directly by a series of sensors placed on it. On the other hand, as mentioned, it can be known with sufficient accuracy the average fluid temperature; a very simple way is to find the average temperature of the fluid at the entrance and exit of the collector.

If the absorber plate and tubes which flows the heat transfer fluid through had an infinite thermal conductivity factor, then the fluid and plate temperatures would be equal, but this really never happens and not all the heat absorbed in the absorbing surface passes the fluid to become useful heat energy. So, it is necessary to take into account a correction factor, called factor of efficiency or heat transfer coefficient (F_R) which is always less than 1. Using this factor the temperature of the fluid absorber plate may be replaced by the temperature of the absorber plate. This factor is practically independent of the intensity of the incident radiation, but is a function of the fluid flow and the collector characteristics (material, thickness, distance between tubes...).

$$Q_U = F_R \cdot A \cdot [I \cdot (\tau \cdot \alpha)_{eff} - U \cdot (t_c^o - t_a^o)] \qquad [2.8]$$

And applying the Bliss equation:

$$Q_U = A \cdot [F_R \cdot I \cdot (\tau \cdot \alpha)_{eff} - U_L \cdot (t_c^o - t_a^o)] \qquad [2.9]$$

The value of the collector efficiency may be calculated using the previous results as:

$$\eta = Q_U / A \cdot I \qquad [2.10]$$

$$\eta = F_R \cdot (\tau \cdot \alpha)_{eff} - U_L \cdot [(t_m^o - t_a^o) / I] \qquad [2.11]$$

In practice, $(\tau \cdot \alpha)_{eff}$ and U_L may be consider as constants and the efficiency may be expressed as a straight line according to $(t_c^o - t_a^o) / I$.

Normally the curve efficiency is given by the manufacturer according to the equation:

$$\eta = b - m \cdot [(t_c^o - t_a^o) / I] \qquad [2.12]$$

Where b and m are two parameters that indicate the efficiency when t_c^o is equal to t_a^o, and the slope of the efficiency curve.

The figure below compares the performance (in terms of $[(t_c^o - t_a^o) / I]$) of different types of collectors [F2.5]:

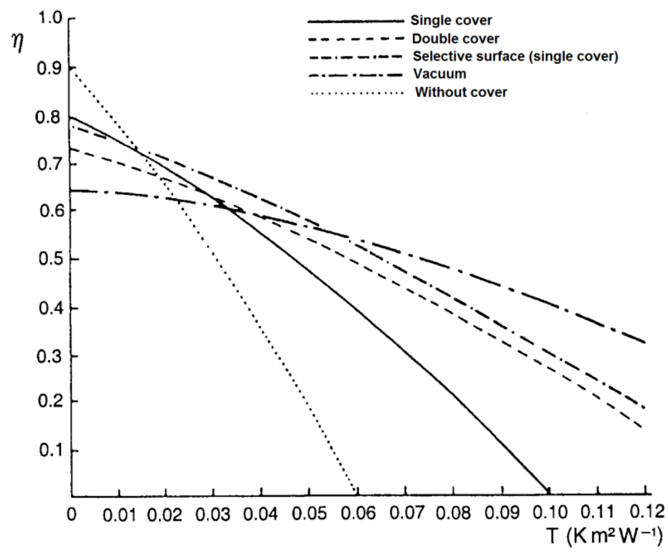


Figure 2.5: Efficiency of different types of collectors

Once exposed the operation of individual collectors, the next step is the analysis of the link among a group of collectors **[6]**.

The serial coupling of the collectors results in an increase in water temperature reducing the efficiency of the system due to when the fluid goes from one collector to another one, the inlet temperature in each one increases and thus the overall efficiency decreases as the efficiency equation shows.

The most common setting is the parallel link among the collectors, or if possible a setting in different rows; in this case each row should have the same number of collectors in a parallel setting and aligned among each other.

When choosing the number of absorbers that can be connected in serial or parallel [F2.6], the limitations of manufacturer should be taken into account. On the other hand, in these systems, it is important to install shutoff valves on the input and output of the groups of collectors and on the pumps to facilitate the maintenance, replacement, etc.

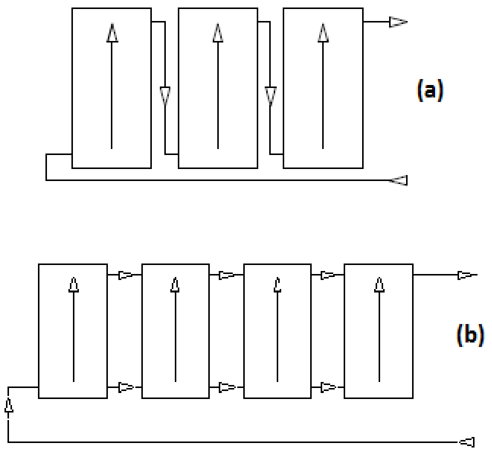


Figure 2.6: Serial (a) and parallel (b) captation

It is also necessary the fluid to have the same way in all the collectors; otherwise, thermal jumps in the collectors would be different and the overall efficiency would decrease. Invest return connection among collectors is used to ensure the energy balance [F2.7].

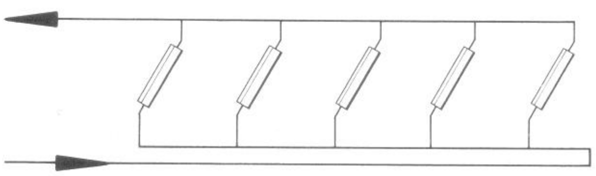


Figure 2.7: Wiring diagram known as inverted return

The heat transfer fluid flow should be higher than 0.8 liters per square meter and minute in order to ensure a right coefficient of heat transfer between the absorber and the fluid. An optimum value would be about 1 liter per square meter and minute.

The fluid track in the system should be as short as possible to decrease hydraulic and heat losses on all the devices. When designing the system, it is also important taken into account that all the devices should be assembled and disassembled to make easy its maintenance.

Finally, we analyze the different types of collectors and which one is the appropriate for a water heating system in the conditions that are required.

The collectors can work with a passive system (the storage tank acts as both storage and solar collector) or with an active system (employ a pump to circulate water or a heat transfer fluid between the collector and the storage tank). Collectors used for water heating are, either flat plate collectors, or the vacuum tube **[8].**

- Flat plate collectors: They are made up of an insulated metal box and a plastic or glass cover (allows solar energy to pass through but reduces heat losses). There is also a dark-colored absorber plate, which absorbs solar radiation [F2.8][F2.10][F2.11].

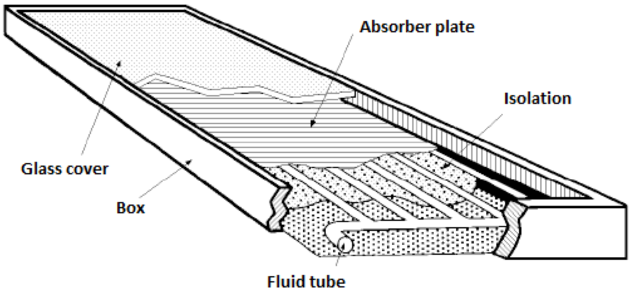


Figure 2.8: Structure of a flat plate solar collector

This radiation is turned into heat energy and a fluid (air, antifreeze or water) that is carried via tubes throughout the collector gets hot [F2.9][F2.11].

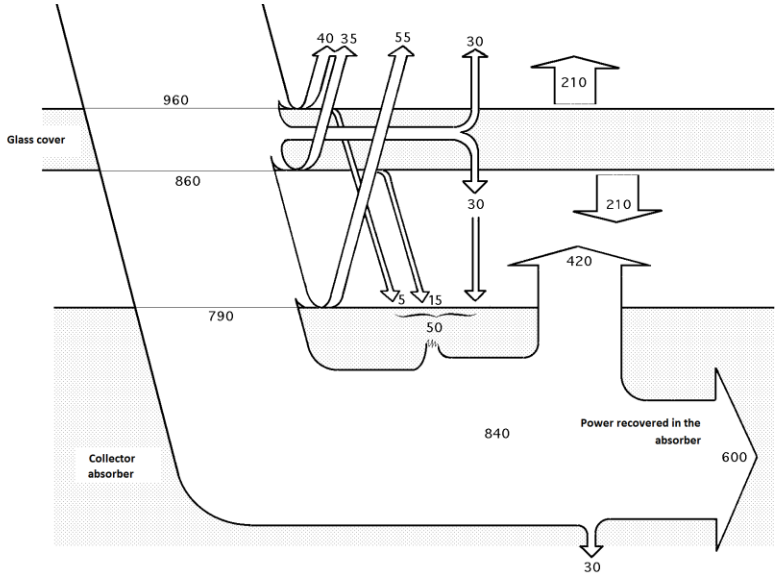


Figure 2.9: Values of the incident energy on a flat plate collector with a cover

There are also some flat plate collectors without the transparent cover. Although they are much cheaper than the ones with cover, their efficiency is much lower.

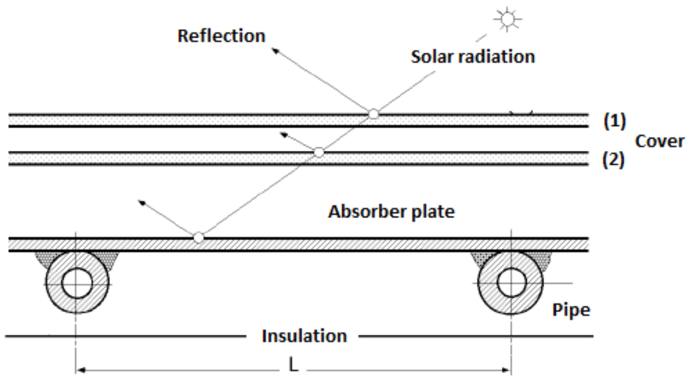


Figure 2.10: Flat plate collector with two covers

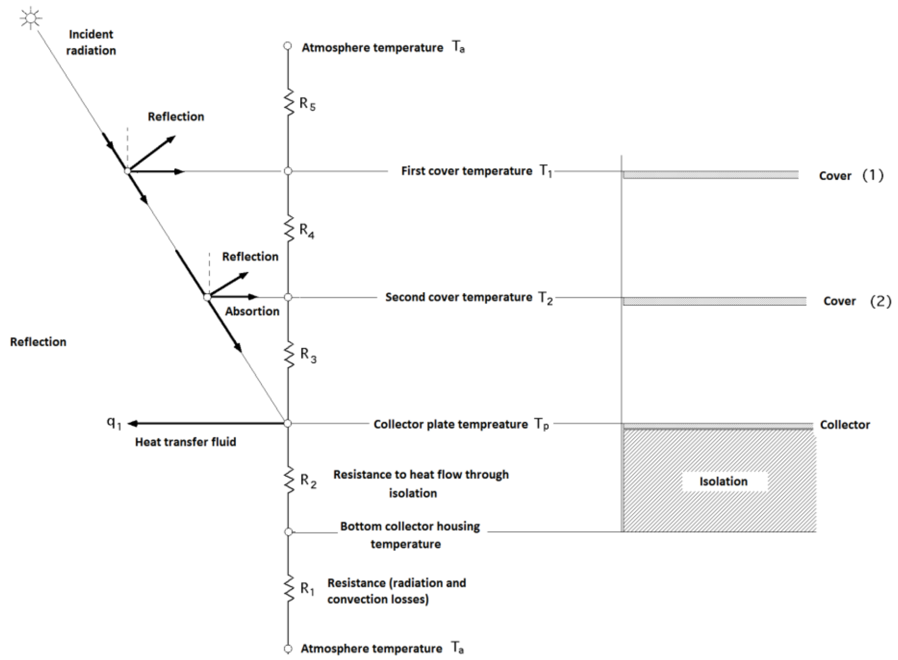


Figure 2.11: Thermal circuit of a flat plate collector with two covers

- Vacuum tube collectors [F2.12]: In this type of vacuum collector, the absorber strip is located in a vacuum and pressure proof glass tube. The heat transfer fluid flows through the absorber directly in a U-tube or in countercurrent in a tube-in-tube system. Several single tubes make up the solar collector. A heat pipe collector incorporates a special fluid which begins to vaporize even at low temperatures. The steam rises in the individual heat pipes and warms up the carrier fluid in the main pipe by means of a heat exchanger. The condensed liquid then flows back into the base of the heat pipe. The pipes must be angled at a specific degree above horizontal so that the process of vaporizing and condensing functions.

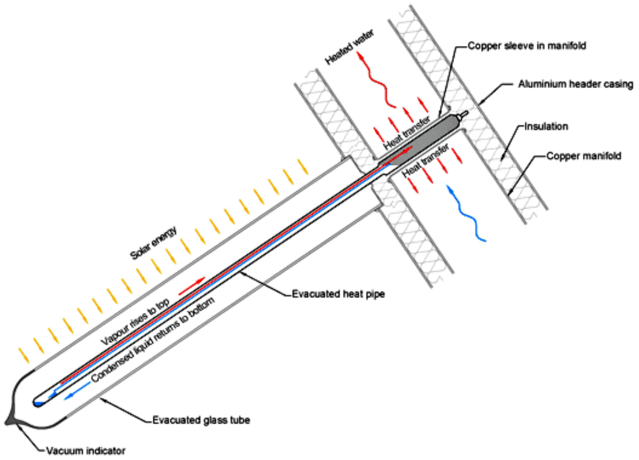


Figure 2.12: Vacuum tube collector

This type of collectors is more used in locations with extreme weather and they can be placed with much more flexibility. However, they are more expensive than flat plate collectors.

The orientation and inclination of the collector system and the shadows on it will be such that the losses about the optimum are lower than the values in the following table. Three different cases can be considered: general, collectors overlay and architectural integration. All cases will accomplish three conditions: loss of orientation and inclination, shading losses and total losses will be lower than the limit values about the optimum [T2.1].

Table 2.1: Maximum loss due to orientation, tilt, shadows and architectural integration

	Orientation and inclination (OI)	Shadows (S)	Total (OI+S)
General	10 %	10 %	15 %
Overlap	20 %	15 %	30 %
Architectural integration	40 %	20 %	50 %

The optimum orientation is south to take advantage of the sun displacement throughout the day and the optimum inclination is equal to the latitude in systems that have the same use throughout the year, to achieve a compromise between the inclination of the collector in winter and summer [F2.13], because it is fixed. However, there are three different cases depending on the period of maximum use **[16]**.

- Constant annual constant consumption: latitude.
- Higher winter consumption: latitude + 10°.
- Higher summer consumption: latitude - 10°.

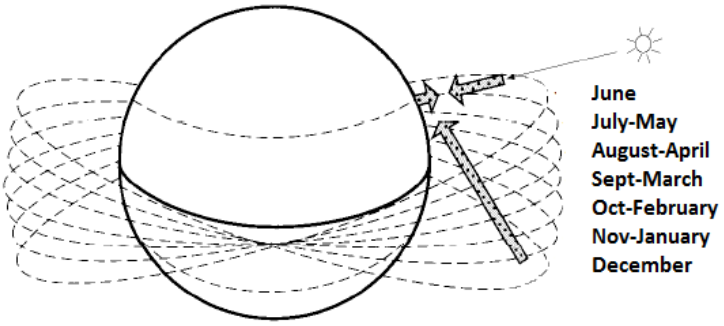


Figure 2.13: Positions of the sun throughout the year

In any case, the reduction of properties due to the inclination and orientation of the collector area should be evaluated. The following image will be used to calculate the limits of acceptable inclination according to the maximum losses respecting the optimum inclination established [F2.14]**[16]**.

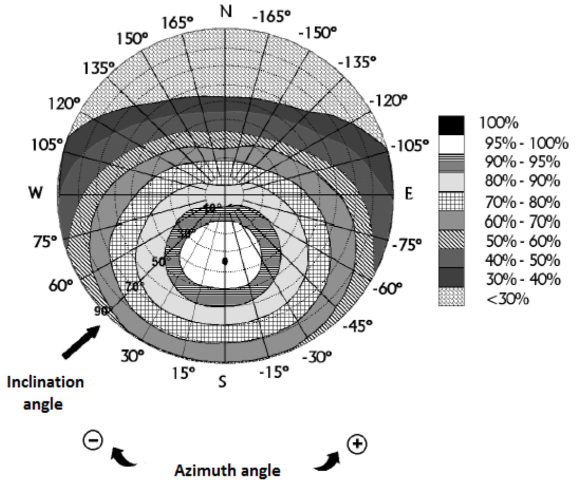


Figure 2.14: Losses depending on inclination and azimut

When the azimuth angle is known, the inclination limit could be known looking where the white region (losses between 0% and 5%) intersects with the azimuth line.

It is considered that there is architectural integration when collectors have a both functions: energetic and architectural and collectors replace conventional architectural elements. It is considered that there is architectural overlap when collectors are parallel to the building envelope. An important rule to achieve architectural integration is to maintain the alignment of collectors with building principal axes.

Finally, the distance, d, measured horizontally, between a row of collectors and an obstacle of height h, which may give a shadow over the facility must ensure a minimum of 4 hours of sun around midday on the winter solstice [F2.15]**[16]**.

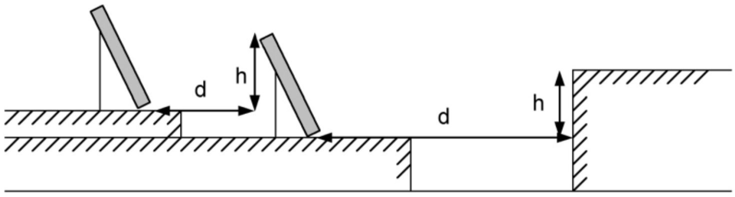


Figure 2.15: Examples of data collection on h and d

This distance d has to be greater than the value obtained by the expression:

$$d = \frac{h}{\tan\,(61° - l)} \qquad [2.13]$$

- l is the latitude.
- $\frac{1}{\tan\,(61° - l)}$ is a dimensionless coefficient called k [T2.2].

Table 2.2: Significant values of k depending on the latitude

Latitude	29°	37°	39°	41°	43°	45°
k	1.600	2.246	2.475	2.747	3.078	3.487

Separation between the back of a row and the beginning of the following shall not be less than the value obtained by the previous expression, when h is the height difference between the top of a row and the bottom of the next, making all measures according to the plane containing the bases of the collectors.

2.3.2 Storage elements

The system must have a storage system to support the water demand when solar radiation is not sufficient. The simplest and most common way of storing hot water is accumulators, which are usually from steel, stainless steel, aluminum or reinforced fiberglass **[8]**.

The shape of the accumulator is usually cylindrical, and its height should be bigger than the diameter, making it so to improve the phenomenon of stratification (the greater the water temperature, the smaller the density). So, the water is extracted at the top of accumulator, while liquid from the collector is supplied to the bottom of the storage [F2.16]. In this way, the collector operates at the lowest possible temperature and the efficiency increases.

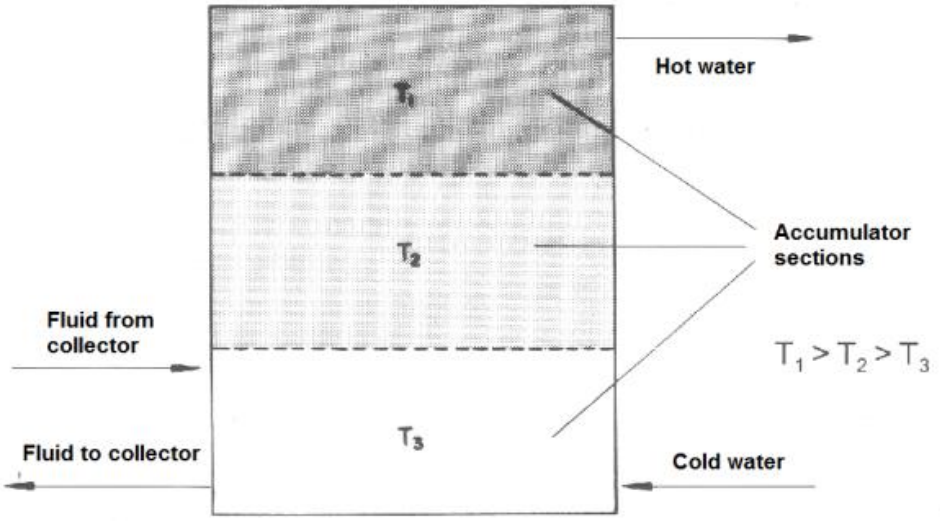


Figure 2.16: Water stratification in an accumulator

A thermostatic mixing valve is normally placed in the accumulator exit in order to limit the temperature that the water is drawn at. This valve does not affect the global efficiency.

2.3.3 Heat transfer elements

They consist of those elements of the system in charge of transferring the captured energy in solar collectors to the storage accumulator of hot water. Among the items that will be used in this group are the heat exchanger, piping, valves…

According to the heat transfer way used in, the system can be divided into two groups, direct and indirect thermal transfer [F2.17]**[20]**.

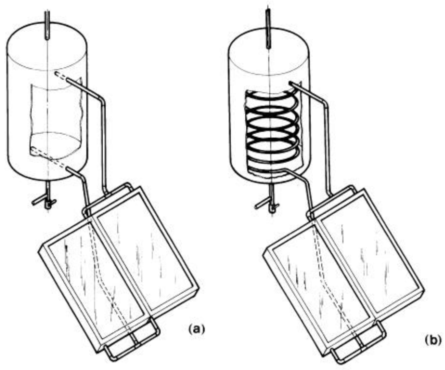


Figure 2.17: Direct (a) and indirect (b) system

In this second group, there is a heat exchanger such that the primary fluid is not in contact with hot water. This way of operation is most common than the direct transfer due to the problems that direct systems have, such as the need to use materials that do not pollute the water in the collector circuit, with the consequent risk of freezing because antifreeze cannot be added to the fluid. Moreover, the risk of vaporization or corrosion increases with this system. This system involves also problems in the collector due to the water pressure and the impossibility of some collectors to operate in such pressure.

On the other hand, another distinction can be made according to the type of flow circulation in the system: natural circulation (also called thermosiphon) or forced circulation using pumps in the primary circuit [F2.18]. While natural circulation is used in simple systems (single family dwellings), forced circulation is used in more complex systems (apartment blocks or flats).

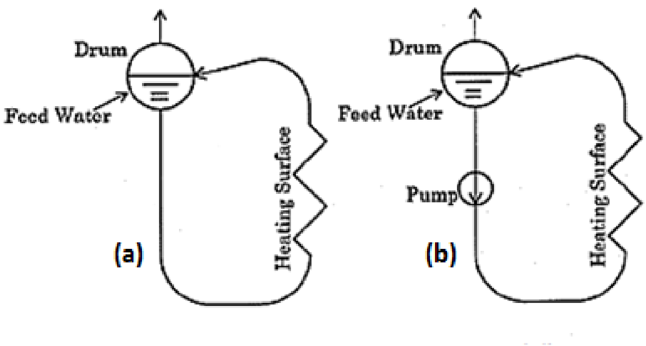


Figure 2.18: Natural and forced circulation

The forced circulation system provides some advantages such as the accumulator has not be placed above the collectors to produce the fluid flow, there are much more possibilities to design the hydraulic circuit (hydraulic losses are corrected with more power in the sizing of the system), the maximum temperature of water in the deposit can be limited (in summer water temperature can reach 60 ° C, with the consequent risk to persons or to the system due to corrosion in the tank) and there are no problems to prevent freezing of the fluid in the collector, which occurs on termosyphon systems; additives to prevent freezing increase the viscosity of the fluid and therefore makes it difficult to circulate.

However, forced circulation systems are more expensive because an extra system to force the circulation; a circuit called exchanger is also required to separate primary from secondary circuit. In the forced circulation systems, there is a loss of efficiency because it is necessary a temperature difference between primary and secondary fluids (from 3 °C to 10 ° C), which makes the collector to be run at a temperature above the fluid secondary.

Finally, the last element in the heat transfer system is an expansion tank, whose function is to absorb the expansion of water, and water hammer by sudden closure of valves.

When choosing an indirect heat transfer system, a heat exchanger is included in the global system to transfer the energy stored from primary circuit to the secondary one.

Depending on its position in the facility, the exchangers can be distinguished in indoors or outdoors exchangers. And depending on its construction, they are classified as coil (helical or tubular beam), double wrap, plate or shell and tube.

The parameters that define a heat exchanger are basically the performance and efficiency of trade. Performance is defined as the ratio of output and input energy. It should not be less than 95%. On the other hand, efficiency is the ratio of heat power exchanged and the maximum that could, theoretically, be exchanged. Its value should not be less than 0.7.

In an internal exchanger, the most used in a heating water system, the efficiency could be expressed as:

$$\varepsilon = (t^o_i - t^o_o) / (t^o_i - t^o_a) \qquad [2.14]$$

- t^o_i is the inlet temperature of the heat transfer fluid.
- t^o_o is the outlet temperature of the heat transfer fluid.

- t^{o}_{a} is the accumulated water temperature.

When the installation needs big accumulators (higher than 3000 l), an external heat exchanger is more useful than an internal one. There are two types of commercial external heat exchangers: tubular beam or steel plates.

A coil heat exchanger can be either helical, if the tubes are coiled and placed in the bottom of the accumulator, or tubular beam [F2.19]. Inside the heat exchanger, the circulation of the fluid is forced circulation; meanwhile, the fluid circulation is due to natural convection outside the heat exchanger.

Figure 2.19: Coil and tubular beam heat exchanger

On the other hand, in the double wrap exchanges, the primary circuit covers the secondary one and energy transfer occurs across the surface in contact with accumulated liquid.

A plate exchanger uses metal plates to transfer the energy [F2.20]. It has a major advantage over a conventional heat exchanger in that the fluids are exposed to a much larger surface area because the fluids spread out over the plates. This facilitates the transfer of heat, and greatly increases the speed of the temperature change. Normally, these systems are not used due to its low efficiency.

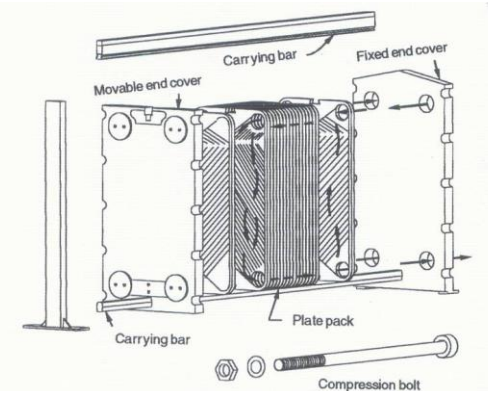


Figure 2.20: Plate exchanger

Shell and tube exchangers consist of a series of tubes [F2.21]. One set of these tubes contains the fluid that must be either heated or cooled. The second fluid runs over the tubes that are being heated or cooled so that it can either provide the heat or absorb the heat required. A set of tubes is called the tube bundle and can be made up of several types of tubes: plain, longitudinally finned, etc. Shell and tube heat exchangers are robust due to their shape, so they are typically used for high-pressure applications (with pressures greater than 30 bar and temperatures greater than 260°C).

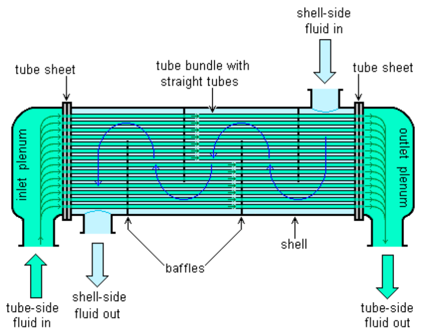


Figure 2.21: Straight-tube heat exchanger

These last two types of exchangers can be either alternative (both fluids pass through the same space alternately but the mixing between fluids is negligible; they are not used for water heating) and by surface (heat transfer takes place through a surface, cylindrical or flat, without allowing direct contact). And, finally, the surface exchanges can be parallel flow exchanger (liquid - liquid) and cross-flow exchanger (liquid - gas), depending on the direction of the fluid circulation and with one or more steps of the fluid through the tubes [F2.22].

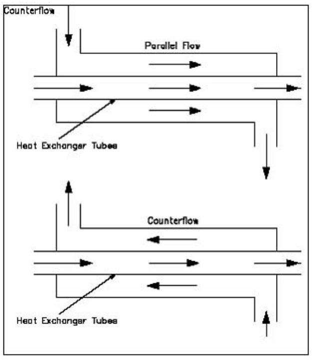


Figure 2.22: Parallel and cross flow exchanger

However, in these exchangers, as in the tubes one, the circulation of both fluids is, normally, forced.

When designing a heat exchanger is important take into account the following factors:

- Material: it is considered its heat transmission coefficient, its corrosion resistance and dilation and it cost when selecting the most proper material for the heat exchanger.
- Tubes or plates geometry: it is necessary calculate the length (it depends on the available space and on the thermo physical characteristics of each fluid), the diameter (it should be large enough if there is risk of contamination but never larger than 50 mm or 2”) and the thickness (it is function of the fluid pressure and the risk of corrosion).
- Tubes distribution: at this point, it is taking into account the cleaning needs and the possibility of increasing the heat transfer. In any case, the space among tubes should be more than a quarter of the external diameter and greater than 6mm (1/4”).
- Fluid used: all the previous factors can be changed depending on the fluid that is going to be used and its heat transmission coefficient, its viscosity and pressure and the appearance of any scale in the tubes.

There are several methods to size a heat exchanger: ΔT_{lm} (log mean temperature difference; it is used when inlet and outlet fluid temperatures are known), NTU (number of transmission units; it is suitable for calculating exchanger design when structural characteristics are known or when just inlet temperatures are known).

For a water heating installation, the second method is more suitable:

- Number transfer units: $NTU = \frac{UA}{\dot{m}c_{p_{min}}}$; where U is the oversized transmission coefficient due to the dirt and pressure drop [T2.3]: $\frac{1}{U} = \frac{1}{U_{clean}} + R$ [2.15]

Table 2.3: Values of Uclean depending on the surface

Water	$U_{clean}(\frac{W}{K\ m^2})$
Soft and clean water	60
Hard water	120
Hard and dirty water	180
Closed circuit	80

- Flow rate (C_R): $C_R = \frac{(\dot{m}C_p)_{min}}{(\dot{m}C_p)_{max}}$; where c_p is the specific heat.
- Exchanger efficiency: it is function of the geometry and the distribution of flows. From this equation it follows an outlet temperature and it is calculated with empirically known tables.

$$\varepsilon = f(NTU, C_R) = \frac{q}{q_{max}} \qquad [2.16]$$

$$q_{max} = c_{min}(T_{h,i} - T_{c,i}) \qquad [2.17]$$

$$c_{min} = min(\dot{m}_h c_{p_h}, \dot{m}_c c_{p_c}) \qquad [2.18]$$

- $T_{\square,i}$: inlet hot fluid temperature.
- $T_{\square,o}$: outlet hot fluid temperature.
- $c_{p_\square}$: hot fluid specific heat.
- $T_{c,i}$: inlet cold fluid temperature.
- $T_{c,o}$: outlet cold fluid temperature.
- c_{p_h}: cold fluid specific heat.

$$\text{If } \dot{m}_c c_{p_c} < \dot{m}_\square c_{p_\square} \text{ then } \varepsilon = \frac{T_{\square,i} - T_{\square,o}}{T_{\square,i} - T_{c,i}} \qquad [2.19]$$

$$\text{If } \dot{m}_{\square} c_{p_{\square}} < \dot{m}_c c_{p_c} \text{ then } \varepsilon = \frac{T_{c,o} - T_{c,i}}{T_{\square,i} - T_{c,i}} \quad [2.36]$$

And one of the outlet temperatures can be known by these formulas. The second one is known by the energetic balance of the heat exchanger.

$$q = \dot{m}_c c_{p_c} (T_{c,o} - T_{c,i}) = \dot{m}_{\square} c_{p_{\square}} (T_{\square,i} - T_{\square,o}) \quad [2.20]$$

However, nowadays there are thousands of possibilities in the market for choosing a proper exchanger to achieve the objectives of the heating system, and it is not necessary to build a specific exchanger.

Along with the exchanger, there is a second essential element in the heat transfer system: the heat transfer fluid. It is responsible for going through the collectors and absorbs thermal energy and then transfers it to the secondary circuit in the heat exchanger. Usually, there are four types of fluids that can be used **[6]**:

- Water: it can be used in open circuit; the hot water goes directly through the collectors, so the system should be made from a material suitable for transporting drinking water. In many cases, it is prohibited by law. It can also be used in closed circuit, but may present problems of freezing, making it necessary the use of antifreeze.
- Water with antifreeze: it is the most common option, but it is necessary take into account some characteristics of the fluid such as toxicity, increasing of viscosity, increasing of expansion, decreasing of stability, decreasing of specific heat or increasing of the boiling point.
- Organic fluids: they require the same precautions than water with addition of antifreeze in toxicity, viscosity or dilation. Furthermore, these fluids are from synthetic materials or petroleum, and they are at risk of fire, but are stable at high temperatures.
- Silicone oils: While they are a good chance due to them best technical characteristics, the high cost do not make them an attractive option in most cases.

Anyway, it must be remembered that due to the toxicity of antifreeze is necessary to ensure the impossibility of mixing between the heat transfer fluid and consumption water. The usual way of achieving this goal is by making the primary circuit pressure lower than the secondary pressure, so if the exchanger breaks in a point and there is a contact between the two fluids, water goes to the primary circuit but not reverse. Moreover the safety valve of the primary circuit must be calibrated to a lower pressure than the consumption water to protect the collectors of the high pressure of the water.

2.3.4 Support energy elements

Not in all cases the temperature of the storage water is the one that is required for the application, so sometimes it is necessary a support system that provides the energy needed to accomplish the required conditions [F2.23]**[9]**.

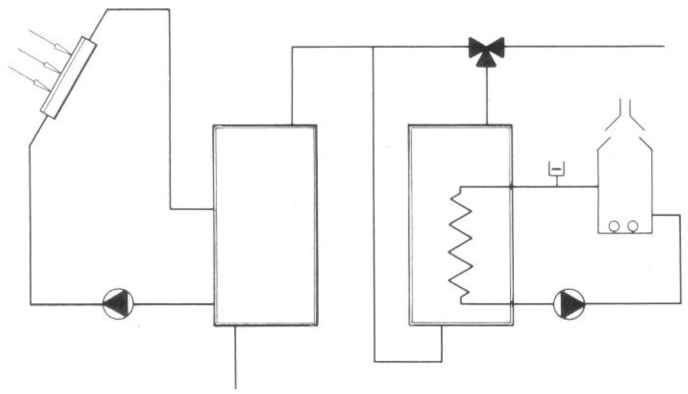


Figure 2.23: Support energy located in a second accumulator fed by the first one

There are different possibilities to apply the support energy to the solar energy system:

- Apply directly the support energy to the accumulator when required.

-Apply the support energy to a second accumulator that is fed by the first accumulator.

-Place a support energy system right after the accumulator.

Sometimes there is an extra accumulator due to the previous installation, so the second option can be applied without an extra expense.

If there is not a previous support energy system, it system has to be designed to be able to heat all the water needed for consumption to the required temperature. The output temperature should not be higher than the consumption temperature. So, although the temperature in the tank must be above 65º to prevent any kind of bacteria (specially legionella), a temperature control is needed in the system output in order to limit the mentioned temperature.

The auxiliary system is normally a fuel or gas system because with these systems is easy to control the outlet temperature, it burns combustible only when needed, the installation and maintenance cost is low, the combustible is cheaper than electricity and finally, it does not affect the solar energy system.

2.3.5 Control elements

The work of these elements is applying useful power when required. The most common way to control the system is a differential controller which compares the temperature of the collector with the temperature in the bottom of the accumulator, and when the temperature in the collector is higher than the temperature in the collector in a fixed amount in the controller, it starts to work [F2.24].

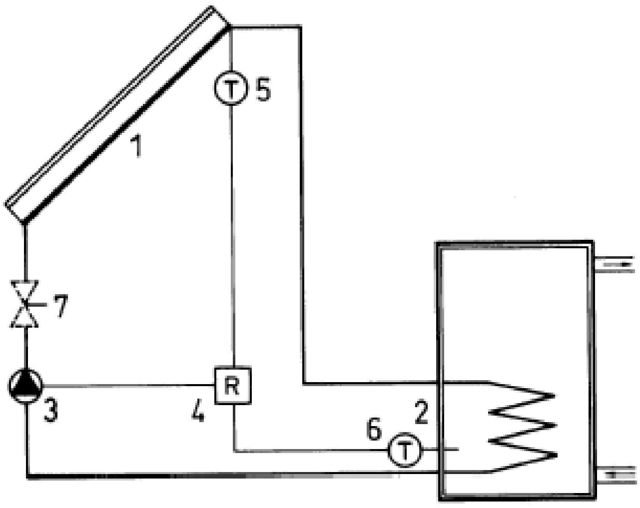


Figure 2.24: Regulation by differential thermostat acting on pump: 1 collector, 2 accumulator, 3 pump, 4 differential controller, 5 and 6 temperature sensor, 7 throttle valve.

It is important keep in mind that the temperature difference should be large enough to ensure a benefit in performance, this is because there are different phenomena that can lead to a malfunction of the installation.

Some of these phenomena could be: the loss of temperature in the feedback loop that may be of around 1 ° C; the tolerances of the probe and the regulator (about 1 or 2 º C); a difference between the heat exchanger temperature and collector temperature for proper operation (around 4 ° C), and the generation of more energy than the one consumed by the electrocirculator (3 ° C). Due to all these phenomena, an usually differential could be 6 ° C.

Anyway, control system should ensure that in any case the pumps work if the differences between the output of collectors and the storage are less than 2 º C and in any case stop with differences of more than 7 º C.

Sometimes, mainly in party systems and depending on each system, the regulation is more effective using a switching valve [F2.25].

In this regulation system, the controller activates the circulating pump when the temperature reaches the minimum usable temperature; at the same time, a switching valve makes a bypass to the primary circuit, leaving the road through accumulators closed. When the temperature exceeds the defined in the controller, the valve is opens the passage of fluid through the heat exchanger.

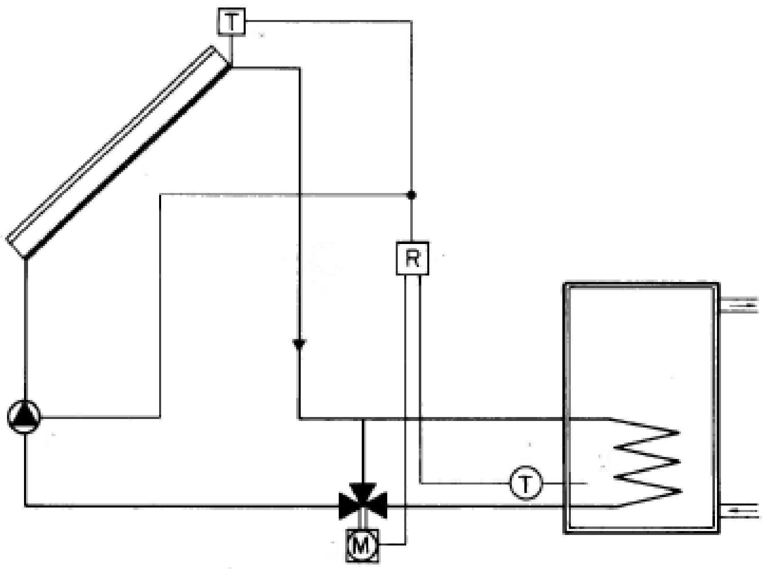


Figure 2.25: Regulation by differential temperature and switching valve

Furthermore, the control system will ensure that at no point the working fluid temperature drops below a temperature three degrees higher than the freezing temperature of the fluid.

The temperature probes for differential control are placed in the top of the collectors; representing its maximum temperature and accumulation temperature probe is preferably placed at the bottom in a space that is not influenced by circulation of the secondary circuit or the heating exchanger if it were built.

2.4 Legislation

According to the Spanish legislation, the following decrees and laws affect the installations and equipment of Solar Power [T2.4].

Table 2.4: Solar energy law in Spain

LAW	TITLE	DATE	APPLICATION
Real Decreto 1027/2007	Real Decreto 1027/2007, by approving the Reglamento de Instalaciones Térmicas en los Edificios (Regulation of Thermal Installations in Buildings).	20-07-2007	Solar thermal energy
Orden ITC/71/2007	Orden ITC/71/2007, by adopting the standards and technical instructions for the homologation of solar panels.	22-01-2007	Solar thermal energy
Real Decreto 47/2007	Real Decreto 47/2007, by adopting the basic procedure for energy performance of new buildings.	19-01-2007	Building
Real Decreto 314/2006	Real Decreto 314/2006, by approving the Código Técnico de la Edificación (Technical Building Code [CTE]).	17-03-2006	Building
Real Decreto 865/2003	Real Decreto 865/2003, by setting laying down the hygienic criteria for the prevention and control of legionellosis	04-07-2003	Solar thermal energy
Ordenanza Municipal ANM 2003\3	Ordenanza Municipal ANM 2003\3, specifications of national laws at regional	27-03-2003	Solar thermal energy

LAW	TITLE	DATE	APPLICATION
Real Decreto 1218/2002	Real Decreto 1218/2002, by approving the Reglamento de Instalaciones Térmicas en los Edificios (Rules Thermal Installations in Buildings [RITE]) and created the advisory commission for thermal installations in buildings.	22-11-2002	Building
Real Decreto 769/1999	Real Decreto 769/1999, by dictating the rules implementing the directive of the European Parliament and Council 97/23/EC about pressure equipment.	07-05-1999	Solar thermal energy
Real decreto legislativo 1175/1990	Real Decreto Legislativo 1175/1990, by approving the fees and instruction of the Tax on Economic Activities	28-09-1990	Several applications

Most of these laws and specifications are based in other European laws or recommendations as Directive 2009/28/EC of the European Parliament (on the promotion of the use of energy from renewable sources).

These laws specify and report on most aspects needed when developing a solar energy system as the technical documentation design and dimensioning of the thermal plants, the conditions for the use and maintenance of the facility (use of thermal plants, preventive maintenance program...), preparation of hot water for sanitary purposes, solar energy contribution to sanitary hot water production, limiting the use of conventional energy for heat production and water heating, water treatment or building integration of the water heating elements.

When designing and developing a water heating system, it is necessary to take into account all these specifications to adapt the system to the legislation.

The following regulations can be mentioned among all the previous ones:

- When the supply source of energy source is diesel, propane or natural gas, the minimum contribution of solar energy depending on the total water demand and the climatic region where the system is in will be the value reflected in the table below [T2.5] **[16][17]**:

Table 2.5: Minimum share (%) from the hot water need to be achieved using solar energy

Total water demand ($\frac{l}{d}$)	I	II	III	IV	V
50-5000	30	30	50	60	70
5000-6000	30	30	55	65	70
6000-7000	30	35	61	70	70
7000-8000	35	45	63	70	70
8000-9000	30	52	65	70	70
9000-10000	30	55	70	70	70
10000-12500	30	65	70	70	70
12500-15000	30	70	70	70	70
15000-17500	35	70	70	70	70
17500-20000	45	70	70	70	70
20000-25000	52	70	70	70	70
>25000	52	70	70	70	70

And the climatic region where the intallation is in can be found in the following map [F2.26]:

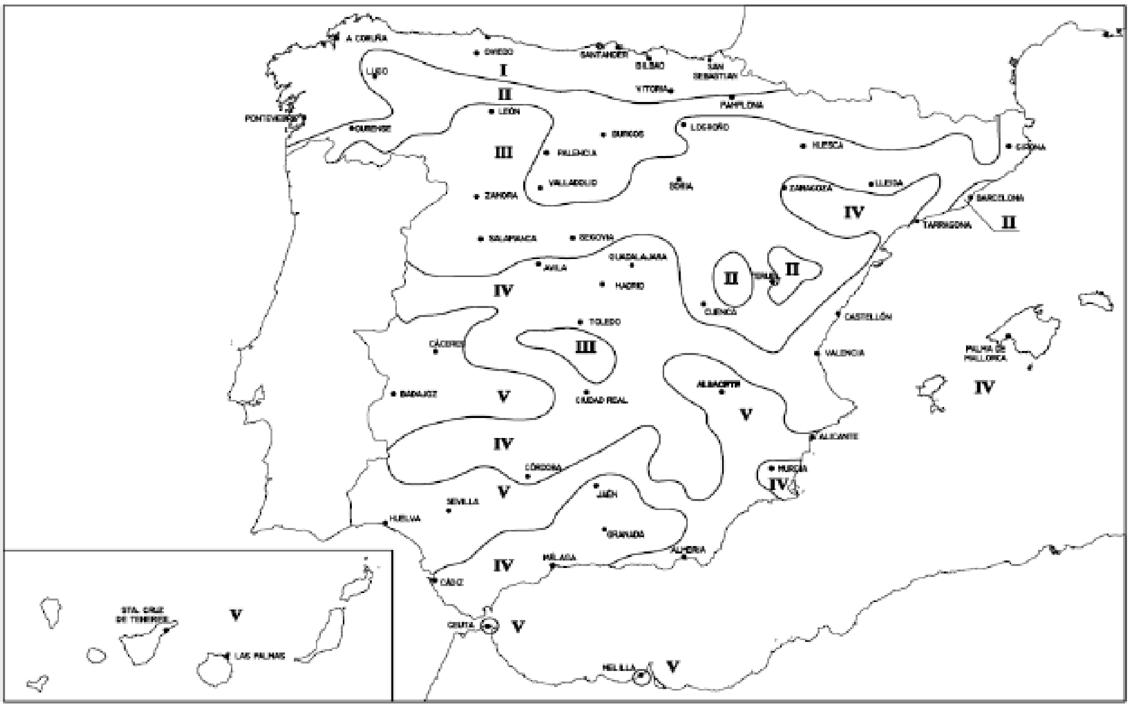


Figure 2.26: Climatic region

In some cases, this minimum contribution should not be applied (when the building does not have sufficient access to sunlight due to external elements, when this minimum value exceeds the criteria that make the law...).

- Solar energy can be replaced, totally or partially, by other kind of renewable energy, by cogeneration processes or by waste energy sources from heat exchangers outside the own generation of heat from the building. These replacements should have an environmental impact equivalent to the one achieved by solar energy.
- Regulations to prevent the disfigurement the landscape or the architectural harmony and regulations to protect buildings, environments and landscapes will be applied when installing a solar energy system.

These regulations also set the recommended values (water per person...) for designing the facilities depending on the use of the buildings (hotel, office, private residence...), their size and people who use them.

3. CASE STUDY

3.1 Location Information

The place where the system will be installed in the flat building located in 36, Santiago de Compostela street in Madrid, Spain.

Madrid is in the center of the Iberian Peninsula [F3.1], which is situated in the extreme southwest of Europe. It is situated at 40°23′N 3°43′W and it is 667 meters over the sea level.

The building is located in the north of the city in a suburb. The block has 13 floors and 2 apartments with 3 rooms per floor [F3.2]. On the basement, there is a parking with two floors. Moreover, on the ground floor there are two bars and a carpets shop. Behind the building there is a little garden.

The building is surrounded by tall buildings that throw shadows on the garden so the system cannot be installed there and should be installed on the roof of the building.

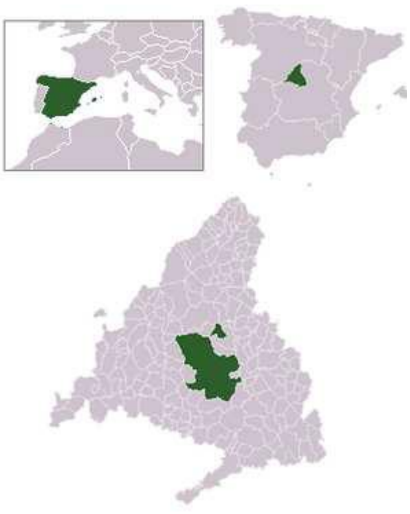


Figure 3.1: Location of Madrid

Figure 3.2: Building where the system will be in

The roof is a rectangle of 10x28 meters with a construction on the middle of the north part with a rectangular shape of 4x6x0.5 meters **[Appendix D]**.

3.2 Wheather Information

The weather in the area means that the annual energy on a horizontal surface is 1668.05 kWh/m^2; so, at first, the profitability compared with other places in Spain is not so greater, but it is good enough to reach the goals mentioned.

According to CENSOLAR (Solar Energy Training Centre), the incident energy (MJ) on one square meter of horizontal surface on an average day of each month is [T3.1]**[21]**.

On the other hand, before calculating the energy needed to heat the water, it is also necessary to know the temperature of the supply water [T3.1], the ambient temperature [T3.1] and hours of sun per day [T3.1]**[21]**.

Table 3.1 : Daily incident solar energy (MJ) on a horizontal surface, supply water temperature, ambient temperature and hours of sun per day.

	H (MJ/m^2)	T_{supply}	Ambient temperature	Sun hours
January	6.7	6	6	8
February	10.6	7	8	9
March	13.6	9	11	9
April	18.8	11	13	9,5
May	20.9	12	18	9,5
June	23.5	13	23	9,5
July	26	14	28	9,5
August	23.1	13	26	9,5
September	16.9	12	21	9,5
October	11.4	11	15	9
November	7.5	9	11	9
December	5.9	6	7	8
Yearly	15.4			

3.3 Consumption Information

For doing the calculations of the hot water consumed, data will be taken according to Real Decreto 1218/2002 (Table 4, Legislation): the amount of 60 ºC water consumed in a flat building is 22 l per person and day and the number of tenants in a 3 rooms apartment is 4. Thus, the hot water consumed per day will be:

$Total\ water\ (l) =$
$Water\ per\ peron\ (l)\ x\ Tenants\ per\ apartment\ x\ Number\ of\ aprtments =$
$22\ x\ 4\ x\ 26 = 2288\ l = 2.288\ m^3$ [3.1]

However, there is a monthly change in the water consumption due to the changing weather and partial occupation in the different seasons of the year. This variation is according to the next graphic [F3.3]**[16]**.

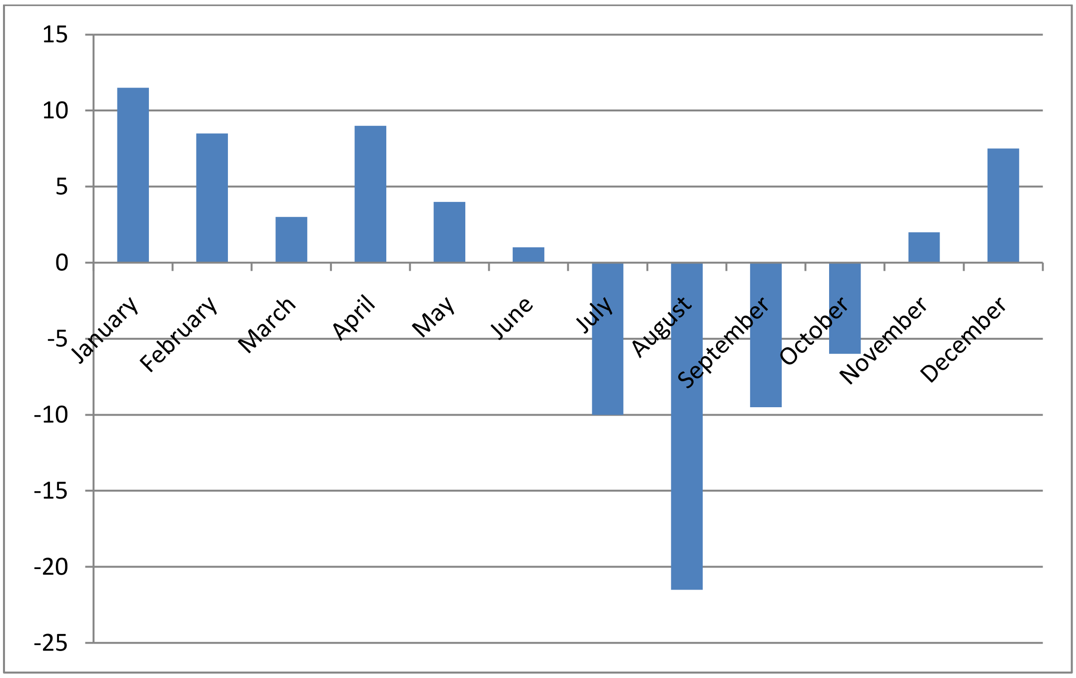


Figure 3.3: Average change in water consumption (%)

As Madrid is located in the climatic region IV (Figure 2.33) and the support energy system will be natural gas, the minimum solar contribution according to the water demand will be 60 % (Table 2.5). However, to make the most of the solar facility, when designing it, the minimum solar contribution will be as close as possible to 70% of minimum solar contribution.

Taking into account the previous factors, the following results are obtained for the building [T3.2].

Table 3.2: Installation consumption

	Days/month	Monthly consumption (l/month)	Consumption varitation (%)	Fixed monthly consumption (l/month)	Sun hours	$\dot{m}_{cons}$ (l/h)	$\dot{m}_{cons}$ (kg/h)
January	31	70928	0.115	79084.72	8	318.89	318.89
February	28	64064	0.085	69509.44	9	275.83	275.83
March	31	70928	0.03	73055.84	9	261.85	261.85
April	30	68640	0.09	74817.6	9.5	262.52	262.52
May	31	70928	0.04	73765.12	9.5	250.48	250.48
June	30	68640	0.01	69326.4	9.5	243.25	243.25
July	31	70928	-0.1	63835.2	9.5	216.76	216.76
August	31	70928	-0.215	55678.48	9.5	189.06	189.06
September	30	68640	-0.095	62119.2	9.5	217.96	217.96
October	31	70928	-0.06	66672.32	9	238.97	238.97
November	30	68640	-0.02	67267.2	9	249.14	249.14
December	31	70928	0.075	76247.6	8	307.45	307.45
Yearly	365	835120	-0.00375	831379.1	9.083	252.68	252.68

As the work fluid is water the density will be $\rho = 1000\ \frac{kg}{m^3} = 1\ \frac{kg}{l}$.

And the yearly average consumption will be:

$$\dot{m}_{cons} = 252.68 \frac{kg}{h} = 0.07 \frac{kg}{s} \quad [3.2]$$

3.4 Energy Information

During the winter, the hot water demand is higher and the incident energy is lower; thus, the inclination of the collector will be 50° to improve the system during the winter months. For calculating the incident energy on an inclined surface, it will be used the correction factor K represents the ratio between the total incident solar energy on a day on an inclined surface at a specific angle, and the total incident solar energy on a day on a horizontal surface [T3.3]**[16]**.

Table 3.3: Factor k

Jan	Feb	March	April	May	June	July	Aug	Sep	Oct	Nov	Dec
1.41	1.28	1.13	0.98	0.87	0.83	0.87	0.99	1.18	1.39	1.54	1.52

In some cases, the radiation on horizontal surface is also modified by another factor that depends on the location of the installation. This factor is:

- 0.95 if the facility is within a town.
- 1.05 if atmosphere is clean or mountain area.
- more severe correction coefficients in the case of providing important shadows in the winter.

This second coefficient is not going to take into account when designing the solar energy system.

The incident energy on an average day on an inclined surface E (50°) is [T3.4]:

Table 3.4: Daily radiation on a inclinated flat collector (50)

	Jan	Feb	March	April	May	June	July	Aug	Sep	Oct	Nov	Dec
H (MJ/m^2)	6.7	10.6	13.6	18.8	20.9	23.5	26	23.1	16.9	11.4	7.5	5.9
k	1.41	1.28	1.13	0.98	0.87	0.83	0.87	0.99	1.18	1.39	1.54	1.52
E (MJ/m^2)	9.45	13.57	15.37	18.42	18.18	19.51	22.62	22.87	19.94	15.85	11.5	8.97

- E is the daily incident energy on a inclined surface ($E = H \cdot k$)[3.3].

And the annual value will be $E = 17.96\ \frac{MJ}{\mathrm{m2}}$.

On the other hand, the irradiation will be [T3.6]:

Table 3.5: Solar radiation

	H (MJ/m^2)	k	E(MJ/m^2)	Sun hours	Irradiation
January	6.7	1.41	9.45	8	328.02
February	10.6	1.28	13.57	9	418.77
March	13.6	1.13	15.37	9	474.32
April	18.8	0.98	18.42	9.5	538.71
May	20.9	0.87	18.18	9.5	531.67
June	23.5	0.83	19.51	9.5	570.32
July	26	0.87	22.62	9.5	661.4
August	23.1	0.99	22.87	9.5	668.68
September	16.9	1.18	19.94	9.5	583.1
October	11.4	1.39	15.85	9	489.07
November	7.5	1.54	11.55	9	356.48
December	5.9	1.52	8.97	8	311.39
Yearly	15.4	1.16	17.96	9.083	549.36

- I is the irradiation ($I = \frac{E[\frac{J}{m^2}]}{sun\ hours[s]}\ [\frac{J}{m^2 \cdot s}]$) [3.4], that is the incident energy per time.

  And finally, the total monthly energy required to get the hot water necessary will be [T3.7]:

Table 3.6: Required energy

	Days/month	Fixed monthly consumption (l/month)	$C_p\ (\frac{kJ}{K\cdot kg})$	T_{use}	T_{supply}	Q(MJ)
January	31	79084.7	4.188	60	6	17885.17
February	28	69509.4	4.188	60	7	15428.59
March	31	73055.8	4.188	60	9	15603.85
April	30	74817.6	4.188	60	11	15353.47
May	31	73765.1	4.188	60	12	14828.56
June	30	69326.4	4.188	60	13	13645.93
July	31	63835.2	4.188	60	14	12297.72
August	31	55678.5	4.188	60	13	10959.53
September	30	62119.2	4.188	60	12	12487.45
October	31	66672.3	4.188	60	11	13681.96
November	30	67267.2	4.188	60	9	14367.47
December	31	76247.6	4.188	60	6	17243.55
Yearly	365	831379			10.25	173783.2

$$Q = m \cdot C_p \cdot (T_{use} - T_{supply}) \qquad [3.5]$$

- Q is the heat energy necessary (MJ).
- m is the water consumed (kg).
- C_p is the specific heat of water ($4.188\ \frac{kJ}{K\cdot kg}$).
- T_{use} is the hot consumption water temperature (60°).
- T_{supply} is the supply network water temperature.

So, the total energy required is:

$$Q = 173783.2 \text{ MJ} \qquad [3.6]$$

$$Q_T^{\prime} = \dot{m}_{cons} \cdot C_{pcons} \cdot \left(T_{use} - T_{supply}\right) = 14.62\ KW \quad [3.7]$$

Therefore, the minimal heat power to be supplied by solar system will be:

$$Q_T^{\prime} = 0.7 \cdot 14.62\ KW = 10.234\ KW \quad [3.8]$$

4. STUDY OF OBTAINABLE SOLAR ENERGY

As previous mentioned, the system will be a forced circulation system with an indirect heat exchange system. So, it consists of three circuits, primary (solar), secondary (supply), for which two fluids circulate for, and the third circuit (consumption circuit) [F4.1] **[9][5]**.

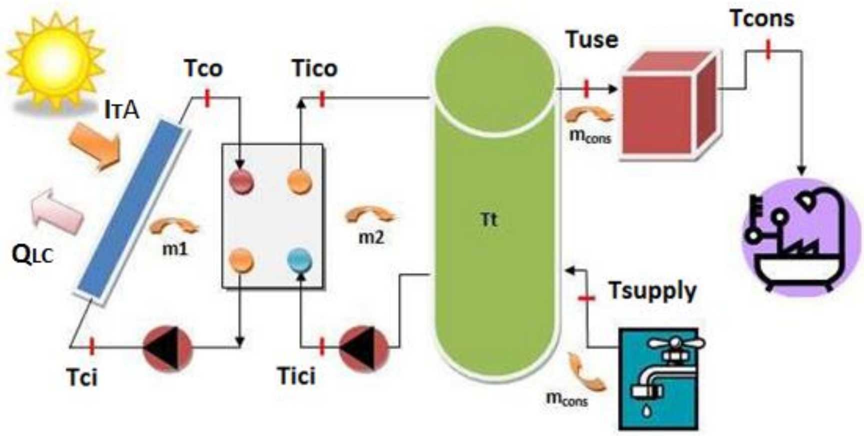


Figure 4.1: Solar energy diagram

4.1 Solar Circuit

The primary circuit will be composed by the solar collectors, the pipes, valves, the expansion tank, the circulation pump and the heat exchanger [F4.2]; it will be as the circuit shown in the following image:

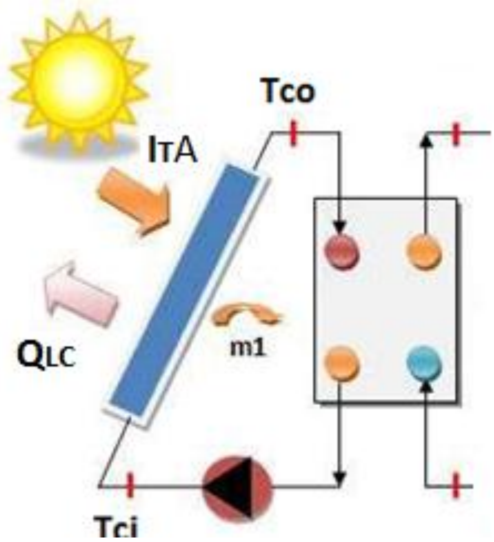


Figure 4.2: Solar circuit

- Q_I is the incident energy in the solar collector.
- Q_{LC} are the optic losses in the collector.
- $\dot{m}_1$ is s the flow of fluid flowing through the installation.
- T_{co} is the outgoing collector temperature.
- T_{ci} is the ingoing collector temperature.

The solar or primary circuit includes the collection system and part of the heat exchanger. The heat transfer fluid flows through the circuit and is responsible for transferring the thermal energy obtained from solar radiation from the primary circuit to the secondary circuit through the heat exchanger.

The design selected for the primary circuit is based on the idea of forcing the circulation of fluid by an electric pump, which although is expensive to install, allows a frost protection of heat transfer fluid when using low freezing point fluid and control heat transfer between collector and accumulation.

In addition to the electric pump in the primary circuit, an expansion tank is placed for absorbing volume expansion in the fluid due to temperature changes.

4.1.1 Heat Exchange Fluid

This fluid carries the heat from the primary circuit to the secondary; it should be able to work during cold temperatures season at which water would freeze, thus it should also protect the system against frost. There are a lot of fluids that can be used for this purpose but for this system, it is going to be used propylene glycol. For knowing the percentage of propylene glycol in water, it is necessary to relate it with the freezing temperature.

The historical minimum temperature history in Madrid is -16 °C; so, to ensure the proper work of the system in any condition, the freezing temperature will be taken as 5°C less than the historical minimum temperature **[16]**.

$$T_{in} = T_{min} - 5°\text{C} = -21°\text{C} \qquad [4.1]$$

When determining the necessary properties of the fluid, it is necessary to use some equations that relate the concentration of propylene glycol with various parameters such as freezing temperature, density, specific heat, thermal conductivity, dynamic viscosity or the number of Prandlt. For the calculation there is a table that provides the constants that give the mathematical models of fluid properties [T4.1]**[2][30]**:

Table 4.1: Constants to calculate the fluid properties

Order parameter (A)	$\rho\ (\frac{kg}{m^3})$	$C_p\ (\frac{kJ}{kg\cdot K})$	$K\ (\frac{W}{m\cdot K})$	Pr	$T_{freezing}$
0	-	-	-	-	1
1	508.411	4.476	1.188	6.661	-0.037
2	-182.408	0.608	-1.491	-6.6994	-0.400
3	965.765	0.714	-0.696	-18.551	-
4	280.291	-1.938	1.136	152.046	-
5	-472.225	0.478	0.067	14.477	-

- ρ is the fluid density.
- C_p is the specific heat.
- Pr is the Prandtl number.
- k is the thermal conductivity.
- $T_{freezing}$ is the freezing temperature.

The concentration of propylene glycol is calculated through the following equation, when $T_{freezing} = -21\ °C = 252\ K$:

$$T_{freezing} = (A_{T0} + A_{T1} \cdot \xi + A_{T2} \cdot \xi^2) \cdot 273.15 \Rightarrow \ \xi = 0.395 \qquad [4.2]$$

- ξ is the concentration of propylene glycol necessary.

Thermo physical properties for the solution can be obtained depending on the working temperature and weight concentration. These properties are dependent on the working temperature (it will be taken an average of about 30 °C = 303 K); it will not be the actual working temperature because the primary circuit may reach 80 °C, but there are no tables with values for these ranges of temperatures, it is noteworthy that the variation of thermo physical properties with temperature are quite small so do not generate large errors in calculation.

When the concentration of propylene glycol is known ($\xi = 0.395$), the other parameters can be calculated as following [T4.2]:

<u>Density</u>

$$\rho = A_{\rho 1} + A_{\rho 2} \cdot \xi + A_{\rho 3} \cdot \frac{273}{303} + A_{\rho 4} \cdot \xi \cdot \frac{273}{303} + A_{\rho 5} \cdot \left(\frac{273}{303}\right)^2 \quad [4.3]$$

$$\rho = 1023\ \frac{kg}{m^3} \quad [4.4]$$

<u>Specific heat</u>

$$C_p = A_{C_p 1} + A_{C_p 2} \cdot \xi + A_{C_p 3} \cdot \frac{273}{303} + A_{C_p 4} \cdot \xi \cdot \frac{273}{303} + A_{C_p 5} \cdot \left(\frac{273}{303}\right)^2 \quad [4.5]$$

$$C_p = 3.771\ \frac{kJ}{kg \cdot K} \quad [4.6]$$

<u>Thermal conductivity</u>

$$k = A_{k1} + A_{k2} \cdot \xi + A_{k3} \cdot \frac{273}{303} + A_{k4} \cdot \xi \cdot \frac{273}{303} + A_{k5} \cdot \left(\frac{273}{303}\right)^2 \quad [4.7]$$

$$k = 0.431\ \frac{W}{m \cdot K} \quad [4.8]$$

<u>Prandtl number</u>

$$\ln(Pr) = A_{Pr1} + A_{Pr2} \cdot \xi + A_{Pr3} \cdot \frac{273}{303} + A_{Pr4} \cdot \xi \cdot \frac{273}{303} + A_{Pr5} \cdot \left(\frac{273}{303}\right)^2 \quad [4.9]$$

$$Pr = 25.127 \quad [4.10]$$

To calculate the dynamic and kinematic viscosity, there are conventional definitions, based on the data obtained previously.

<u>Dinamic viscosity</u>

$$\mu = \frac{Pr \cdot k}{C_p} = 2.869 \cdot 10^{-3}\ \frac{kg}{m \cdot s} \quad [4.11]$$

<u>Kinematic viscosity</u>

$$\upsilon = \frac{\mu}{\rho} = 2.805 \cdot 10^{-6}\,\frac{m^3}{s} \quad [4.12]$$

Table 4.2: Summary data

Heat exchanger fluid: Propylene Glycol	
Density	$1023\,\frac{kg}{m^3}$
Specific heat	$3.771\,\frac{kJ}{kg \cdot K}$
Thermal conductivity	$0.431\,\frac{W}{m \cdot K}$
Prandtl number	25.127
Dynamic viscosity	$2.869 \cdot 10^{-3}\,\frac{kg}{m \cdot s}$
Kinematic viscosity	$2.805 \cdot 10^{-6}\,\frac{m^3}{s}$

4.1.2 Solar Collector

When designing a solar energy system, the first thing that have to be taken into account is the solar collector; after that election, and when the collector characteristics are known, the others parts of the system will be calculated and the storage elements, the supply energy system... will be chosen depending on the first election.

For a water heating system in a flat building, the most proper collector is a flat plate collector due to it is possible to aim all the goals with an acceptable cost.

As previously mentioned, a flat plate collector consists of four main elements: the cover, the absorber, the insulation and the shell [F4.3]**[5]**.

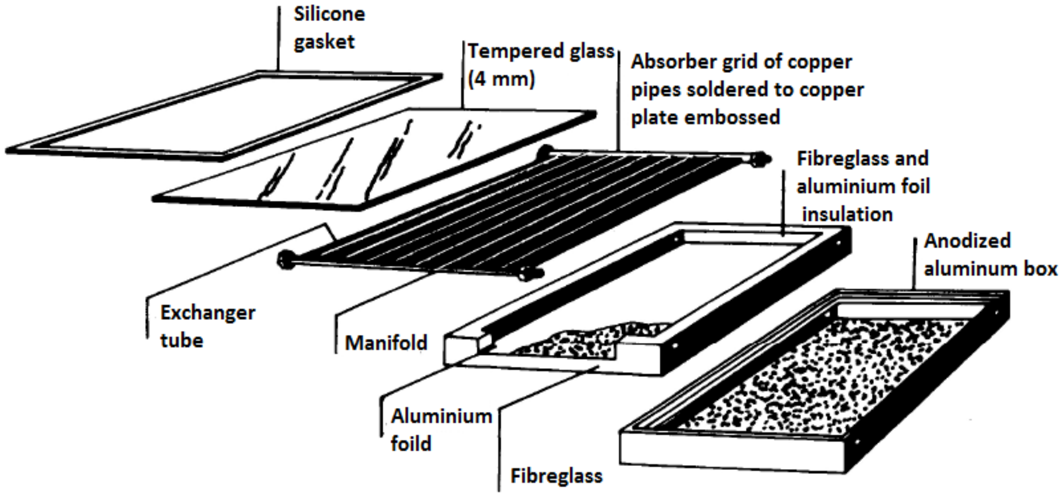


Figure 4.3: Collector parts

A proper collector should have a high rate of transmission for high solar radiation in the range from 0.3 to 3 mm, and low for more than 3 mm radiation. It should also have a low thermal conductivity, which prevents the passage of heat from the interior surface to the exterior one. Thus, the collector should have a small expansion coefficient, the inside of the cover will be warmer than outside and, therefore, the inside part will expand more increasing the risk of breakage or deformation of the cover. The main materials used in the covers are glass and transparent plastics.

When choosing a glass cover, it should be chosen with annealing or milding treatment because not only its optical properties do not decrease but also the mechanical properties increase. That is important because the cover must resist the wind pressure, the hail, the weight of ice and snow, etc and it must also have a low risk of spontaneous rupture due to the effect of internal contractions resulting from the different temperatures cover.

For this water heating system, the proper cover is a annealed transparent glass cover, not only due to the advantages of glass comparing to plastic (better thermal conductivity, low coefficient of expansion, more hardness and chemical stability under the action of agents external) but also due to the breaking and bending strength. Moreover, in case of accidental breakage it fragments into small pieces.

Although it is possible to use a double glass cover, which increases the greenhouse effect and reduce losses by convection. However, it is not usually used due to its high price and the problems resulting from high temperature to be borne by the lower cover, and the differential expansion between the two covers for supporting these different temperatures.

The absorber receives solar radiation, transforms it into heat and transmits it to the heat transfer fluid. It usually has two metal plates separated a few millimeters. Between the two plates, there are welded tubes through the heat transfer fluid. There are also plastic covers, but most of them are designed for pool heating.

The part of absorber exposed to the sun is usually covered with a coating to absorb sunlight. This coating is usually done by painting or selective surfaces. The effectiveness of the coating is given by its emissivity and absorptance values. Selective surfaces have an absorption coefficient similar as the paintings (0.8 or 0.9), but the emission coefficient is considerably less, (0.1 for the selective surfaces and 0.8 or 0.9 for paints). They also have better global performance and durability, but its cost is higher.

When choosing the absorber, it is important to take into account the following characteristics:

- The pressure drop in systems by thermosyphon.
- The internal corrosion: in order of avoid the corrosion, it is necessary to not couple copper and iron materials in the circuit. Furthermore, although initially the heat transfer fluid is not corrosive, it can degrade due to temperature and becomes corrosive.
- The thermal inertia: in areas where there is an strong alternating climate, a stong thermal inertia of the absorber would not allow the fluid temperature range to be achieved in periods of continuous radiation.
- The uniformity of heat transfer fluid circulation: If there is not a proper fluid flow, the heat provided to these areas will be poorly distributed, the temperature will rise abnormally and heat losses will be greater.
- The heat transmission from the absorbing plate to the heat transfer fluid: it depends on the conductivity and thickness of the metal from the absorbing plate is

made, the separation between the tubes, their diameter, the fluid regime, and the welding between the plate and tubes.

- The inlet and outlet pressure drop in the absorber.
- Thermal bridges between the absorber and the non insulated collector.
- The pressure resistance, either by direct connection between the absorber and the network or due to obstruction of the primary circuit in a forced circulation system.

The insulation protects the bottom of the absorber from the heat losses. A proper insulation should have the following characteristics:

- Good behavior to high temperature, in some cases a reflective metal foil reflective is placed between the absorber and the insulation to prevent the absorber to receive direct radiation.
- Low vapor release as a result of a high warming.
- Durability.
- Homogeneity of their properties with moisture conditions.

The aim of the shell is to protect and support various elements of the collector and join the collector and the support structure. The shell should have the following features:

- High stiffness.
- Strength of fasteners.
- Resistance to temperature variations.
- Resistance to corrosion and chemical instability.
- Ventilation of the interior of the collector.
- Retention of water, ice and snow outside the collector
- Easy removal of the cover or the top of the shell to access the absorber.

There are a lot of collectors in the market with all the previous characteristics and similar prizes, but for starting the calculations the collector choosing will be SOL 2800 selective from Salvador Escoda S.A. [T4.3][F4.4]**[23][24]**:

Table 4.3: Collector characteristics

Solar collector	
Manufacturer	Salvador Escoda S.A.
Model	SOL 2800 selective
Type	Plate collector
Area	2.78 m^2
Absorber	Copper (0.12 mm)
Number of absorber tubes	10 + 2 collector tubes
Price	595.00 €

Figure 4.4: SOL 2800 selective collector

The distance between two rows of collectors, taking into account its inclination will be **[16]**:

$$d = \frac{h}{\tan\,(61° - l)} = 4.6\ m \quad [4.13]$$

- $h = L \cdot \sin\alpha = 2.307 \cdot sin\ 50° = 1.767\ m.$
- α is the inclination of the collector ($\alpha = 50°$).
- L is the length of the collector ($L = 2.307\ m$).
- l is the latitude (l=40°).

Thus, due to the size of the roof (10x28m) and the distance between collectors calculated previously, it is possible to install just two rows of collectors. The width of the chosen collector is 1.206 m and it is necessary to leave a two meters free space in the each row to facilitate the installation and maintenance. Then, it is possible to install 21 collectors in each row:

$$Number\ of\ collectors\ per\ row = \frac{Roof\ width - free\ space}{Collector\ width} = \frac{28.06 - 2}{1.206}$$
$$= 21.55\ collectors \qquad [4.14]$$

And the number maximum of collectors in the installation will be 42; however, when calculating the exact number of collectors needed the energy required, irradiation, lost in sewers, pipes, etc. and heat exchanger efficiency will be taken into account.

The best way to connect the collectors is the parallel connection; the flow through each collector is the result of dividing the total flow by the number of collectors. The main reason is the large diameter pipe inside the panel, which allows the connection of multiple panels.

The collector has ten parallel absorber tubes and two cross tubes that work as collectors of the parallel tubes. Each absorber tube has a length of 2255 mm and the collector tubes have a width of 1200 mm.

4.1.3 Calculation of required surface and solar fraction

4.1.3.1 Calculation of the losses due to the efficiency

The first estimation for the collector surface will be done using the average annual radiation and temperature without taking into account the losses due to the wind. Later, this first result of collectors and temperatures at different points of the system may be used to initialize successive iterations of calculation. When the collection area is calculated, it is possible calculated the solar fraction.

For this first approximation, the average daily incident solar radiation on a 50° inclined surface will be:

$$E = 17.96\ \frac{MJ}{m^2} = 4.988\ \frac{kW \cdot h}{m^2} \qquad [4.15]$$

And the daily irradiation is:

$$I_T = \frac{I}{t} = 0.5482\ \frac{kW}{m^2} \quad [4.16]$$

- h is the average hours of sun per day.

The inclination and orientation losses are in the range 0-5 %. On the other hand, there is no element that shades on the installation; thus there are no losses due to the shadows. So, the losses are within the established limit.

The design of the collector area, in a first approximation, has been done using a system of characteristics equations of the circuit **[9]**.

The problem to be solved contains seven unknowns:

- T_{ci} is the ingoing collector temperature.
- T_{co} is the outgoing collector temperature.
- T_{ici} is the ingoing heat exchanger temperature.
- T_{ico} is the outgoing heat exchanger temperature.
- T_t is the average tank temperature.
- T_{use} is the outgoing tank temperature.
- A is the collector area.

The values that are known because have been previously calculated or because they are properties of the elements, are:

- I_T is the radiation on the inclined plane and it depends on the latitude and inclination. This parameter was calculated previously ($I_T = 0.5482\ \frac{kW}{m^2}$).
- ε_{HC} is the heat exchanger (ε_{HC} = 0.7).
- m_{cons} is the HSW consumption ($m_{cons} = 0.07\ kg/s$).
- T_{supply} is the average supply water temperature ($T_{supply} = 10.3\ °C$).
- T_{amb} is the average ambient temperature ($T_{amb} = 15.58\ °C$).
- C_{p1} is the specific heat of the heat exchange fluid ($C_{p1} = 3.771\ \frac{kJ}{kg{\cdot}K}$).
- C_{p2} is the specific heat of the water (C_{p2} =4.188 $\frac{kJ}{kg{\cdot}K}$).
- C_{pcons} is the specific heat of the consumption water ($C_{pcons} = C_{p2}$ =4.188 $\frac{kJ}{kg{\cdot}K}$).
- ρ_1 is the fluid density ($\rho_1 = 1023\ \frac{kg}{m^3}$).

- $ρ_2$ is the water density ($ρ_2 = 1000 \frac{kg}{m^3}$).
- T_{sum} is the consumption temperature of heat sanitary water ($T_{sum} = 60\ °C$).
- f is the solar fraction ($f = 0.7$).
- A_{est} is the first estimation of the collector area ($A = 0.5 \cdot number\ persons = 0.5 \cdot 104 = 52\ m^2$).

And, finally, the variable values are the flow through the primary circuit ($\dot{m}_1$) and the secondary circuit ($\dot{m}_2$) because they can be regulated by the circulation pump.

The variation of the flow will influence the amount of energy transferred to the secondary circuit. The manufacturer recommends a flow on the collectors around 76 l/h·m^2. The heat exchanger will be a balanced exchange; so, $m_1 = m_2$.

$$m_1 = m_2 = m_{man} \cdot A_{est} \cdot ρ_1 = 0.4433 \frac{kg}{s} \quad [4.17]$$

ANNUAL CALCULATION

The seven equations to solve the seven unknown values are the following ones:

1.Energy balance in solar collectors

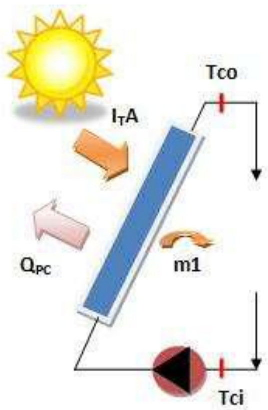


Figure 4.5: Energy balance in the collector

According to the previous figure [F4.5], the stored effect (stored energy per unit time) is:

$$\frac{dE_c}{dt} = I_T \cdot A - \dot{Q}_{TL} - m_1 \cdot C_{p1} \cdot (T_{co} - T_{ci}) = 0 \quad [4.18]$$

- The system is stationary; thus $\frac{dE_c}{dt} = 0$.
- I_T is the radiation on the inclined plane and it depends on the latitude and inclination. This parameter was calculated previously.
- A is the collector area.
- a_o is the optical coefficient of the collector, which corresponds to 0.704 for the chosen collector.
- a_1 is the coefficient of conduction and convection losses.
- Q_{TL} is defined as the heat that comes to the collector but they are not able to grasp. It can be obtained from the efficiency curve.

$$\eta_c = a_0 - a_1 \frac{(T_{ci} - T_{amb})}{I_T \cdot \mathrm{A}} \qquad [4.19]$$

$$\eta_c = \frac{Q_C}{I_T \cdot \mathrm{A}} = \frac{m_1 \cdot C_{p1} \cdot (T_{co} - T_{ci})}{I_T \cdot \mathrm{A}} = 1 - \frac{Q_{TL}}{I_T \cdot \mathrm{A}} \qquad [4.20]$$

$$Q_{TL} = I_T \cdot \mathrm{A} \cdot \left(1 - a_0 + a_1 \frac{(T_{ci} - T_{amb})}{I_T}\right) \qquad [4.21]$$

- $m_1 \cdot C_{p1} \cdot (T_{co} - T_{ci})$ is the useful heat.

And finally, the energy balance in the collector is:

$$m_1 \cdot C_{p1} \cdot (T_{co} - T_{ci}) = I_T \cdot \mathrm{A} \cdot \left(1 - a_0 + a_1 \frac{(T_{ci} - T_{amb})}{I_T}\right) \qquad [4.22]$$

2.Energy balance in the heat exchanger

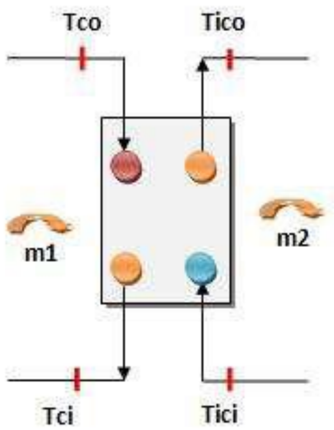


Figure 4.6: Energy balance in the heat exchanger

$$\dot{m}_1 \cdot C_{p1} \cdot (T_{co} - T_{ci}) = \dot{m}_2 \cdot C_{p2} \cdot (T_{ico} - T_{ici}) \quad [4.23]$$

As previously mentioned, the heat exchanger [F4.6] is symmetric or balanced; so, $\dot{m}_1 = \dot{m}_2$.

3.Heat exchanger efficiency

Efficiency is the ability to transmit energy exchanger over the maximum possible. Typical values of efficiency plate heat exchangers range between 70% and 80%. For this initial estimate $\varepsilon_{HC} = 0.7$ will be taken as the value of exchanger efficiency.

$$\varepsilon_{HC} = \frac{\dot{m}_2 \cdot C_{p2} \cdot (T_{ico} - T_{ici})}{C_{min} \cdot (T_{co} - T_{ici})} \quad [4.24]$$

- C_{min} is the minimum value between $\dot{m}_1 \cdot C_{p1}$ and $\dot{m}_2 \cdot C_{p2}$.

4.Minimum solar contribution

The fourth equation will be the relationship between the difference outgoing temperature of the tank (T_{use}) and the consumption temperature (T_{cons}), with the supply water temperature (T_{supply}) [F4.7].

This relation is called the minimum solar contribution or solar fraction and it is defined as the percentage of energy supplied by solar collectors on the total energy required to heat the entire HSW consumed in the building.

As previously mentioned, the minimum solar fraction should be 70%, which represents what temperature can be achieved through the exclusive use of our solar system.

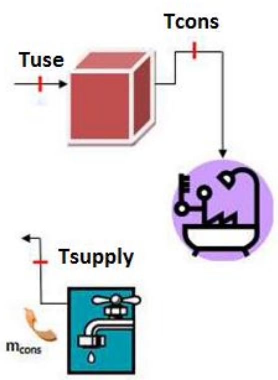


Figure 4.7: Temperature diagram

$$f = \frac{m_{cons} \cdot C_{pcons} \cdot (T_{cons} - T_{supply})}{m_{cons} \cdot C_{pcons} \cdot (T_{use} - T_{supply})} \qquad [4.24]$$

$$f = \frac{(T_{use} - T_{supply})}{(T_{cons} - T_{supply})} \qquad [4.25]$$

5.Energy balance in the tank

The energy balance in the tank, according to the following figure [F4.8], will be:

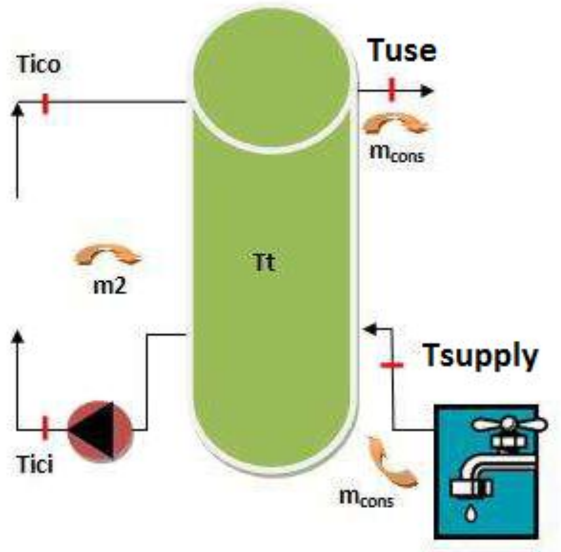


Figure 4.8: Tank energy balance

$$m_{cons} \cdot C_{pcons} \cdot (T_{use} - T_{supply}) = m_2 \cdot C_{p2} \cdot (T_{ico} - T_{ici}) \qquad [4.26]$$

- m_{cons} is the HSW consumption ($m_{cons} = 0.07\ kg/s$).
- m_2 is the secondary fluid flow in the heat exchanger.

6.Tank stratification

The meaning of this equation and the previous one is the heat transfer that takes place within the tank between the hot water from the exchanger and cold water from the external supply network and the temperature gradient that is formed within it due to stratification.

The stratification is a dimensionless coefficient that represents the temperature gradient that will occur within the tank [F4.9]:

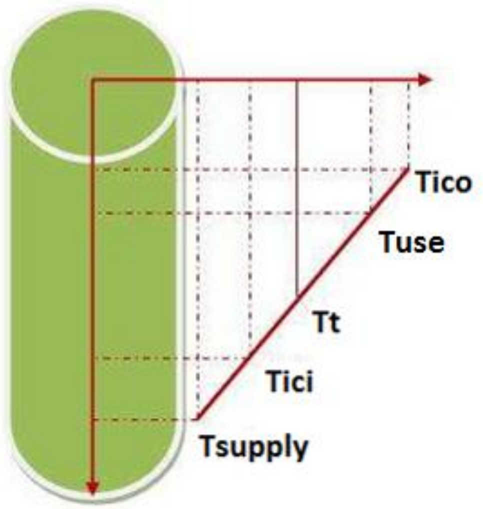


Figure 4.9: Tank stratification

$$0 \leq E_t \leq 1 \qquad [4.27]$$

The gradient within the tank is supposed to be lineal and in the extremes of the tank, it is:

$$\begin{Bmatrix} If \ \ E_t = 0, then\ T_{use} = T_t \\ If \ \ E_t = 1, then\ T_{use} = T_{ico} \end{Bmatrix} \qquad [4.28]$$

And according to the figure [F4.10]:

$$T_{ico} > T_{use} > T_t > T_{ici} > T_{supply} \qquad [4.29]$$

In a first approximation, this coefficient will be 0.25:

$$E_t = 0.25 \qquad [4.30]$$

$$E_t = \frac{T_{use} - T_t}{T_{ico} - T_t} \qquad [4.31]$$

7.Average temperature within the tank

Assuming that the tank is completely mixed and doing energy balance:

$$m_2 \cdot C_p \cdot T_{ico} + \quad m_2 \cdot C_p \cdot T_{supply} = (m_2 + m_{cons}) \cdot C_p \cdot T_t \qquad [4.32]$$

$$T_t = \frac{T_{ico} \cdot m_2 + T_{supply} \cdot m_{cons}}{m_2 + m_{cons}} \qquad [4.33]$$

ANNUAL DATA

And working out with annual data this system of seven equations with seven unknown values, the values of the temperature and the area are obtained [T4.4]**[32]**:

Table 4.4: Operating temperatures and area for the annual data

T_{use}		T_t		T_{ico}	
K	°C	K	°C	K	°C
318	45	316.7	43.7	322	49
T_{ici}		T_{co}		T_{ci}	
K	°C	K	°C	K	°C
316.5	43.5	325.3	52.3	319.2	46.2
		Area (m^2)			
		36.9			

$$T_{ico} > T_{use} > T_t > T_{ici} > T_{supply} \qquad [4.34]$$

$$322 \text{ K} > 318 \text{ K} > 316.7 \text{ K} > 316.5 \text{ K} > 283.3\ K \qquad [4.35]$$

$$Number\ of\ collectors = \frac{A}{A_{collector}} = 13.27 \approx 14\ collector \qquad [4.36]$$

And the total collection area is:

$$A = 14 \cdot 2.78 = 38.92\ m^2 \qquad [4.37]$$

After having calculated the initial estimation for the number of collectors required, the study should continue by analyzing monthly temperature and solar fraction. At this moment, it is not required to comply with the rules regarding excess capacity of the facility because the losses due to wind are not taken into account, so in this section shall only be assessed losses covered by the standard curve, these being the loss optical (a_o) and losses by conduction and convection (a_1).

The solar fraction and the actual operating temperatures are going to be calculated by the previous; but this time, instead of being the collection area one of the unknown values, it will be the solar fraction.

The mass flow through the circuit also changes when varying the absorption area; the new flow will be:

$$m_1 = m_2 = m_{man} \cdot A \cdot \rho_1 = 30 \frac{\mathrm{l}}{\mathrm{h} \cdot m^2} \cdot \frac{1\,h}{3600\,s} \cdot 38.92\,m^2 \cdot \frac{1.023\,kg}{1l}$$
$$= 0.332 \frac{kg}{s} \qquad [4.38]$$

Thus, operating temperatures for the actual annual average will be [T4.5]:

Table 4.5: Operating temperatures

T_{use}		T_t		T_{ico}	
K	°C	K	°C	K	°C
319.3	46.3	318	45	323.5	50.5
T_{ici}		T_{co}		T_{ci}	
K	°C	K	°C	K	°C
317.7	44.7	326.8	53.8	320.5	47.5

And the solar fraction will be:

$$f = 0.72 \qquad [4.39]$$

<u>MONTHLY CALCULATION</u>

The previous procedure is also useful when calculation the monthly solar fraction and working temperatures, but the data (ambient temperature, irradiation…) changes every month [T4.6].

Table 4.6: Necessary data for monthly calculation

	$I_t\ (\frac{W}{m^2})$	T_{amb}	T_{supply}	$\dot{m}_{cons}$ (kg/h)
January	328.02	279	279	318.89
February	418.77	281	280	275.83
March	474.32	284	282	261.85
April	538.71	286	284	262.52
May	531.67	291	285	250.48
June	570.32	296	286	243.25
July	661.4	301	287	216.76
August	668.68	299	286	189.06
September	583.1	294	285	217.96
October	489.07	288	284	238.97
November	356.48	284	282	249.14
December	311.39	280	279	307.45

According the legislation, the energy produced by the system should not exceed 110 % over the energy required for a single month, and it should not be higher than the energy required during three months in a row. For this purpose, those periods in which demand is below to 50% of the average for the rest of the year are not taken into account. If this point is not accomplished, heat dispersion systems (i.e. heaters) should be installed

The data obtained are shown in the table below [T4.7][T4.8]:

Table 4.7: Working temperatures and solar fraction (I)

	Jan	Feb	Mar	Apr	May	Jun
T_{use}(°C)	26.5	33.5	39.3	45.3	47.1	51.8
T_t(°C)	25.5	32.2	37.8	43.6	45.4	49.9
T_{ico}(°C)	29.6	37.5	43.8	50.5	52.4	57.7
T_{ici}(°C)	25.2	31.9	37.4	43.2	45	49.5
T_{co}(°C)	32.1	40.8	47.6	54.7	56.8	62.5
T_{ci}(°C)	27.3	34.5	40.5	46.7	48.5	53.4
f (%)	38	50	59.4	70.9	73.6	82.6

Table 4.8: Working temperatures and solar fraction (II)

	Jul	*Aug*	Sep	Oct	Nov	Dec
T_{use}(°C)	59.8	58.9	51.3	42.8	31.9	25.8
T_t(°C)	57.5	56.6	49.3	41.2	30.8	24.8
T_{ico}(°C)	66.7	65.8	57.2	47.6	35.4	28.8
T_{ici}(°C)	57	56.1	48.9	40.9	30.5	24.6
T_{co}(°C)	72.4	71.5	62.1	51.6	38.2	31.2
T_{ci}(°C)	61.6	60.7	52.9	44.1	32.8	26.6
f (%)	99.5	97.7	81.9	65	44.9	36.6

The monthly solar fraction is shown in the following figure [F4.10]:

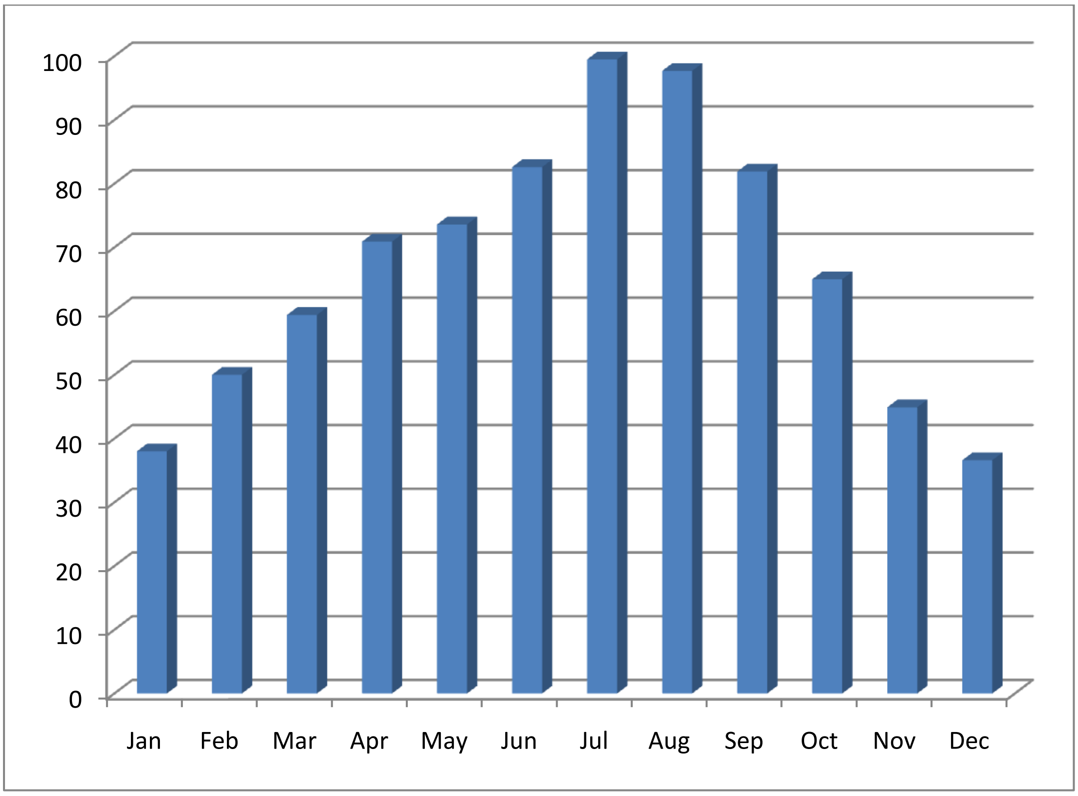


Figure 4.10: Monthly solar fraction

4.1.3.2 Calculation of the system with losses due to wind

There are two types of losses to be considered in the solar energy system: forced convection losses due to wind on the items located on the roof (collectors and pipes) and natural convection losses in the section from the roof to the room where the accumulator will be located, but these losses will be negligible **[Appendix B]**.

<u>LOSSES IN THE COLLECTOR</u>

The total thermal resistance produced by the collector due to wind will be calculated by an equivalent thermal diagram, showing all the resistances of conduction and convection resulting in the transfer of heat produced.

Thermal contact resistances have been neglected, for simplicity in calculations, and because their value is very small and can be neglected without altering any numerical result. Here is a scheme of the section of a solar panel and successive thermal resistances that are consider [F4.11]:

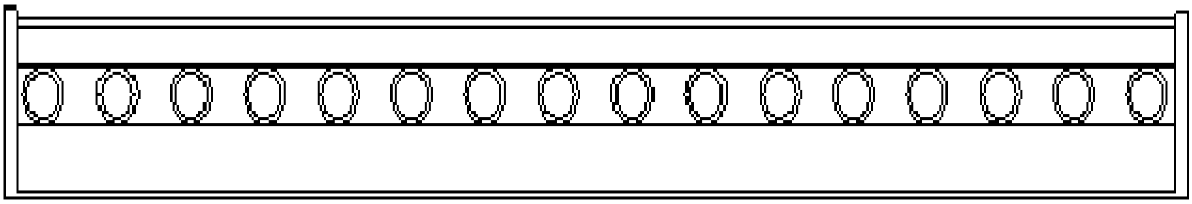


Figure 4.11: Section of a solar collector

There are 6 thermal resistances from inside the pipe to the ambient [F4.12]:

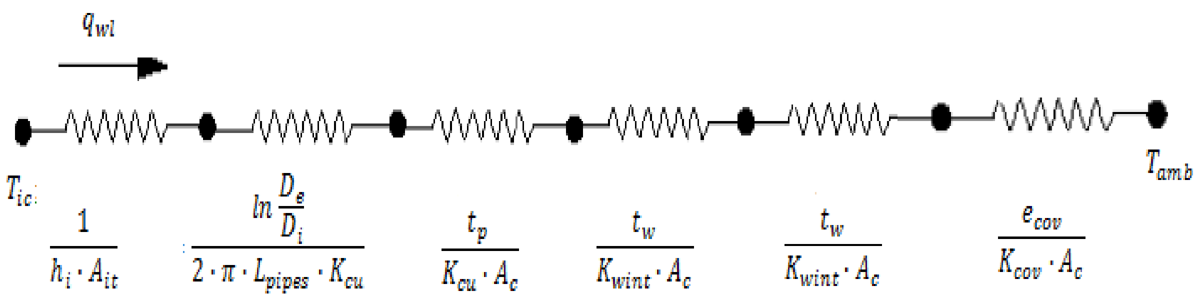


Figure 4.12: Thermal resistances (I)

- Q_{WL} is the heat that the fluid within the coil loses for the environment.
- $R_{conv,ip} = \frac{1}{h_i \cdot A_{ip}}$ is the convection thermal resistance inside the coil.
- $R_{pipe} = \frac{ln\frac{D_e}{D_i}}{2 \cdot \pi \cdot L_{pipes} \cdot k_{cu}}$ is the pipe thermal resistance.
- $R_w = \frac{t_p}{k_{cu} \cdot A_c}$ is the paint thermal resistance.
- $R_{cov} = \frac{t_w}{k_{wint} \cdot A_c}$ is the thermal resistance in the space between the absorbers and the cover.
- $R_w = \frac{e_{cov}}{k_{cov} \cdot A_c}$ is the cover thermal resistance.
- $R_o = \frac{1}{h_o \cdot A_c}$ is the convection thermal resistance on the panel surface.
- T_{amb} is the ambient temperature.
- T_{ic} is the average temperature within the coil.

And, as can be seen in Appendix B, the thermal resistances in the collector are [T4.9]:

Table 4.9: Thermal resistances (K /W)

$R_{conv,ip}$	R_{pipe}	R_p	R_w	R_{cov}	R_o
$5.1 \cdot 10^{-4}$	$7.9 \cdot 10^{-8}$	$2.7 \cdot 10^{-8}$	0.0348	$2.8 \cdot 10^{-4}$	0.004

So, the global equivalent resistance will be:

$$R_{eq} = 0.0396 \frac{K}{W} \quad [4.40]$$

$$U \cdot A = \frac{1}{R_{eq}} = 25.25 \ \frac{W}{K} \quad [4.41]$$

LOSSES IN PIPELINES.

When calculating the pipe losses, firstly, it must be differentiated the losses on the pipes depending on where they are located

- For the pipes that are outside the building, it must be taken into account the wind speed, which means the generation of forced convection.
- For pipes that are inside the building, where there is no wind, the losses are calculated depending on the natural convection due to air that is around them, these losses are really small and therefore, they are negligible.

From inside the pipe to the thermal environment, there are 4 resistances whose outline is as follows [Appendix B][F4.13]:

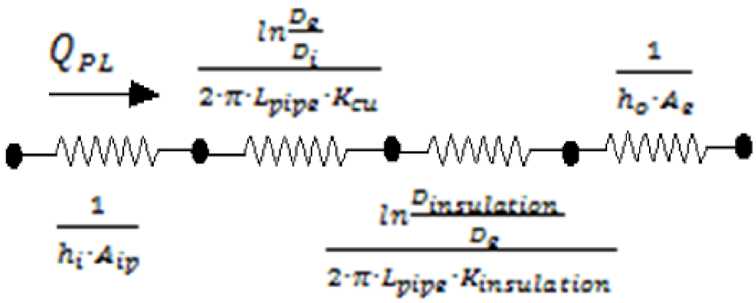


Figure 4.13: : Thermal resistances (II)

- Q_{PL} is the heat lost by the fluid from inside the pipe to the environment.

- $R_{conv,ip} = \frac{1}{h_i \cdot A_{ip}}$ is the convection thermal resistance inside the pipe.
- $R_{pipe} = \frac{ln\frac{D_e}{D_i}}{2 \cdot \pi \cdot L_{pipe} \cdot k_{cu}}$ is the copper pipes thermal resistance.

- $R_{insulation} = \frac{ln\frac{D_{insulation}}{D_e}}{2 \cdot \pi \cdot L_{pipe} \cdot k_{insulation}}$ is the insulation thermal resistance for conduction transmission.
- $R_o = \frac{1}{h_o \cdot A_e}$ is the convection thermal resistance from the insulation to the environment.

When calculating the losses of the pipes exposed to the wind, it is important also to make a distinction depending on the temperature of the water that flows through it. However, the length of both pipes will be the same:

- $L_{hw,pipe} = 20\ m.$
- $L_{cw,pipe} = 20\ m.$

Therefore, the fluid properties change because they will not be at the same temperature inside the cold and hot water pipes [F4.10].

Table 4.10: Properties of the fluid inside the cold and hot water temperature

Pipes	Heat exchanger fluid: Propylene Glycol					
	μ($\frac{Kg}{m \cdot s}$)	υ($\frac{m^3}{s}$)	ρ($\frac{Kg}{m^3}$)	k($\frac{W}{m \cdot K}$)	Pr	Cp($\frac{KJ}{kg \cdot K}$)
Cold	$2.372 \cdot 10^{-6}$	$2.328 \cdot 10^{-3}$	1019	0.433	20.731	3781
Hot	$1.676 \cdot 10^{-6}$	$1.657 \cdot 10^{-3}$	1011	0.437	14.587	3801

So, according to Appendix B, the thermal resistances in the pipes will be [T4.11]:

Table 4.11: Thermal resistances (K /W)

Pipe	$R_{conv,ip}$	R_{pipe}	$R_{insulation}$	R_o
Cold water	$3.3 \cdot 10^{-4}$	$1.9 \cdot 10^{-6}$	0.0114	0.078
Hot water	$2.8 \cdot 10^{-4}$	$1.9 \cdot 10^{-6}$	0.0114	0.078

So, the global equivalent resistance will be:

$$Cold\ water\ pipe \rightarrow R_{eq,c} = 0.0897 \frac{K}{W} \qquad [4.42]$$

$$Hot\ water\ pipe \rightarrow R_{eq,h} = 0.0896 \frac{K}{W} \qquad [4.43]$$

And finally, taking into account the small difference between both values and with all the thermal resistances, the global heat transfer coefficient can be calculated:

$$U \cdot A = \frac{1}{R_{eq}} \qquad [4.44]$$

$$Cold\ water\ pipe \rightarrow (U \cdot A)_c = 11.144 \frac{W}{K} \qquad [4.45]$$

$$Hot\ water\ pipe \rightarrow (U \cdot A)_h = 11.16 \frac{W}{K} \qquad [4.46]$$

All these values are taken in to account just the average data for a year but when calculating the monthly solar fraction it will be also necessary the monthly data.

ANNUAL CALCULATION

After the calculation of the total thermal resistance due to losses by wind in the collectors and piping, the solar fraction will be calculated with the same method used for the calculation of the solar fraction without the losses due to wind **[5][9]**.

The diagram with the working temperatures will be [F4.14]:

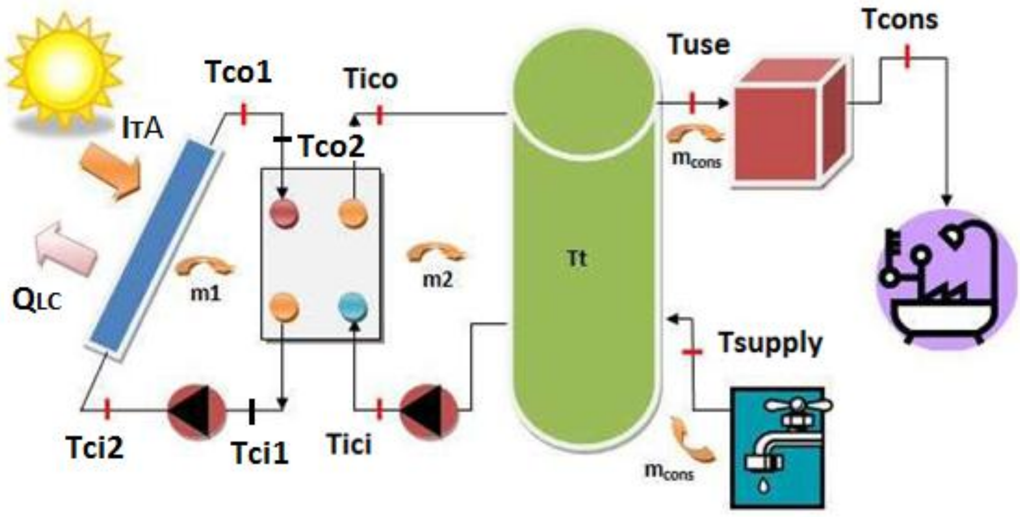


Figure 4.14: Solar energy diagram

- T_{ci1} is the ingoing collector temperature.
- T_{ci2} is the ingoing collector temperature taking into account the losses in the external pipes.
- T_{co1} is the outgoing collector temperature.
- T_{co2} is the outgoing collector temperature taking into account the losses in the external pipes..
- T_{ici} is the ingoing heat exchanger temperature.
- T_{ico} is the outgoing heat exchanger temperature.
- T_t is the average tank temperature.
- T_{use} is the outgoing tank temperature.
- f is the solar fraction.

There are some differences between this diagram and the former one. In this second diagram, there are two collector outlet temperatures due to the losses in the pipes; there are also two collector inlet temperatures.

The equations will get doing the energy balance in different circuit elements, such as in the calculation without loss, but taking into account de differences mentioned.

1. Energy balance in solar collectors

According to the previous figure [F4.15], the stored effect (stored energy per unit time) is:

$$\frac{dE_c}{dt} = I_T \cdot A - \dot{Q}_{TL} - m_1 \cdot C_{p1} \cdot (T_{co1} - T_{ci2}) = 0 \qquad [4.47]$$

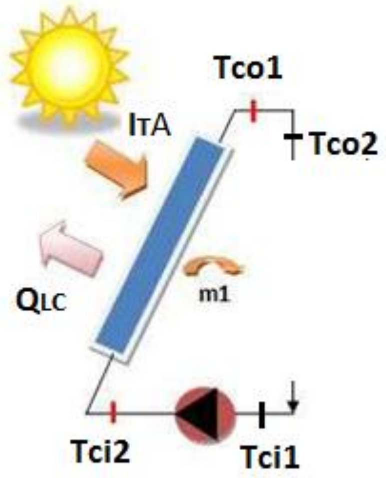


Figure 4.15: Energy balance in the collector

- The system is stationary; thus $\frac{dE_c}{dt} = 0$.
- I_T is the radiation on the inclined plane and it depends on the latitude and inclination. This parameter was calculated previously.
- A is the collector area.
- a_o is the optical coefficient of the collector, which corresponds to 0.704 for the chosen collector.
- a_1 is the coefficient of conduction and convection losses.
- Q_{TL} is defined as the heat that comes to the collector but they are not able to grasp. It can be obtained from the efficiency curve and the heat losses in the collector.

$$\eta_c = a_0 - a_1 \frac{(T_{ci2} - T_{amb})}{I_T \cdot \mathrm{A}} \qquad [4.48]$$

$$\eta_c = \frac{Q_C}{I_T \cdot \mathrm{A}} = \frac{m_1 \cdot C_{p1} \cdot (T_{co1} - T_{ci2})}{I_T \cdot \mathrm{A}} = 1 - \frac{Q'_{TL}}{I_T \cdot \mathrm{A}} \qquad [4.49]$$

$$Q'_{TL} = I_T \cdot \mathrm{A} \cdot \left(1 - a_0 + a_1 \frac{(T_{ci2} - T_{amb})}{I_T}\right) \qquad [4.50]$$

$$Q_{TL} = Q'_{TL} - \left(\frac{1}{R_{eq,col}}\right) \cdot \left[\frac{(T_{co1} - T_{amb}) - (T_{ci2} - T_{amb})}{ln\left(\frac{T_{co1} - T_{amb}}{T_{ci2} - T_{amb}}\right)}\right]$$

- $m_1 \cdot C_{p1} \cdot (T_{co1} - T_{ci2})$ is the useful heat.

And finally, the energy balance in the collector is:

$$m_1 \cdot C_{p1} \cdot (T_{co1} - T_{ci2}) = I_T \cdot A - \left[1 - \left(a_o - \frac{T_{ci2} - T_{amb}}{I_T} \cdot a_1\right)\right] \cdot I_T \cdot A - \left(\frac{1}{R_{eq,col}}\right) \cdot \left[\frac{(T_{co1} - T_{amb}) - (T_{ci2} - T_{amb})}{ln\left(\frac{T_{co1} - T_{amb}}{T_{ci2} - T_{amb}}\right)}\right] \quad [4.51]$$

2. Energy balance in the heat exchange

In the same way, accorfing to the figure [F4.16], the energy balance in the heat exchanger is:

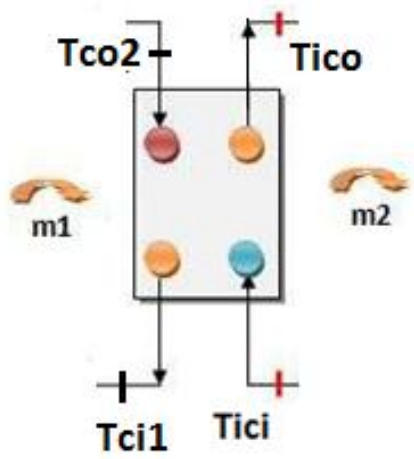


Figure 4.16: Energy balance in the heat exchanger

$$m_1 \cdot C_{p1} \cdot (T_{co2} - T_{ci1}) = m_2 \cdot C_{p2} \cdot (T_{ico} - T_{ici}) \quad [4.52]$$

As previously mentioned, the heat exchanger in symmetric or balanced; so, $\dot{m}_1 = \dot{m}_2$.

3. Heat exchanger efficiency

The heat exchanger efficiency will be:

$$\varepsilon_{HC} = \frac{\dot{m}_2 \cdot C_{p2} \cdot (T_{ico} - T_{ici})}{C_{min} \cdot (T_{co2} - T_{ici})} \qquad [4..53]$$

- C_{min} is the minimum value between $\dot{m}_1 \cdot C_{p1}$ and $\dot{m}_2 \cdot C_{p2}$.

4. Energy balance in the secondary circuit

The energy balance in the secondary circuit will be:

$$m_{cons} \cdot C_{pcons} \cdot \left(T_{use} - T_{supply}\right) = m_2 \cdot C_{p2} \cdot (T_{ico} - T_{ici}) \qquad [4.54]$$

5. Solar fraction

Taking into account use, consumption and supply temperatures [F4.17], the solar fraction (f) will be:

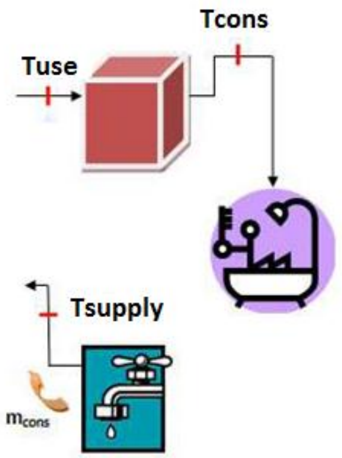


Figure 4.17: Temperature diagram

$$f = \frac{m_{cons} \cdot C_{pcons} \cdot \left(T_{cons} - T_{supply}\right)}{m_{cons} \cdot C_{pcons} \cdot \left(T_{use} - T_{supply}\right)} \qquad [4.55]$$

$$f = \frac{\left(T_{use} - T_{supply}\right)}{\left(T_{cons} - T_{supply}\right)} \qquad [4.56]$$

6. Stratification in the tank

The stratification temperature in the tank will be according to the following formula [F4.18]:

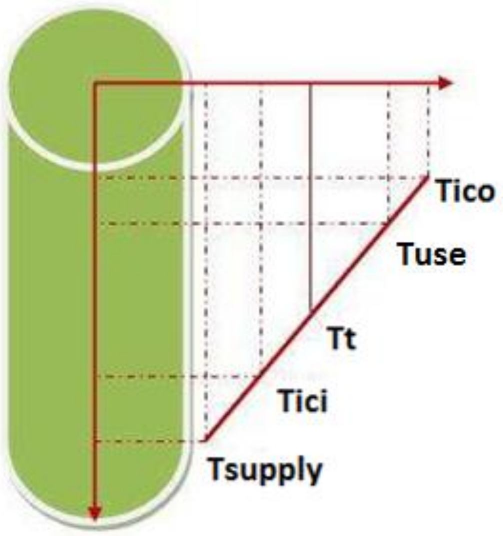


Figure 4.18: Tank stratification

$$E_t = \frac{T_{use} - T_t}{T_{ico} - T_t} \qquad [4.57]$$

7. Average temperature within the tank

The average temperature within in the tank will be:

$$T_t = \frac{T_{ico} \cdot m_2 + T_{supply} \cdot m_{cons}}{m_2 + m_{cons}} \qquad [4.58]$$

8. Energy balance in the hot water pipe

The energy balance in the hot water pipe will be:

$$m_1 \cdot C_{p1} \cdot (T_{co1} - T_{c02}) = \frac{1}{R_{eq,h}}) \cdot \left[\frac{(T_{co1} - T_{amb}) - (T_{co2} - T_{amb})}{ln\left(\frac{T_{co1} - T_{amb}}{T_{co2} - T_{amb}}\right)} \right] \qquad [4.59]$$

9. Energy balance in the cold water pipe

And finally, the energy balance in the cold water pipe will be:

$$m_2 \cdot C_{p2} \cdot (T_{ci1} - T_{ci2}) = \frac{1}{R_{eq,c}}) \cdot \left[\frac{(T_{ci1} - T_{amb}) - (T_{ci2} - T_{amb})}{ln\left(\frac{T_{ci1} - T_{amb}}{T_{ci2} - T_{amb}}\right)} \right] \quad [4.60]$$

After all equations have been developed all the equations, it is necessary to calculate the annual and monthly working temperatures and solar fraction.

The annual data obtained will be [T4.12]:

Table 4.12: Yearly solar fraction and working temperatures

T_{use}(°C)	44.9
T_t(°C)	43.6
T_{ico}(°C)	49
T_{ici}(°C)	43.4
T_{co1}(°C)	52.5
T_{co2}(°C)	52
T_{ci1}(°C)	46.3
T_{ci2}(°C)	45.9
f (%)	70.4

MONTHLY CALCULATION

As before, for the monthly calculation, it will be needed the equivalent resistances in the pipes and the collector and the previously calculated variable data calculated for each month are the data shown in the previous points[T3.1][T3.6][T4.9][T4.11]:

And dhe data obtained are shown in the table below [T4.13][T4.14]**[32]**:

Table 4.13: Working temperatures and solar fraction (I)

	Jan	Feb	Mar	Apr	May	Jun
T_{use}(°C)	25.7	32.7	38.33	44	45.8	50.3
T_t(°C)	24.7	31.4	36.8	42.3	44.12	48.4
T_{ico}(°C)	28.6	36.6	42.7	49	51	56
T_{ici}(°C)	24.4	31.1	36.5	42	43.7	48
T_{co1}(°C)	31.4	40	46.7	53.44	55.5	60.9
T_{co2}(°C)	30.9	39.	46.2	52.9	55	60.4
T_{ci1}(°C)	26.8	34	39.8	45.7	47.5	52.1
T_{ci2}(°C)	26.3	33.3	39.3	45.2	47	51.6
f (%)	36.8	48.7	57.9	68.8	71.5	80

Table 4.14: Working temperatures and solar fraction (II)

	Jul	Aug	Sep	Oct	Nov	Dec
T_{use}(°C)	58.1	57.3	49.7	41.3	30.8	24.9
T_t(°C)	55.8	55.1	47.7	39.8	29.7	23.9
T_{ico}(°C)	64.8	64	55.4	46	34.1	27.7
T_{ici}(°C)	55.3	54.6	47.3	39.5	29.4	23.7
T_{co1}(°C)	70.5	69.9	60.4	50	37.1	30.4
T_{co2}(°C)	70	69.4	59.9	49.5	36.6	29.9
T_{ci1}(°C)	60.1	59.4	51.5	42.9	32	26
T_{ci2}(°C)	59.6	58.9	51	42.4	31.5	25.4
f (%)	96.6	95	79.3	62.7	43.3	35.3

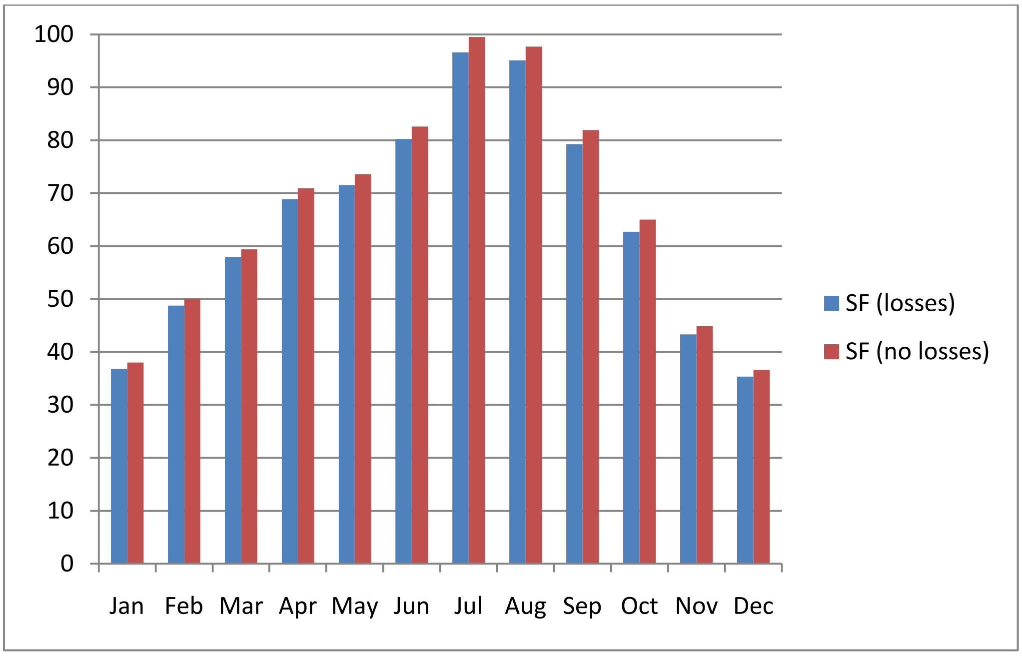


Figure 4.19: Monthly solar fraction

The monthly solar fraction is shown in the following figure [F4.19]. Therefore, there is not any month in which the energy obtained is greater than the energy required, so it is not necessary to add any element that the excess energy.

4.1.4 Heat Exchanger

As previous mentioned, the heat exchanger is a key element in the system as it is the one which transfers the energy captured in the solar collector (primary circuit: water/propylene glycol) to the storage tank of hot water (auxiliary circuit: drinking water).

The proper kind for this system will be a plate heat exchanger with the same flow for both circuits (symmetric flow) and forced circulation. It will be a cross-flow exchanger to provide a greater efficiency and there will be one step through the exchanger per fluid.

When dimensioning the exchanger, it is necessary to know the annual ingoing and outgoing temperature of both fluids that have been calculated previously.

The temperature values for the primary circuit, the fluid that comes from the collector, are:

$$T_{ih} = T_{co2} = 52.02\ °\mathrm{C} = 325.02\ \mathrm{K} \quad [4.61]$$

$$T_{oh} = T_{ci1} = 46.36\ °\mathrm{C} = 319.36\ \mathrm{K} \quad [4.62]$$

And the temperatures values for the secondary circuit (consumption water) are:

$$T_{ic} = T_{ico} = 48.97\ °\mathrm{C} = 321.971\ \mathrm{K} \quad [4.63]$$

$$T_{oc} = T_{ici} = 43.42\ °\mathrm{C} = 316.42\ \mathrm{K} \quad [4.64]$$

However, for the design of heat exchanger is necessary to know the heat to transfer power from primary to secondary circuit. Then, with its data, this coefficient should be equal to that calculated.

For this purpose, firstly, the overall coefficient of heat transfer will be calculated choosing a particular exchanger model [T4.15][F4.20]**[27]**.

Table 4.15: Heat exchanger

Heat exchanger	
Manufacturer	Alfa Laval
Model	M3-FM
Type	Plate exchanger
Material of the plates	Stainless steel
Maximum working pressure	1 MPa
Maximum working temperature	140 °C
Price	601 €

And some necessary data of this exchanger will be:

- a is the exchange area per plate ($a = 0.032\ m^2$).

- b ihe distance between plates ($b = 0.0024\ m$).
- L is the high of each plate ($L = 0.246\ m$).
- $w_p$is the width of each plate ($w_p = 0.180\ m$).
- t_p is the thickness of each plate ($t_p = 0.0005\ m$).
- K_{Al} is the thermal conductivity of stainless steel ($K_{Al} = 15\ \frac{W}{m \cdot K}$).
- L_c is the characteristic length ($L_c = 2 \cdot b = 0.0048\ m$).
- A_{plate} is the area of each plate ($A_{plate} = L \cdot w_p = 0.044\ m^2$).
- A_{canal} is the area of each canal ($A_{canal} = b \cdot w_p = 4.32 \cdot 10^{-4}\ m^2$).

Figure 4.20: Alfa Laval M3-FM heat exchanger

The properties of both fluids (water/propylene glycol in the primary and water in the secondary) will be [T4.16]:

Table 4.16: Fluid properties

Circuit	Heat exchanger fluid: Propylene Glycol; Secondary fluid:Water					
	μ($\frac{Kg}{m \cdot s}$)	υ($\frac{m^2}{s}$)	ρ($\frac{Kg}{m^3}$)	k($\frac{W}{m \cdot K}$)	Pr	Cp($\frac{KJ}{kg \cdot K}$)
Primary	$2.869 \cdot 10^{-3}$	$2.805 \cdot 10^{-6}$	1023	0.431	25.127	3.771
Secondary	10^{-3}	10^{-6}	1000	0.63	6.64	4.188

And the flow used for the calculations is the same that the one used in previous calculations for wind losses:

$$m_1 = m_2 = 0.332 \frac{kg}{s} \quad [4.65]$$

When all this information, it is possible to calculate the output heat:

$$Q_1 = Q_2 = 7.08\,K \quad [4.138]$$

$$Q_1 = m_1 \cdot c_{p_1} \cdot \Delta T_1 = m_1 \cdot c_{p_1} \cdot (T_{co2} - T_{ci1}) \quad [4.66]$$

$$Q_2 = m_2 \cdot c_{p_2} \cdot \Delta T_2 = m_2 \cdot c_{p_2} \cdot (T_{ico} - T_{ici}) \quad [4.67]$$

And, as previous mentioned, according to the log mean temperature difference the transfer heat will be **[4][20]**:

$$Q = U \cdot A \cdot \Delta T_{lm} \rightarrow U \cdot A = \frac{Q}{\Delta T_{lm}} \quad [4.68]$$

- ΔT_{lm} is log mean temperature difference ($\Delta T_{lm} = \frac{(T_{co2}-T_{ico})-(T_{ci1}-T_{ici})}{\ln(\frac{T_{co2}-T_{ico}}{T_{ci1}-T_{ici}})} = 299\ K$).

$$U \cdot A = \frac{Q}{\Delta T_{lm}} = \frac{Q}{\Delta T_{lm}} = 2364.02\ \frac{W}{K} \quad [4.69]$$

And the equivalent resistance will be:

$$R_{eq} = \frac{1}{U \cdot A} = 4.22 \cdot 10^{-4}\ \frac{K}{W} \quad [4.70]$$

And when the equivalent resistance is known, it is possible to know the number of plates (n) needed for the hear exchange, that will be 155 **[27]**. Then, taking into account, the overall heat coefficient will be calculated through the following equations:

$$\frac{1}{U \cdot A} = \frac{1}{h_1} + \frac{t_p}{K_{Al}} + \frac{1}{h_2} + R_1 + R_2 = R_{eq} \quad [4.71]$$

R_1 and R_2 are resistances of the are the resistances of the contaminants deposits of each fluid. Normally, it is around$10^{-5}\ \frac{m^2 \cdot K}{W}$; so, it is not going to take into account during designing this exchanger and the equivalent resistance will be:

$$\frac{1}{U \cdot A} = \frac{1}{h_1} + \frac{t_p}{K_{Al}} + \frac{1}{h_2} = R_{eq} \quad [4.72]$$

When the number of plates is known, it is necessary to calculate the convection coefficient h_1 and h_2; as have been done in the previous steps:

$$h = \frac{Nu \cdot k}{L_c} \quad [4.73]$$

- A is the total area of the heat exchanger ($A = A_{plate} \cdot n = 6.82\ m^2$).
- m_{canal} is the flow through each exchanger canal ($m_{canal} = \frac{2 \cdot m_1}{n+1} = 0.00425 \frac{kg}{s}$).
- $-\ u_{1,canal}$ and $u_{2,canal}$ are the velocities of each flow through the exchanger canals. These values are needed to calculate the Reynolds number.

$$u_{1,canal} = \frac{m_{canal}}{\rho_1 \cdot A_{canal}} = 9.61 \cdot 10^{-3} \frac{m}{s} \quad [4.74]$$

$$u_{2,canal} = \frac{m_{canal}}{\rho_2 \cdot A_{canal}} = 9.8 \cdot 10^{-3} \frac{m}{s} \quad [4.75]$$

- Re_1 and Re_2 are the Reynolds numbers of the fluids.

$$Re_1 = \frac{\rho_1 \cdot u_{1,canal} \cdot L_c}{\mu_1} = 16.44 \quad [4.76]$$

$$Re_2 = \frac{\rho_2 \cdot u_{2,canal} \cdot L_c}{\mu_2} = 47 \quad [4.77]$$

And the Nusselt number will be:

$$Nu_1 = 0.4 \cdot {Re_1}^{0.64} \cdot {Pr_1}^{0.4} = 8.71 \quad [4.78]$$

$$Nu_2 = 0.4 \cdot {Re_2}^{0.64} \cdot {Pr_2}^{0.4} = 10.02 \quad [4.79]$$

- $Pr_1$and $Pr_2$are the Prandtl numbers ($Pr_1 = 25.127$ and Pr_2=6.64).

And after all these values are calculated, the convection coefficient will be:

$$Nu = \frac{h \cdot L_c}{k} \rightarrow h_1 = \frac{k_1 \cdot Nu_1}{L_c} = 782.08 \frac{W}{m^2 \cdot K} \quad [4.80]$$

$$h_2 = \frac{k_2 \cdot Nu_2}{L_c} = 1312.5 \frac{W}{m^2 \cdot K} \quad [4.81]$$

And the equivalent resistance will be:

$$R_{eq} = \frac{1}{h_1} + \frac{t_p}{K_{Al}} + \frac{1}{h_2} = 0.000672 \frac{K}{W} \quad [4.82]$$

Finally, the global transmission heat coefficient will be:

$$R_{eq} = \frac{1}{U \cdot A} \to U \cdot A = \frac{1}{R_{eq}} \quad [4.83]$$

$$U \cdot A = 1488.09 \frac{W}{K} \quad [4.84]$$

And this values is similar to the one calculated at the beginning of the heat exchanger, so the chosen exchanger is proper for this facility.

4.1.5 Circulation Pump

The heat transfer fluid transport along the primary circuit is done with the help of circulating pumps, which are devices powered by an electric motor capable of supplying the fluid an amount of energy to overcome the resistance of the fluid passing through the pipe and other elements **[12]**.

There are three main types of circulating pumps:

- Reciprocating pumps.

- Rotary pumps.

- Centrifugal pumps which are the type used in solar energy systems.

Due to the length of the circuit, the load losses are important and they are necessary to be calculated for the correct choice of pumps. The solar collector is located on the roof of the building, while the heat exchanger, the storage tank and the boiler in the basement of the building, without being outdoors.

The building is 51 meters long from the basement to the roof, while the piping system, both upstream and downstream is 142 meters.

The largest pressure drop will be on this circuit, because the height difference has to be overcome there to transport the fluid from the basement to the roof.

The power of the pump needed for the facility will be calculated using the following expressions**[12]**:

$$W_p = \frac{m \cdot \Delta p}{\rho \cdot \eta} \quad [4.85]$$

- W_p is the pump power (W).
- Δp is the pressure variation along the circuit (Pa).
- m is the flow through the primary circuit ($\frac{kg}{s}$).
- ρ_1 is the fluid density ($\frac{kg}{m^3}$).
- η is the efficiency of the conversion from electrical energy to flow energy.

And the pressure drop experienced by the fluid will be:

$$\Delta p_{pump} = \Delta p_{circ} + \Delta p_{he} + \Delta p_{col} \quad [4.86]$$

Pressure drop in the collectors

According to the manufacturer, the pressure drop in the collectors used in this facility will be:

$$\Delta p_{1,col} = 260\ Pa \quad [4.87]$$

As the collectors are connected in parallel, the pressure drop of all collectors will be equivalent to the pressure drop in a single collector; so the total pressure drop in the collectors will be:

$$\Delta p_{col} = 260\ Pa \quad [4.88]$$

Pressure drop in the heat exchanger

The pressure drop in plate heat exchangers is normally calculated as:

$$\Delta p_{he} = 4 \cdot f \cdot \frac{L}{L_c} \cdot \frac{\rho_1 \cdot u_{canal}{}^2}{2} \quad [4.89]$$

- L is the length pipe ($L = 0.246\ m$)
- L_c is the characteristic length ($L_c = 0.0048 m$).
- $\rho_1$is the heat exchanger fluid density ($\rho_{wind} = 1023 \frac{kg}{m^3}$).
- u_{canal} is the velocity of the fluid through a canal of the heat exchanger ($u_{canal} = 9.61 \cdot 10^{-3} \frac{m}{s}$).
- f is the friction factor.

$$f = C \cdot Re^M \quad [4.90]$$

And C and M are constant values depending on the Reynolds number [T4.17]:

Table 4.17: C and M constant values.

Re	C	M
$Re < 10$	17	-1
$10 < Re < 101$	6.29	-0.57
$101 < Re < 855$	1.141	-0.2
$Re > 855$	0.581	-0.1

Reynolds number will be:

$$Re_1 = \frac{\rho_1 \cdot u_{1,canal} \cdot L_c}{\mu_1} = 16.44 \quad [4.91]$$

And therefore, the friction factor and the pressure loss in the heat exchanger will be:

$$f = 6.29 \cdot Re^{-0.57} = 1.27 \quad [4.92]$$

$$\Delta p_{he} = 12.29\ Pa \quad [4.93]$$

Pressure drop in the circuit

The pressure drop the circuit is normally calculated as:

$$\Delta p_{circ} = 0.5 \cdot \rho_1 \cdot u_1{}^2 \cdot (f \cdot \frac{L_{pipec}}{D_i} + k_{losses}) \quad [4.94]$$

Continuous load losses are those generated by the friction of the fluid with the pipe wall and therefore they depend on:

- L_{pipe} is the length of the hot and cold water pipes($L_{pipe} = 142\ m$).
- $\rho_1$is the heat exchanger fluid density ($\rho_{wind} = 1023 \frac{kg}{m^3}$).
- D_i is the internal diameter ($D_i = 20\ mm$).
- u_1 is the fluid velocity ($m = \rho \cdot u \cdot A \rightarrow u = \frac{m}{\rho \cdot A} \rightarrow u_1 = \frac{0.332}{1023 \cdot \frac{\pi \cdot 0.02^2}{4}} = 1 \frac{m}{s}$) [4.95].
- f is the friction factor, is calculated from the Reynolds number and relative roughness, through the curves of the Moody diagram.

The roughness coefficient of copper is $\varepsilon_{Cu} = 0.0015mm$, so the relative roughness coefficient will be:

$$k = \frac{\varepsilon_{Cu}}{D_i} = \frac{0.0015\ mm}{20\ mm} = 0.000075 \quad [4.96]$$

And the Reynolds number will be:

$$Re_1 = \frac{\rho_1 \cdot u_1 \cdot D_i}{\mu_1} = 7131 > 2300 \rightarrow Turbulent\ flux. \quad [4.97]$$

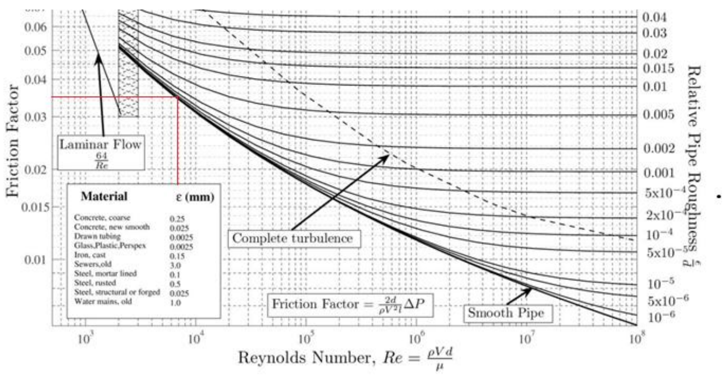


Figure 4.21: : Moody diagram

When the value of the Reynolds number and relative roughness, it is possible to obtain the friction factor with the Moody diagram [F4.21]:

$$f = 0.035 \quad [4.98]$$

And this value can be checked using the Gnielinski correlation shown below, concluding that the friction factor is similar to the one obtained with the Moody diagram:

$$f = (0.7901 \cdot ln\, Re - 1.64)^{-2} = 0.0347 \quad [4.99]$$

And the located pressure losses depend on:

- K_{losses} are the losses in the elements of the circuit.

They can be produced by the following elements:

- Input or output pipes.
- Abrupt widening or contraction.
- Curves, elbows and T´s.
- Valves open or partially closed.
- Gradual widening or contractions.

Just the elbows and the entrances and exits of the deposit will be taken into account, neglecting the losses due to partially closed valves.

Usually localized pressure losses are measured experimentally and correlated with flow parameters, so they depend largely on the manufacturer of the elements and the values of this study will be estimations.

In the solar circuit there are no abrupt entrances and exits of consideration, so the pressure loss due to elbows is the only important loss [T4.18]:

Table 4.18: Losses in the primary circuit

Element of the circuit	k	Amount	Total
Pipe elbow	1.2	12	14.4

And the losses along the primary circuit will be:

$$\Delta p_{circ} = 0.5 \cdot \rho_1 \cdot u_1{}^2 \cdot \left(f \cdot \frac{L_{pipec}}{D_i} + k_{losses}\right)$$

$$= 0.5 \cdot 1023 \cdot 1^2 \cdot \left(0.035 \cdot \frac{142}{0.02} + 14.4\right) = 134473\ Pa \quad [4.100]$$

And therefore, the total pressure loss in the primary circuit will be:

$$\Delta p_{pump} = \Delta p_{circ} + \Delta p_{he} + \Delta p_{col} = 134476\ Pa \quad [4.101]$$

And with this data, the power of the electric motor is calculated by the equation shown below:

$$W_p = \frac{m \cdot \Delta p}{\rho \cdot \eta} \quad [4.102]$$

The actual efficiency of the pump depends on the type and size. For a small size, as those used in solar energy systems, its value is around 0.5. So, the power will be:

$$W_p = 87.27\ W \quad [4.103]$$

However, it is advisable to use a slightly larger power pump because over time the pressure loss may increase due to wear of the instrument, dirt, etc…

The pump chosen for the facility will by a centrifugal pump from the manufacturer Grundfos and the most proper model is Grundfos Solar 15-65 [T4.19]**[25]**.

Table 4.19: Pump

Pump	
Manufacturer	Grundfos
Model	Solar 15-65
Type	Centrifugal pump
Maximum working pressure	10 bar
Price	350 €

4.1.6 Expansion tank

The expansion tank is a tank whose purpose is to absorb the excess pressure of the fluid that goes through the circuit, due to the volume increase that occurs when the fluid expands by heating. It will prevent fluid loss that would occur when the safety valve is activated the safety if high pressures are achieved, so it would not be necessary to fill the system with new fluid to keep it pressurized and therefore it would not increase the cost of operation and maintenance.

Deposits are classified into open or closed expansion tanks, but in any case, its capacity must be sufficient to support the expansion of the heat transfer fluid.

In solar energy systems, the expansion tank is closed because they are easy to assemble and require no insulation and they do not absorb oxygen from the air.

The data necessary for the selection of the tank is:

- Total volume of water (liters) at the facility.
- Minimum operating temperatura ($T = 4$ °C).
- Maximum temperature that can reach the water during operation of the facility.
- Minimum and maximum working pressures (1.5 and 4 bar).
- Expansion volume (l).

The coefficient of expansion of water for temperatures from 4 °C to 90 °C, taking into account the increased volume of water accompanied by an increase in the volume available due to the simultaneous expansion of circuit components, can be expressed by the following equation:

$$Ce = (0.24 \cdot T^2 + 102.13 \cdot T - 2708.3) \cdot 10^{-6} \quad [4.104]$$

$$Ce = 0.044 \quad [4.105]$$

If the heat transfer fluid is a solution of propylene glycol in water, Ce expansion coefficient should be multiplied by the following correction factor:

$$f = a \cdot (1.8 \cdot T + 32)^b \quad [4.106]$$

- $a = -0.0134 \cdot (G^2 - 143.8 \cdot G - 1918.2)$
- $b = 3.5 \cdot 10^{-4} \cdot (G^2 - 94.57 \cdot G + 500)$

- G is the percentage of propylene glycol in water, valid for the range 20% -50% by volume ($G = 0.395$).

$$a = 26.46; b = 0.162; f = 23.11 \quad [4.107]$$

$$Ce = 0.044 \cdot 23.11 = 1.02 \quad [4.108]$$

The expansion coefficient is always positive and less than 1 and represents the relationship between the effective volume of expansion tank, which must be equal the fluid volume expansion, and the fluid volume in the system.

Then it is necessary to calculate the pressure coefficient valid for closed expansion tanks without fluid transfer out of the system. This value is based on the state equation for perfect gases, taking into account that the volume change takes place at constant temperature. The coefficient is positive and bigger than 1 and it represents the ratio between the total and net volume of the expansion tank.

$$Cp = \frac{P_{max}}{P_{max} - P_{min}} \quad [4.109]$$

- P_{max} is the maximum pressure in the expansion tank, it will be equal to the static pressure ($P_{max} = 0.5\ bar$).
- P_{min} is the minimum pressure in the expansion tank, it will be similar to the opening value of safety valve ($P_{min} = -0.5\ bar$).

$$Cp = \frac{0.5}{0.5 - (-0.5)} = 0.5 \quad [4.110]$$

For a closed expansion tank, with fluid in direct contact with a pressurized gas, the total tank volume can be calculated using the following expression:

$$Vt = Vfac \cdot Cp \cdot Ce \quad [4.111]$$

- $Vfac$ is the total volume of the facility. It is the sum of the volume of collectors, heat exchanger and piping system.

$$Vfac = (Vcol \cdot 14) + b \cdot a \cdot (n-1) + L_{pipe} \cdot \frac{\pi \cdot D_i{}^2}{4}$$

$$= 1.64 \cdot 10^{-3} \cdot 14 + 0.032 \cdot 0.0024 \cdot (155-1) + 142 \cdot \frac{\pi \cdot 0.02^2}{4}$$

$$= 0.077\ m^3 = 77\ l \quad [4.112]$$

Finally, the total volume will be:

$$Vt = 77 \cdot 0.5 \cdot 1.02 = 46.2\ l \quad [4.113]$$

The expansion tank chosen for the faciliry is the model Solar-Plus 25 from Zilmer manufacturer [T4.20][F4.22]**[26]**.

Table 4.20: Expansion tank

Expansion tank	
Manufacturer	Zilmet
Model	Solar-Plus 50
Type litres	50
Diameter mm	300
Height mm	392
Max pressure	10
Price	150 €

Figure 4.22: Zilmet Solar-Plus 50 expansion tank

4.2 Secondary Circuit

The auxiliary circuit is responsible for heating the water from heat exchanger to supply the HSW required in the building.

The fluid flowing through the auxiliary circuit is the water from supply network that comes to the solar collector. The fluid in the bottom of the tank is driven by pump to heat exchanger, where captures the energy of the heat transfer fluid and re-enters in the tank. At the top of the tank there is outlet water that goes to the support system.

4.2.1 Circulation pump

The method to calculate this pump will be the same that the used in the calculation of the primary circuit pump **[12]**:

$$W_p = \frac{m \cdot \Delta p}{\rho \cdot \eta} \quad [4.113]$$

- W_p is the pump power (W).
- Δp is the pressure variation along the circuit (Pa).
- m_s is the flow through the primary circuit ($\frac{kg}{s}$).
- ρ_2 is the fluid density ($\frac{kg}{m^3}$).
- η is the efficiency of the conversion from electrical energy to flow energy.

The flow through the secondary or auxiliary circuit is the same that flows through the solar or primary circuit, as have been justified in the previous calculations.

$$\Delta p_{pump} = \Delta p_{pipes} + \Delta p_{he} + \Delta p_{ac} \quad [4.114]$$

- Δp_{pipes} are the losses in the piping system.
- Δp_{he} are the losses in the heat exchanger
- Δp_{ac} are the losses in the accumulator.

Pressure drop in piping system

The pipe system is approximately 1 meter long; 0.5 meters for the trip from the accumulator to the exchanger and 0.5 meters for the trip from the exchanger to the accumulator

So, when the fluid through the circuit ($m_1 = 0.332 \frac{kg}{s}$) and the internal diameter ($D_i = 20\ mm$) are known, the pressure losses per meter can be calculated with the copper losses diagram [F4.23]:

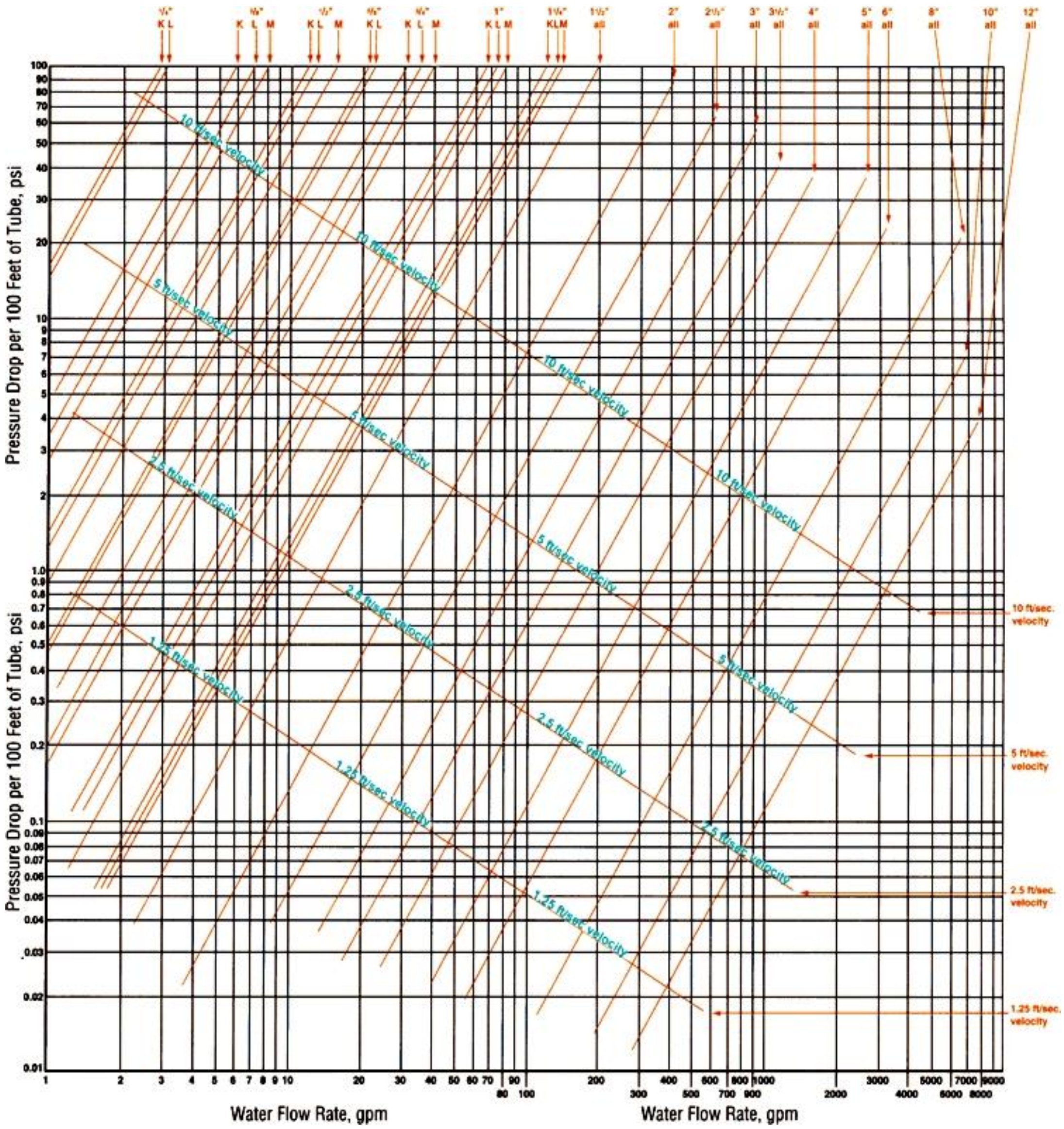


Figure 4.23: Pressure losses diagram (copper)

$$\Delta p_{pipes} = 120\ Pa \quad [4.115]$$

Pressure drop in heat exchanger

The losses in the heat exchanger are considered the same as in the primary circuit:

$$\Delta p_{he} = 12.29\ Pa \quad [4.116]$$

Pressure drop in accumulator and heat exchanger

The pressure loss in the accumulators is estimated at 10% of the total sum of exchanger and pipe losses.

$$\Delta p_{ac} = 0.1 \cdot \left(\Delta p_{pipes} + \Delta p_{he}\right) = 13.229 Pa \quad [4.117]$$

And therefore, the pressure drop at the pump in secondary auxiliary circuit is:

$$\Delta p_{pump} = \Delta p_{pipes} + \Delta p_{he} + \Delta p_{ac} = 157.8\ Pa \quad [4.118]$$

And with this data, the power of the electric motor is calculated by the equation shown below:

$$W_p = \frac{m \cdot \Delta p}{\rho \cdot \eta} = 0.314\ W \quad [4.119]$$

Although the auxiliary circuit pump power is a significantly lower than the solar circuit pump power, the model chosen for the facility will be the same than the solar circuit model [T4.22]**[25]**.

Table 4.21: Secondary circuit pump

Secondary circuit pump	
Manufacturer	Grundfos
Model	Solar 15-65
Type	Centrifugal pump
Maximum working pressure	10 bar
Price	350 €

4.2.2 Solar accumulator

The solar thermal energy facility for HSW requires an energy storage system that enables docked, in time, the supply of solar radiation and hot water demand **[16][19]**.

According to legislation, solar accumulation volume must meet the following relationship with the collector area:

$$50 < \frac{V}{A} < 180 \quad [4.120]$$

Therefore, the accumulation volume must be the similar to the average daily hot water demand and according to the collector area calculated previously will be within the following ranges:

$$50 \cdot A < V < 180 \cdot A \rightarrow 50 \cdot 38.92 < V < 180 \cdot 38.92 \rightarrow 1946 < V < 7000.56 \quad [4.121]$$

$$Q_d = 2288 \frac{litres \cdot person}{day} \quad [4.122]$$

And a solar accumulator proper to this facility could be [T4.21][F4.24]**[28]**:

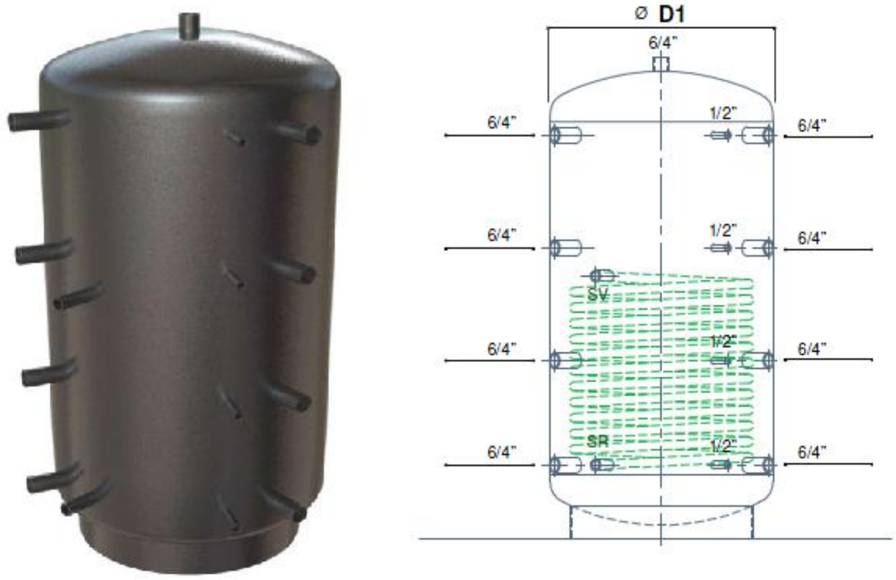


Figure 4.24: Solar accumulator Acro ISO PS 2500

Table 4.22: Solar accumulator

Solar accumulator	
Manufacturer	Acro ISO
Model	PS series
Volume	2500 l
Price	2000 €

4.3 Support Circuit

The support circuit is responsible for heating the water from the auxiliary system at 60 °C when it does not reach that temperature **[9]**.

The water comes from the auxiliary circuit, if its temperature is less than 60°C, it passes through the boiler and then enters the hot water storage tank. If its temperature is 60 °C or more, it takes the alternative route, bypassing the boiler and entering directly into the hot sanitarian water accumulator.

4.3.1 Accumulation tank

It is the element when the hot sanitarian water is stored at 60 °C before the consumption.

When dimensioning the accumulator tank, firstly, it is necessary to calculate a shape factor and a mix factor that is the ratio between the useful volume of the accumulator and the real volume.

$$f_s = 2.5 \quad [4.123]$$

$$f_m = 0.63 + 0.14 \cdot f_s = 0.98 \quad [4.124]$$

Peak period T is the peak consumption duration and it is calculated using the following equation:

$$T = 5 \cdot \frac{N^{0.905}}{15 + N^{0.92}} \quad [4.125]$$

- N is the number of flats in the bilding (N =26).

$$T = 5 \cdot \frac{26^{0.905}}{15 + 26^{0.92}} = 2.72\ h \quad [4.126]$$

Then, it is necessary to calculate the simultaneity factor:

$$\varphi = \frac{1}{\sqrt{N-1}} + 0.17 = 0.37 \quad [4.127]$$

Once that simultaneity factor and the peak period are calculated and the peak period; the maximum flow which will be measured at the maximum consumption hour.

$$C_{max} = \frac{Q_p \cdot \varphi \cdot N}{3600} = \frac{0.75 \cdot Q_d \cdot \varphi \cdot N}{3600} = \frac{0.75 \cdot Q_d \cdot \varphi \cdot N}{3600} = 0.3 \frac{l}{s} \quad [4.128]$$

- $Q_d = 150\ l$ for sanitarian water at 60°C.

The preparation time is the time needed to heat the water needed in the peak period, assuming it as $t_p = 3\ h$.

With all the necessary data, the effective volume of storage tank can be calculated:

$$V_u = \frac{C_{max}}{\frac{1}{T} + \frac{1}{t_p \cdot f_m}} = \frac{0.3}{\frac{1}{9792} + \frac{1}{1800 \cdot 0.98}} = 448.41\ l \quad [4.129]$$

And therefore, the total volume will be:

$$V_T = \frac{V_u}{f_m} = 457.57\ l \quad [4.130]$$

This value is the minimum volume that the HSW accumulation tank can be.

The accumulator PS 500 from the Acro ISO manufacturer will be the chosen one for this purpose [T4.23]**[28]**.

Table 4.23: Accumulator

Accumulator	
Manufacturer	Acro ISO
Model	PS Series 500
Volume	500 l

4.3.2 Boiler

The supply boiler will be calculated from the volume of the accumulator. The HSW accumulation systems have important losses through the envelope of the storage tank and losses in the distribution network.

These losses must be taken into account in calculating the power of the boiler and some of them are:

- Availability losses are the losses produced because the deposit is permanently ready for use, although sometimes, the consumption does not exist:

$$P_a = \frac{0.55 \cdot V_T}{1000} = \frac{0.55 \cdot V_T}{1000} = 0.251\ W \quad [4.131]$$

– V_T is the total volume ($V_T = \frac{V_u}{f_m} = 457.57\ l$).

- Output power that will be supplied to the boiler in the peak period will be:

$$P_u = \frac{C_2 \cdot (T_{cons} - T_{supply}) \cdot C_{max} \cdot T}{T + t_p} = \frac{4.188 \frac{kJ}{kg \cdot K} \cdot (333 - 279)K \cdot 0.3 \frac{kg}{s} \cdot 9792s}{(9792 + 1800)s}$$
$$= 57.31\ kW \quad [4.132]$$

-Distribution losses are the losses due to the HSW distribution and return flow. They depend largely on the level of insulation of pipes although experimental data confirm that the losses can be calculated by the following equation:

$$P_r = 0.3 \cdot P_u = 17.2\ kW \quad [4.133]$$

The total power of the boiler, assuming that the solar energy system will not be operational, will be the sum of useful power plus the losses of the circuit

$$P = P_r + P_u + P_d = 57.31 + 0.251 + 17.02 = 74.75\ kW \quad [4.134]$$

The boiler is an element installed in the previous conventional energy system, so they can be reused and it is not necessary to add these new items. The boiler previous installed is the model ZWR420 from Junkers [T4.24][F4.25]:

Table 4.24: Boiler

Boiler	
Manufacturer	Junkers
Model	ZWR420
Maximum useful power	420 kW

Figure 4.25: Junkers ZWR420

4.4 System Summary

The system will consist of 14 solar collectors that can supply 64.8 % of the total sanitarian hot water demand. Thus, it can supply 538671 liters of hot water per year.

The system consists of three interconnected circuits: solar circuit, secondary circuit and support circuit. The connections are made by pipes.

Solar circuit

The solar circuit will be composed by the following elements [T4.25]:

Table 4.25: Solar circuit elements

Item	Units
Solar collector Salvador Escoda S.A. SOL 2800 selective	14
Collector bracket Salvador Escoda S.A. SOL 2800 selective	14
Sonnerkraft propylene glycol solution (40 l)	1
Centrifugal pump Grundfos 15-65	1
Expansion tank Zilmet Solar Plus 50	1

Secondary circuit

The secondary circuit will be composed by the following elements [T4.26]:

Table 4.26: Secondary circuit elements

Item	Units
Heat exchanger Alfa Laval M3-FM	1
HSW accumulator Acro ISO PS2500	1
Centrifugal pump Grundfos 15-65	1

Support circuit

And finally, the support circuit will be composed by the following elements [T4.27]:

Table 4.27: Support circuit elements

Item	Units
Boiler Junkers ZWR420	1
HSW accumulator Acro ISO PS500	1

In addition, there will be elements as valves, elbow pipes… along the three circuits.

5. ECONOMIC STUDY

The efficiency of renewable energy is increasing more and more; and sometimes it is over the traditional energy, mainly based on fossil fuel. One of the reasons for the profitability of renewable energies is the rising price of traditional energy sources due to the increasing cost of extraction and the depletion of these resources.

The installation of a collectors system provides a saving, in this facility, in natural gas. However, this saving occurs after an initial investment that is going to be calculated as following.

The initial investment for building a solar heat water facility is much higher than the necessary to install a traditional system as a collective or individual boiler. Therefore, it is necessary to calculate the payback period and the saving in consumption of natural gas or diesel for deciding if the facility is, in long term, a good option

Nowadays, these facilities are becoming more reliable and require not require much maintenance, they can work seamlessly over 20 years, so currently, they are a good option over other traditional sources.

The profitability of these facilities is linked to the efficiency getting from solar radiation; there is a close relation between the profitability of the facility and the solar fraction achieved with it. The higher the solar fraction, most of the energy required by consumers will be satisfied without using traditional sources and therefore, it would generate more savings (and lower environmental impact) and a quicker amortization.

5.1 Initial investment

The price of the different components of eacg circuit are shown in the following table [T5.1].

Table 5.1: Elements budget

Item	Units	Unit price (€)	Total price (€)
Primary circuit			
Solar collector Salvador Escoda S.A. SOL 2800 selective	14	595.00	8330
Collector bracket Salvador Escoda S.A. SOL 2800 selective	14	64.00	896
Sonnerkraft propylene glycol solution (40 l)	1	240	240
Centrifugal pump Grundfos 15-65	1	350	350
Expansion tank Zilmet Solar Plus 50	1	150	150
Secondary circuit			
Heat exchanger Alfa Laval M3-FM	1	601	601
HSW accumulator Acro ISO PS2500	1	2000	2000
Centrifugal pump Grundfos 15-65	1	350	350
Support circuit			
HSW accumulator Acro ISO PS500	1	500	500
Other items (valves, pipes…)			400
		Total price	13817

Moreover, there will be costs due to the installation assembly, installation and labor.

These costs will be around 30 % of the elements cost; so, the total initial investment will be:

$$\begin{aligned} \textit{Initial investment} &= \textit{Elements cost} + \textit{Labor cost} \\ &= \textit{Elements cost} + 0.3 \cdot \textit{Elements cost} = 17962.1€ \quad [5.1] \end{aligned}$$

5.2 Payback Period

The amortization period is the time it would take to recover the initial investment with the traditional energy savings.

For this purpose, the following information is required [T5.2]:

Table 5.2: Energy saved

	Days/month	Q_{daily}(MJ/day)	$Q_{monthly}$(MJ/month)	Solar fraction	Solar energy (kWh)
January	31	575.83	17851	36.8	1824.8
February	28	549.96	15399.12	48.7	2083.2
March	31	502.38	15574.04	57.9	2504.8
April	30	510.8	15324.14	68.8	2928.6
May	31	477.42	14800.23	71.5	2939.5
June	30	453.99	13619.87	80	3026.6
July	31	395.94	12274.23	96.6	3293.6
August	31	352.85	10938.59	95	2886.6
September	30	415.45	12463.6	79.3	2745.4
October	31	440.51	13655.83	62.7	2378.4
November	30	478	14340.02	43.3	1724.8
December	31	555.18	17210.61	35.3	1687.6

And the total energy obtained by solar energy will be:

$$\text{Solar energy} = 30023.9\ \text{kWh} \qquad [5.2]$$

When calculating the savings of the system (S), it is necessary to consider the energy savings produced by the facility, which is equal to the solar fraction, so the energy saving will be kW/h.

This value will be the most important to calculate the NPV (Net Present Value), which will be calculated as follows:

$$S = c \cdot Q_{saving} \qquad [5.3]$$

- Q_{saving} is the energy saved during a year.
- c is the heat prize.

$$c = c' \cdot \frac{1}{\eta \cdot \frac{LHV}{HHV}} \qquad [5.4]$$

- c' is the fuel unit cost.
- $\frac{LHV}{HHV}$ is the ratio between the lower heating value and the higher heating value.
- η is the energy conversion efficiency.

The price of the energy c´ (natural gas) will be 0.04181407 $\frac{€}{kWh}$ **[29]**, the calorific value is around 0.9 for most common fuels, and the efficiency of the facility will be 85%. Therefore, the cost of heat will be:

$$c = 0.05465 \frac{€}{kWh} \; [5.5]$$

And the annual saving will be:

$$Annual\ saving = S = 1640.8 \frac{€}{year} \qquad [5.6]$$

Then, it is necessary to evaluate the investment with the method of the Net Present Value (NPV) that provides a financial valuation in the current context of net cash provided by investment; it is the ratio between the investment incomes and expenses depending on the time.

$$NPV = -I_o + \sum_n \frac{B_n}{(1+i)^n} = -I_o + \frac{S_1}{(1+i)} + \frac{S_2}{(1+i)^2} + \ldots + \frac{S_n}{(1+i)^n} \qquad [5.7]$$

- I_o is the initial investment ($I_o = 17962.1€$).
- S is the saving per year ($S = 1640.8\ €/year$).
- i is the rate of investment return ($i = 2.5\ \%$).
- n is the number of years.

If the investment is profitable, the value of the NPV must be positive [T5.3]. To know the number of years needed to amortize the investment would be sufficient to equal NPV to zero and calculate n. The calculation will be done iteratively until the expression used is zero or positive.

Table 5.3: Net Present Value (NPV)

Year	Benefit (€)	Year	Benefit (€)
0	-17962.1	7	-7544
1	-16361.3	8	-6197.3
2	-14799.5	9	-4883.5
3	-13275.93	10	-3601.7
4	-11789.45	11	-2351.1
5	-10339.22	12	-1131.1
6	-8924.36	13	59.1

Therefore, the facility will be amortized after 13 year, as can be seen in the table below [F5.1].

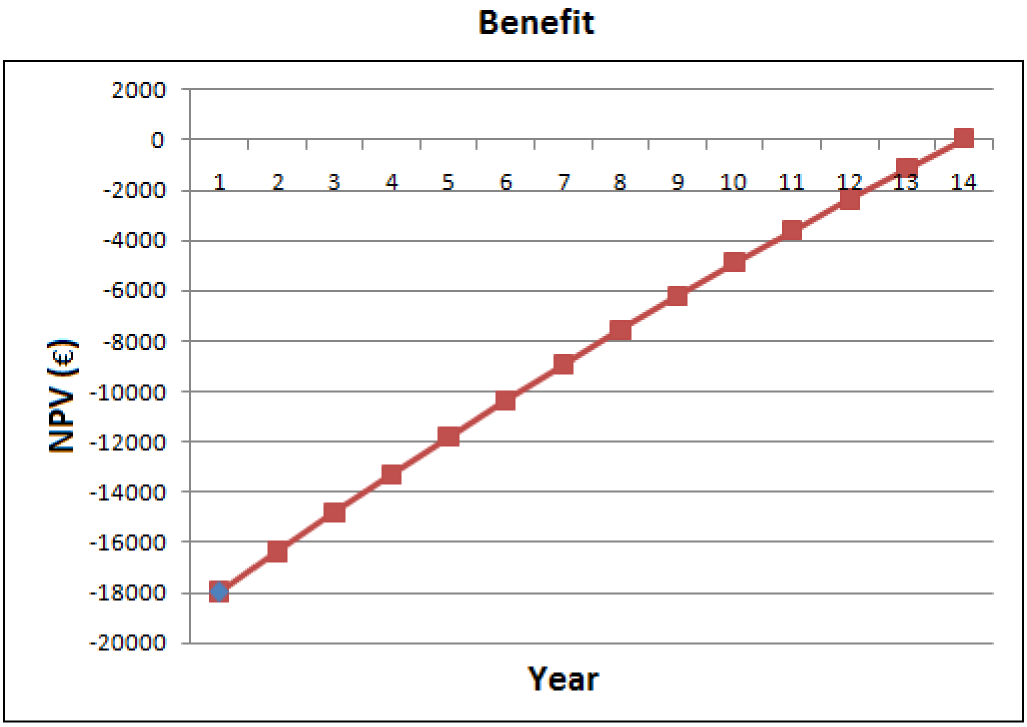


Figure 5.1: NPV evolution

On the other hand, according to the legislation, when a solar energy system is installed, it is possible to apply to subsidies that largely recovered the initial investment. These subsidies have not been taken into account in previous calculations. However, as the half-life of the collectors is 20 years, this facility will be profitable even without subsidies.

6. AMBIENTAL STUDY

The use of renewable energy is not only a source of economic savings, but also a source of emissions and pollution savings. During last decades, lifestyle is guided to more efficient and less damaging consumption due to the current world situation, in which government policies have realized the damage that has been causing the planet and climate.

However, it is necessary to take into account the contamination produced in the manufacture of all components, especially in the management and handling of heavy metals and fluids such as antifreeze, although this field should be controlled by the environmental regulations.

Another aspect linked directly to the facility in the building is the visual impact is, so it is necessary to care for and find the best conditions when installing collectors on buildings.

The reduction of CO_2 emission is the main point in the environmental effectiveness of these systems

But is the reduction of CO_2 that really supports these technologies and making them really cost-effective from an environmental standpoint. The main consequence of the implementation of renewable energy is to reduce the effects on the atmosphere such the greenhouse effect, that is the phenomenon whereby certain gases, which are components of a planetary atmosphere, retain some of the energy emitted by the ground that have been heated by solar radiation.

Nowadays, the greenhouse effect is being pointed at the Earth by increasing the concentration increasing of so-called greenhouse gases: water vapor (H_2O), carbon dioxide (CO_2), methane (CH_4), nitrogen oxides (CO_x), chlorofluorocarbons ($CFCl_3$) and ozone in the troposphere. Solar radiation passes through them, but they trap and retain heat from infrared radiation reflected from the ground surface, increasing the temperature of the atmosphere. The CO_2emissions cause about 30% of the effect.

6.1 Fuel Saving

The solar installation will allow a reduction in the boiler fuel consumption. In our facility, the fuel used to heat the sanitarian water will be Natural Gas. The saving fuel will be calculated as following:

$$Q = V \cdot LHV \cdot \eta_{boiler} \qquad [6.1]$$

- Q is the energy provided by the solar collectors (MJ).
- V is the fuel volume needed to produce the energy (m^3).
- LHV is the lower heating value ($LHV = 39.6\ \frac{MJ}{N \cdot m^3}$).
- η_{boiler} is the boiler efficiency ($\eta_{boiler} = 93\ \%$)

The only unknown value is the fuel consumption that will be:

$$V(m^3) = \frac{Q}{LHV \cdot \eta_{boiler}} \qquad [6.2]$$

And the fuel savings will be shown in the following table [T6.1]:

Table 6.1: Fuel saved

	$Q_{monthly}$ (KJ)	SF	Solar energy (MJ)	Fuel saved (m^3)		$Q_{monthly}$ (KJ)	SF	Solar energy (MJ)	Fuel saved (m^3)
Jan	17851	36.8	6569.2	178.4	July	12274.23	96.6	11856.9	322
Feb	15399.12	48.7	7499.4	203.6	Aug	10938.59	95	10391.7	282.2
March	15574.04	57.9	9017.4	244.9	Sep	12463.6	79.3	9883.6	268.4
April	15324.14	68.8	10543	286.3	Oct	13655.83	62.7	8562.2	232.5
May	14800.23	71.5	10582.2	287.3	Nov	14340.02	43.3	6209.2	168.6
June	13619.87	80	10895.9	295.9	Dec	17210.61	35.3	6075.3	165

Therefore, the annual fuel savings will be 2934.9 m^3.

6.2 CO_2 Emissions Avoided

With the use of solar energy, the installation will stop issuing large amount of CO_2 into the atmosphere. To know how much CO_2 will be stopped emitting, it is necessary to know the composition of natural gas [T6.2]:

Table 6.2: Gas Natural components

Hydrocarbon	Chemical Composition	%
Methane	CH_4	91.4
Ethane	C_2H_6	7.2
Propane	C_3H_8	0.8
Nitrogen	N_2	0.6

The natural gas combustion will be considered as a stoichiometric reaction, and the only products formed will be CO_2, H_2O and N_2.

$$C_xH_y + \left(x + \frac{y}{4}\right)(O_2 + 3.76N_2) \rightarrow xCO_2 + \frac{y}{2}H_2O + 3.76\left(x + \frac{y}{4}\right)N_2 \qquad [6.3]$$

- x and y are number of moles.

As the natural gas composition is expressed in % of volume, it is necessary to convert it into moles using the ideal gas state equation:

$$P \cdot V = n \cdot R \cdot T \qquad [6.4]$$

- P is the pressure (in normal conditions, $P = 101325\ Pa = 1\ atm$).
- V is the gas volume (the volume supposed is $1m^3$).
- n is the number of moles.
- R is the ideal gas constant ($R = 8.314\ \frac{J}{mol \cdot k}$).
- T is the temperature (in normal conditions, $T = 298\ K$).

And the number of moles is:

$$n = 40.89\,mol \qquad [6.5]$$

When the number of moles in 1 $m^3$of gas is known, this value is multiplied by the corresponding percentage of the natural gas components because the volume percentage is the same than the moles percentage [T6.3]**[29]**.

Table 6.3: Number of moles

Hydrocarbon	Chemical Composition	Moles
Methane	CH_4	37.37
Ethane	C_2H_6	2.94
Propane	C_3H_8	0.327
Nitrogen	N_2	0.245

And using balance of carbon, hydrogen, oxygen and nitrogen balance, the number of CO_2 moles will be:

$$37.37CH_4 + 2.94C_2H_6 + 0.327C_3H_8 + 0.245N_2 + 86.66(O_2 + 3.76N_2) \rightarrow 44.23CO_2 + 84.87H_2O + 326.1\,N_2 \qquad [6.6]$$

So, there will be 44.23 moles of CO_2 per cubic meter of natural gas.

To know the corresponding mass to these moles, the following expression will be used:

$$n = \frac{m}{Pm} \qquad [6.7]$$

- n is the number of moles.
- m is the mass (g).
- Pm is the CO_2 molecular weight ($Pm = 44$).

Therefore, the CO_2 mass will be $m = 1.96\,kg$.

To calculate the CO2 emissions avoided, the amount of CO2 saved [T6.4], which has been previously calculated, will be multiplied by the mass:

Table 6.4: CO_2 emissions avoided

	Fuel saved (m^3)	Avoided emissions (kg)
January	178.4	347.2
February	203.6	396.2
March	244.9	476.6
April	286.3	557.1
May	287.3	559.1
June	295.9	575.8
July	322	626.6
August	282.2	549.2
September	268.4	522.3
October	232.5	452.4
November	168.6	328.1
December	165	321.1
Yearly	2934.6	5710.7

The amount of CO_2 that will not be emitted when using this type of solar heating system, will be $5710.7\ kg$.

7. CONCLUSION

During this project, it has been studied the hot water supply in a building of 26 homes, where are estimated to live 104 persons, with a solar energy system.

The study was conducted based on the energy required to heat the water needed, and after choosing the orientation (south), inclination (50°) and type of collector more suitable for the installation, is dimensioned to choose the number of collectors necessary taking into account losses that occur in the system **[3]**. Then, the remaining elements will be calculated to meet the needs of the system. These calculations will be made following the patterns of consumption and energy demand in the current legislation **[16]**.

The system will consist on 14 flat solar collectors, a heat exchanger to transmit the heat obtained from the heat exchange fluid to the hot sanitarian water and two accumulators; one of the to store the water from the heat exchanger and the second one the water is stored before the consumption.

The solar water heating has been developed during last decades and nowadays, in many climates such southern European climate, a solar hot water system can provide up to 85% of domestic hot water energy and in northern European countries, combined hot water and space heating systems are used to provide 15 to 25% of home heating energy.

In this case, solar energy system provides 64.8 % of the thermal needs for the hot water supply ($173783.2\ \mathrm{MJ}$ per year). Thus, it fulfills the regulation because, as the building is located in Madrid (climatic region IV), the minimum solar region should be 60%. There is no excess of solar thermal energy during any month; therefore, the system could not be used in a future swimming pool, air conditioning or underfloor heating.

One of the most notable points when using this type of facility is the great environmental savings as have been studied during the environmental impact point; with this system, the amount of $\mathrm{CO_2}$ that will not be emitted into the atmosphere will be $5710.7\ kg$.

During last years, the quality and efficiency of the equipment needed for this type of facility has increased significantly. However, its greatest handicap is still its high cost ($17962.1\,€$ including both, elements and labor costs) what does that sometimes these facilities are not profitable without subsidies. In this case, the facility without subsidies will be amortized in 13 years.

Finally, as the technical building code mentions **[14]**, it is necessary to set a balance between the system and the architecture of the buildings in order to avoid a visual impact and to improve the facility placing the elements in the best location and avoiding shadow. This point is achieved by placing the installation in the roof of the building, where it does not produce any reflection or a shadow in nearby buildings.

Therefore, in view of the results and as mentioned above, this kind of system has achieved an efficiency and cost that allow its use become popular in countries like Spain or Italy where the sun can provide the energy required.

The awareness of society on environmental problems that cause traditional energies, as well as the potential shortage of such energy, make use of systems such as studied in this work one of the best alternatives for the near future.

REFERENCES

[1] A. E. Dixon, J. D. Leslie “Solar energy conversión: an introductory course”, 1978.

[2] M. Conde Engineering “Thermophysical properties of brines”, 2002.

[3] M. C. Rodríguez, R. Salgado “Transferencia de calor en colectores solares palnos, considerando velocidad y dirección del viento” (“Heat transfer in solar collectors paln, considering wind speed and direction”), 2008.

[4] Frank P. Incropera, David P. Dewitt “Fundamentals of Heat and Mass Transfer”, 2001.

[5] J. Richard Williams “Solar energy: technology and application”, 1977.

[6] John A. Duffe, William A. Beckman “Solar energy thermal processes” 1974.

[7] Martin Kaltschmitt “Renewable Energy: Technology, Economics and Environment”, 2007.

[8] Martin Kaltschmitt, Wolfgang Streicher, Andreas Weiser “Solar thermal heat uitlisation”, 2002.

[9] Pilar Pereda Sequet “Proyecto y cálculo de instalaciones solares térmicas” (“Design and calculation of solar thermal facilities”), 2006.

[10] N. S. W. Williamton “Solar radiation inciden ton inclined surface”, 1978.

[11] M. D. Steven “Angular distribution and interception of diffuse radiation”, 1977.

[12] Frank Akristal, F. A. Annett “Pumps: Types, selection, installation, operation and maintenance”, 1973.

[13] Engineering equipment users association “Thermal insulation of pipes and vessels”, 1963.

[14] “Código Técnico de la Edificación (CTE)” (“Technical building code”)

[15] "Ordenanza sobre captación de energía solar para usos térmicos. Ordenanza solar de Madrid (mayo del 2003)" ("Ordinance about solar energy capture for thermal use. Madrid Municipal Ordinance (May 2003)").

[16] Instituto para la Diversificación y Ahorro de Energía (Institute for Energy Diversification and Saving) "Pliego Oficial de Condiciones Técnicas de Instalaciones de Energía Solar Térmica de Baja Temperatura" ("Official Statement of Technical Low Temperature Solar Power Facilities").

[17] "Reglamento de Instalaciones Térmicas en los Edificios (RITE)" ("Regulation of Thermal Installations in Buildings").

[18] Asociación Técnica Española de Climatización y Refrigeración Documento (Spanish Technical Association of Air Conditioning and Refrigeration) "Preparación de agua caliente para usos sanitarios" ("Preparation of hot water for sanitary purposes").

[19] Asociación Técnica Española de Climatización y Refrigeración Documento (Spanish Technical Association of Air Conditioning and Refrigeration) "Instalaciones Solares Térmicas para Producción de Agua Caliente Sanitaria" ("Facilities Solar Thermal Facilitiea for Hot Water Production").

[20] Thermotechnics, Faculty of Industrial Engineering (Technologic University of Madrid) "Class notes".

[21] CENSOLAR (Centro de estudios de la energía solar, Solar Energy Training center).

[22] Kim Jeongbae, Dong Won Lee, Hee Youl Kwak "Analytical study on the termal characteristics of a flat-plate collector and evacuated solar collector", 2006.

[23] Salvador Escoda S.A webpage (http://www.salvadorescoda.com) "Product Catalog 2009", 2009

[24] Salvador Escoda S.A webpage (http://www.salvadorescoda.com) "Price List Catalog 2009", 2009

[25] Grundfos webpage (http://www.grundfos.com) "Product Catalog 2010", 2010

[26] Zilmet webpage (http://www.zilmet.com), “Product Catalog 2011”, 2011

[27] Alfa Laval webpage (http://www.alfalaval.com), “Product Catalog 2011”, 2011

[28] Acro ISO webpage (http://www.acroiso.com), “Product Catalog 2009”, 2009

[29] Junkers webpage (http://www.junkers.com)

[30] Sonnerkraft (http://www.sonnenkraft.es), “Product Catalog 2011”, 2011

[31] Gas Natural webpage (http://www.gasnatural.es)

[32] Maple webpage (http://www.maplesoft.com), “User Manual, Maple 13”, 2009

APPENDIX A: RADIATION ON AN INCLINED SURFACE

Before calculating the incident solar radiation on a surface with some inclination and orientation, it is necessary to define some terms [FA.1][FA.2][FA.3][FA.4]**[9][10]**:

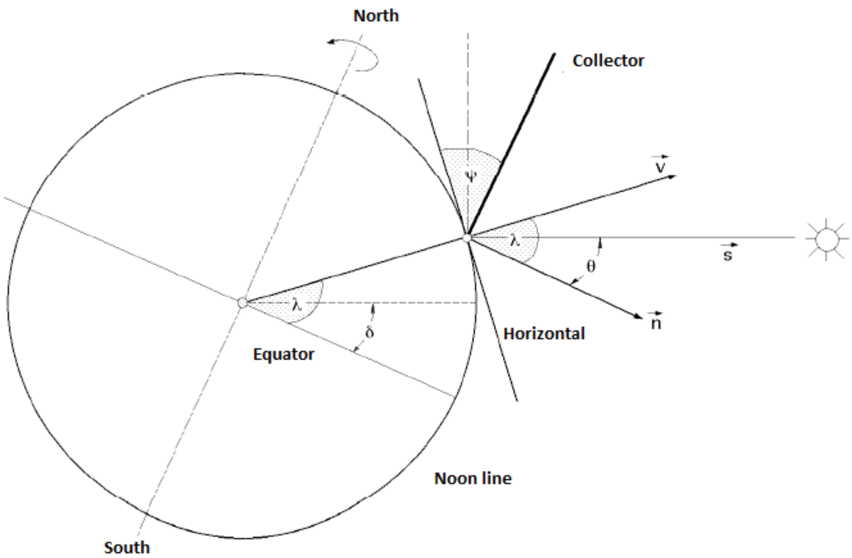


Figure A.0.1: Spatial location of different angles (I)

- Azimuth angle (β_s) is the angle between the horizontal component of direct solar radiation, with the North-South direction in the northern hemisphere. It is measured on the horizontal plane that passes through the place.
- Zenith angle(θ_s) is the angle between the direct solar radiation and the vertical plane, at any point on Earth. This angle varies continuously during the day and depends on the latitude of the place and declination.
- Declination δ is the angular position of the sun at solar noon, the plane of the Earth Ecuador, the value of this angle is usually taken every day at solar noon due to the axis of rotation of the Earth is at an angle of 23 ° 45 ' with the axis of the plane that contains the orbit around the Sun and hence the value of the decline varies between ±23 ° 45´ during the year.

- Hour angle τ is the angular displacement of the Sun, about noon, which is the moment when the sun is higher in the sky and corresponds to a minimum solar zenith angle. Each hour is equal to 15 ° longitude, taking the value (+) in the morning and (-) in the afternoon.

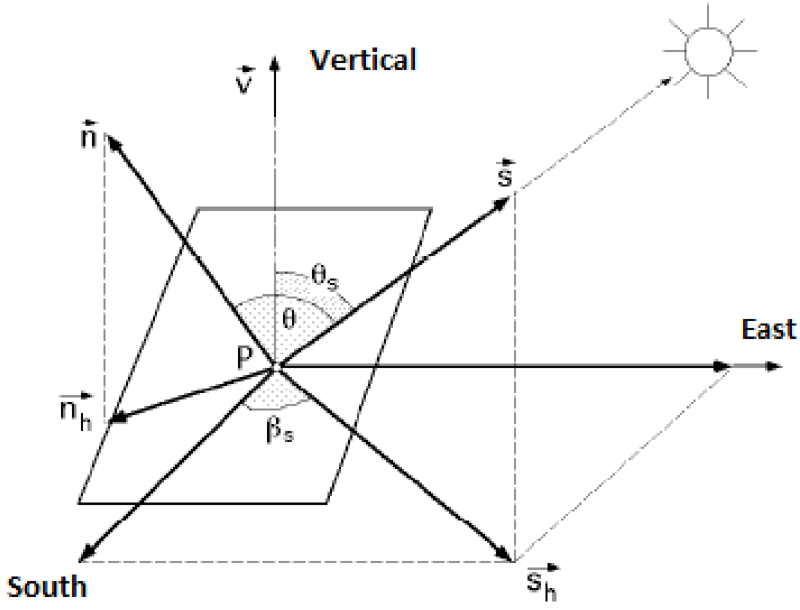


Figure A.0.2: Spatial location of different angles (II)

- The latitude λ is the angular displacement above or below the Ecuador, measured from the center of the Earth and taking the value (+) in the North hemisphere.
- The longitude L is the angle between the plane that passes through the poles in any place of the Earth's surface and the plane that passes through Greenwich meridian. The longitude and latitude are the coordinates that locate any point on Earth's surface.
- -Another angle that must be taken into account is the angle between the ground and the horizontal of the place. This angle is called ψ, and is the zenith angle of the surface and, therefore, the angle between the normal to the surface directly above the site.

Below are a series of trigonometric relationships between these angles to facilitate subsequent calculations[FA.3].

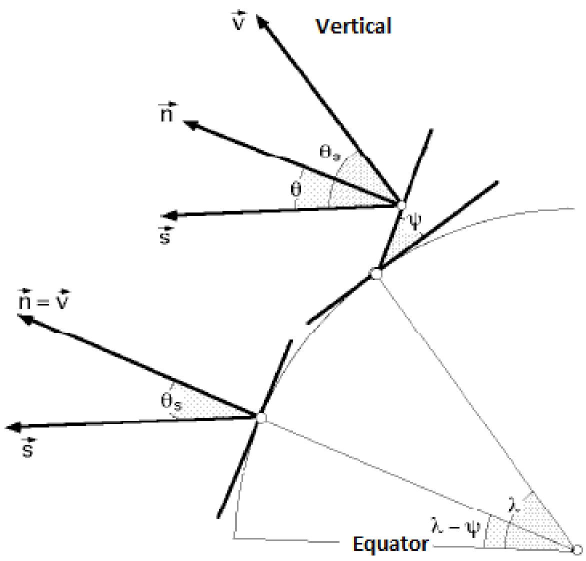


Figure A-3: Spatial location of different angles (III)

$$cos\theta = sin\delta\ sin\lambda \cos\psi - \sin\delta \cos\lambda \sin\psi \cos\beta + \cos\delta \cos\lambda \cos\psi \cos\tau + \cos\delta\ sin\lambda \sin\psi \cos\beta \cos\tau + \cos\delta \sin\psi \sin\beta \sin\tau \qquad [A.1]$$

$$\cos\theta_s = \sin\delta\ sin\lambda + \cos\delta \cos\lambda \cos\tau \qquad [A.2]$$

$$cotan\beta_s = \frac{tan\delta \cos\lambda - sin\lambda \cos\tau}{\sin\tau} \qquad [A.3]$$

Firstly, on a horizontal surface [FA.4], the direction of the direct radiation (I_0) is at angle θs to the vertical and its projection over the vertical is $I_{0(h)}$. The normal of the inclined surface is at angle θ_n to the horizontal plane and the projection of I_0 over the normal to the surface is I_N. Usually, it is assumed that in the Northern hemisphere the surface is oriented to the south to maximize the use of solar radiation.

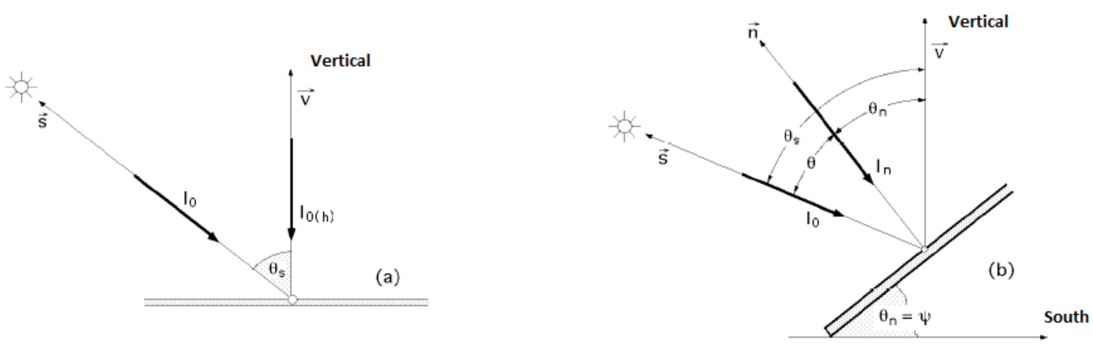


Figure A-4: Spatial location of different angles (IV)

$$I_{O(h)} = I_O cos\theta_s \qquad [A.4]$$

$$I_N = I_O cos(\theta_s - \theta_n) = I_O cos\theta_{s-n} \qquad [A.5]$$

The relationship η_D between the radiation on the inclined surface I_N and the radiation on the horizontal surface $I_{0(h)}$ is:

$$\eta_D = \frac{I_N}{I_{O(h)}} = \frac{cos\theta_{s-n}}{cos\theta_s} = \frac{\sin\delta\ sin\,(\lambda - \theta_n) + cos\delta\ cos(\lambda - \theta_n)cos\tau}{\sin\delta\ sin\lambda + cos\delta\ cos\tau cos\lambda}$$

$$= cos\theta_n + sin\theta_n tan\theta_s \qquad [A.6]$$

- δ is the declination or angular position of the sun at solar noon compared to the equatorial plane.
- λ is the latitude of the place
- τ is the hour angle.

On the other hand, when the diffuse component of solar radiation is spread uniformly across the sky (uniform cloudiness), this component on an inclined surface will depend on the part of sky seen from this surface [FA.5]. It is assumed that the ground and other surfaces reflect the solar radiation, so they are a new radiation source and they are equivalent to the sky and the inclined surface will receive the same diffuse radiation without depending on its orientation; the correction factor of diffuse radiation is always unity.

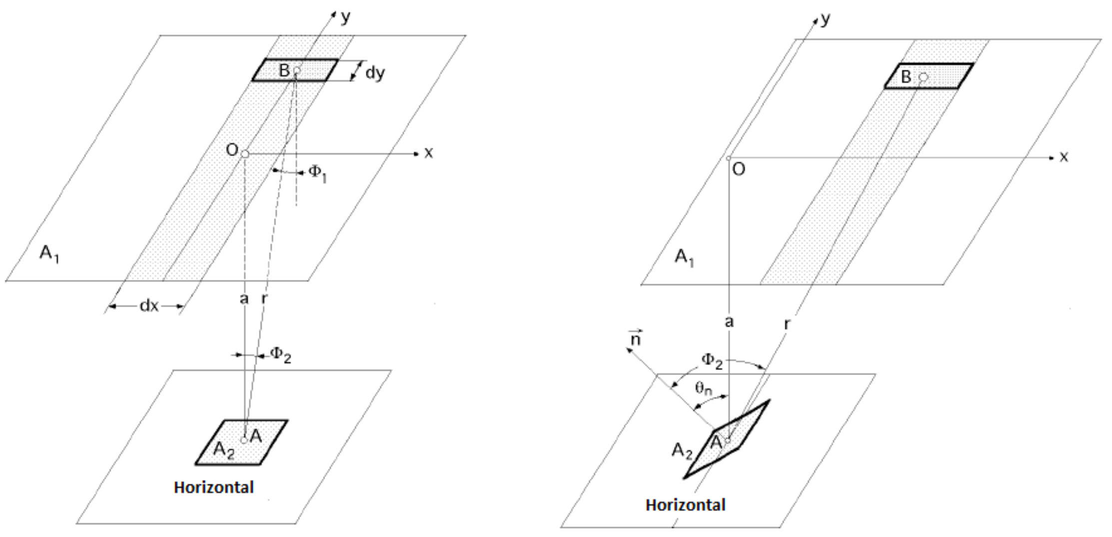


Figure A-5: Direct solar radiation on horizontal and inclined surfaces

In these conditions, the diffuse radiation received on an inclined surface with an angle θ_n and the sky with uniform cloudiness can be represented by a horizontal plane A_1 that radiates diffusely on θ_n oblique plane that contain the surface A_2 Origin (O) and axis (x, y) will be chosen on the plane A_1. This plane can be divided into uniform rectangular bands with a width dx and the influence that each band does on a horizontal portion of the surface A_2 that is at a distance (a) from the plane A_1. The band considered is decomposed into infinitesimal strips of width dy, if the strip in B has an area of dA_1 and the distance from B to A is r (BA = r), then the diffused radiation that reaches A_2 from dA_1 is:

$$dI_{dif(dA_1 \to A_2)} = \frac{\cos\phi_1 \cos\phi_2 A_2 dA_1}{r^2} = \left| \begin{array}{c} \phi_1 = \phi_2 \\ r^2 = a^2 + y^2 \\ dA_1 = dx\,dy \end{array} \right| = \frac{\cos^2\phi\, dx\, dy\, A_2}{a^2 + y^2} \quad [A.7]$$

And the diffuse radiation from the band dx to A_2:

$$I_{dif(dx \to A_2)} = \int \frac{\cos^2\phi\, dx\, A_2}{a^2 + y^2} dy = \left| \begin{array}{c} \tan\phi = \frac{y}{a} \\ dy = \frac{a\, d\phi}{\cos^2\phi} \end{array} \right| = \int_{\frac{-\pi}{2}}^{\frac{\pi}{2}} \frac{a\, dx\, A_2}{a^2(1 + \tan^2\phi)} d\phi$$

$$= \frac{\pi A_2}{2a} dx \quad [A.8]$$

Then it is assumed that O and A are not in the same vertical and the diffused radiation from dx to A_2 is:

$$I_{dif(dx \to A_2)} = \frac{\pi A_2}{2r} \cos^2\theta dx \quad [A.9]$$

And if the plane is inclined at an angle A_2 θ_n:

$$I_{dif(dx \to A_2)} = \frac{\pi A_2}{2r} \cos\theta_1 \cos\theta_2 dx = \left| \begin{array}{c} \theta_1 = \theta + \theta_n \\ dx\cos\theta_1 = r d\theta \end{array} \right| = \frac{\pi A_2}{2} \cos(\theta + \theta_n) d\theta \quad [A.10]$$

$$dI_{dif(dA_1 \to A_2)} = \frac{\pi A_2}{2} \int_{\frac{\pi}{2}}^{\frac{\pi}{2} - \theta_n} \cos(\theta + \theta_n)\, d\theta = \frac{\pi A_2}{2} (1 + \cos\theta_n) \quad [A.11]$$

It is the diffused radiation from the sky to a inclined surface at an angle θ_n [FA.6].

If $\theta_n = 0$ (horizontal surface), the diffuse radiation from the sky to the surface A_2 will be:

$$I_{dif(h)} = \pi A_2 \qquad [A.12]$$

$$I_{dif(sky \to A_2)} = I_{dif(h)} \frac{1 + \cos\theta_n}{2} \qquad [A.13]$$

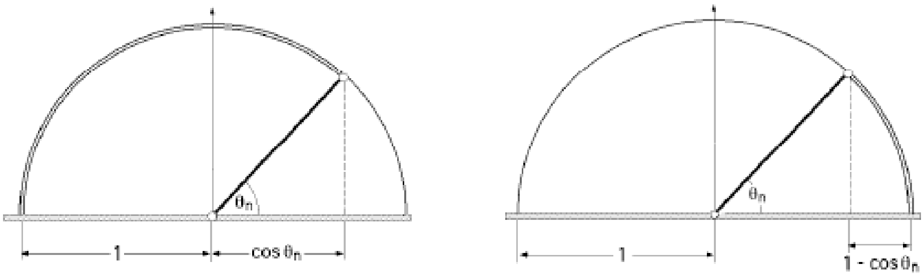


Figure A-6: Model to determine the total, diffuse and albedo radiation

So, a horizontal surface jus receive the radiation from “half” sky.

In practice the diffuse radiation on a horizontal plane can be consider as the third of the direct radiation on the same plane.

Using considerations similar to those of the diffuse radiation can be assumed that the plane A_2 also pick the radiation reflected by the ground due to the inclined surface sees a portion of it or its surroundings. If the reflection coefficient for diffuse radiation in the environment is ρ, the radiation reflected by the ground on the inclined surface, from the total solar radiation, is the $(I_0 + I_{diffused})_h$, multiplied by the ground´s albedo [TA.1] that is is the fraction of incident energy diffused by a luminous body.

$$I_{albedo(A_2)} = (I_0 + I_{dif})_h \frac{1 - \cos\theta_n}{2} \rho \qquad [A.14]$$

- $\frac{1-\cos\theta_n}{2}$ is the albedo

Table A.0.1: Albedo values

Surface	Albedo	Surface	Albedo
Black	0	Clean ground	0.15-0.25
New snow	0.9	Meadow	0.25-0.75
Old snow	0.6	Grass	0.26

Surface	Albedo	Surface	Albedo
Ground with snow	0.7	Ground with leaves	0.3
Ground without snow	0.2	Sea	0.05 (summer) 0.1 (winter)
Wet ground	0.17	Sand	0.4

The diffuse component with the two effects is:

$$I_{dif} = I_{dif(h)}\frac{1+\cos\theta_n}{2} + (I_0 + I_{dif})_h\frac{1-\cos\theta_n}{2}\rho \quad [A.15]$$

So the total radiation on an inclined surface including the albedo is:

$$I_{tot} = I_{0(\theta_n)} + I_{dif} = \eta_D I_{0(h)} + I_{dif}$$
$$= \eta_D I_{0(h)} + I_{dif(h)}\frac{1+\cos\theta_n}{2} + (I_0 + I_{dif})_h\frac{1-\cos\theta_n}{2}\rho \quad [A.16]$$

The effective factor of solar energy in the surface inclined from the horizontal total is:

$$\eta_{ef} = \frac{I_{0(h)}}{I_{tot(h)}}\eta_D + \frac{I_{dif(h)}}{I_{tot(h)}}\frac{1+\cos\theta_n}{2} + \frac{1-\cos\theta_n}{2}\rho \quad [A.17]$$

So the greater θ_n, the greater albedo influence is.

None of these approaches is very satisfactory; a solar collector provides the largest fraction of its total energy input during periods of great radiation time, so it is valid when working with collectors operating at high temperatures; but not, normally, during long periods of cloudiness. Diffuse reflected radiation diffuses into the atmosphere and it rediffused it partially to the ground; for a medium cloudiness, the rediffusion is around 10% and the total radiation increases in the same way.

APPENDIX B: CALCULATION OF THE SYSTEM WITH LOSSES DUE TO WIND

There are two types of losses to be considered in the solar energy system: forced convection losses due to wind on the items located on the roof (collectors and pipes) and natural convection losses in the section from the roof to the room where the accumulator will be located, but these losses will be negligible **[9][3]**.

As in previous calculations, the calculations to know the unknown values will be done solving a equations system of 9 equations with 9 unknown values, and they will be calculated both monthly and yearly.

The collection area and the heat exchanger fluid in the primary circuit and the water fluid in the secondary will be the calculated in the previous point.

$$A = 38.92\, m^2 \quad [B.1]$$

$$m_1 = m_2 = 0.332 \frac{kg}{s} \quad [B.2]$$

LOSSES IN THE COLLECTOR

Before calculating the losses in the collector due to the wind, it is necessary to know its velocity [TB.1][**21**].

Table B.0.1: Wind velocity

	Jan	Feb	Mar	Apr	May	Jun
v (m/s)	2.56	3.91	4.67	3.54	3.72	3.32
	Jul	*Aug*	Sep	Oct	Nov	Dec
v (m/s)	3.18	3.78	2.52	2	2	1.99

And the annual average velocity is:

$$v_{yearly} = 3.1\ \frac{m}{s} \qquad [B.3]$$

The total thermal resistance produced by the collector due to wind will be calculated by an equivalent thermal diagram, showing all the resistances of conduction and convection resulting in the transfer of heat produced.

Thermal contact resistances have been neglected, for simplicity in calculations, and because their value is very small and can be neglected without altering any numerical result. Here [FB.1], there is a scheme of the section of a solar panel and successive thermal resistances that are consider:

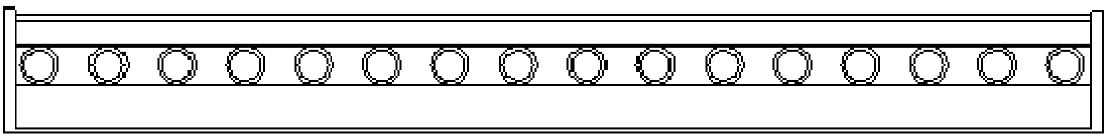


Figure B.0.1: Section of a solar collector

There are 6 thermal resistances from inside the pipe to the ambient [FB.2]:

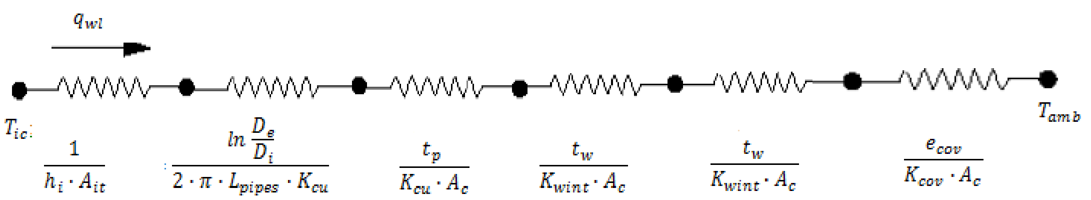


Figure B.0.2: Thermal resistances (I)

- Q_{WL} is the heat that the fluid within the coil loses for the environment.
- $R_{conv,ip} = \frac{1}{h_i \cdot A_{ip}}$ is the convection thermal resistance inside the coil.
- $R_{pipe} = \frac{ln\frac{De}{D_i}}{2 \cdot \pi \cdot L_{pipes} \cdot k_{cu}}$ is the pipe thermal resistance.
- $R_w = \frac{t_p}{k_{cu} \cdot A_c}$ is the paint thermal resistance.
- $R_{cov} = \frac{t_w}{k_{wint} \cdot A_c}$ is the thermal resistance in the space between the absorbers and the cover.
- $R_w = \frac{e_{cov}}{k_{cov} \cdot A_c}$ is the cover thermal resistance.

- $R_o = \frac{1}{h_o \cdot A_c}$ is the convection thermal resistance on the panel surface.
- T_{amb} is the ambient temperature.
- T_{ic} is the average temperature within the coil.

Convection within the coil.

Firstly, the Reynolds number of the fluid flowing inside the coil is going to be calculated to determine the flow regime in order to choose the most appropriate calculation method **[4]**:

$$Re = \frac{u \cdot D_i}{\upsilon} \quad [B.4]$$

- u is the fluid velocity.
- D_i is the internal diameter ($D_i = 7.5\ mm$).
- υ is the air cinematic viscosity ($2.805 \cdot 10^{-6}\ \frac{m^3}{s}$).

When calculating the fluid velocity, it is necessary to take into account the parallel connection of the pipes, so the flow through each is determined by dividing the total flow by the number of collectors.

$$u = \frac{4 \cdot \frac{m_1}{number\ of\ collectors}}{\pi \cdot \rho_1 \cdot {D_i}^2} \quad [B.5]$$

$$u = 0.525 \frac{m}{s} \quad [B.6]$$

Before proceeding, it is necessary to check whether the flow on the absorber tubes is laminar or turbulent, which is defined by the Reynolds number:

$$Re < 2300 \ \rightarrow Laminar\ flux. \quad [B.7]$$

$$Re > 2300 \ \rightarrow Turbulent\ flux. \quad [B.8]$$

After calculating the velocity and m_1, the Reynolds number is:

$$Re = 1403.74 < 2300 \ \rightarrow Laminar\ flux. \quad [B.9]$$

In addition the laminar flow, the flux is totally developed, so the following correlation, assuming a constant flow, can be applied:

$$Nu = 4.36 = \frac{h_i \cdot D_i}{k_1} \rightarrow h_i = 250.55\ \frac{W}{m^2 \cdot K} \quad [B.10]$$

- k_1 is the thermal conductivity of the heat exchange fluid ($0.431\ \frac{W}{m \cdot K}$).

Convection on the panel

The convection coefficient on the panels is going to be calculated as a parallel flow over flat plate, for which, all the collectors are going to be considered as a single surface. As previously, before proceeding, it is necessary to check whether the flow on the panels is laminar or turbulent, which is defined by the Reynolds number **[4]**:

$$Re < 5 \cdot 10^5 \ \rightarrow Laminar\ flux. \quad [B.11]$$

$$Re > 5 \cdot 10^5 \ \rightarrow Turbulent\ flux. \quad [B.12]$$

$$Re = \frac{\rho_{wind} \cdot v_{wind} \cdot L_c}{\mu_{wind}} \quad [B.13]$$

- ρ_{wind} is the air density ($\rho_{wind} = 1.184 \frac{kg}{m^3}$).
- v_{wind} is the average wind velocity (3.1 m/s).
- L_c is the collector characteristic length ($L_c = 1.206 \cdot 14 = 16.884$ m).
- μ_{wind} is the dynamic viscosity of the fluid ($\mu_{wind} = 1{,}82 \cdot 10^{-5}$ Pa · s).

$$Re = \frac{\rho_{wind} \cdot v_{wind} \cdot L_c}{\mu_{wind}} = 3.4 \cdot 10^6 \quad [B.14]$$

So, $Re > 5 \cdot 10^5$ and the flux will be turbulent in any point L in the collectors surface. It happens when $Re = 5 \cdot 10^5$ or when Reynolds number is critical.

$$Re_{critical} = \frac{\rho_{wind} \cdot v_{wind} \cdot L_c}{\mu_{wind}} \rightarrow L = \frac{\mu_{wind} \cdot Re_{critical}}{\rho_{wind} \cdot v_{wind}} = 2.479\ m \quad [B.15]$$

As the laminar flow is approximately in the 50% of the flow on the collectors, it cannot be neglected and therefore, the relationship between laminar heat transfer and Colburn analogy for turbulent flow will be applied:

$$Nu = \left[0.037 \cdot Re^{0.8} - (0.037 \cdot {Re_{critical}}^{0.8} - 0.664 \cdot {Re_{critical}}^{0.5})\right] \cdot Pr^{1/3}$$
$$= (0.037 \cdot Re^{0.8} - 871.32) \cdot Pr^{1/3} \quad [B.16]$$

- Pr is the Prandtl number.

$$Pr = \frac{\mu_{wind} \cdot C_{pwind}}{k_{wind}} \quad [B.17]$$

- k_{wind} is the air thermal conductivity and whose value is estimated in ($k_{wind} = 0.024 \frac{W}{m \cdot K}$).
- C_{pwind} is the wind specific heat ($C_{pwind} = 1007 \frac{J}{kg \cdot K}$).

And Prandtl number and Reynolds number must be:

$$5 \cdot 10^5 \leq Re \leq 10^7 \quad [B.18]$$

$$0.6 \leq Pr \leq 60 \quad [B.19]$$

$$Pr = 0.76 \quad [B.20]$$

So, Nusselt number will be:

$$Nu = 4875 \quad [B.21]$$

$$Nu = \frac{h_o \cdot L_c}{K_{wind}} \rightarrow h_o = \frac{k_{wind} \cdot Nu}{L_c} = 6.93 \ \frac{W}{m^2 \cdot K} \quad [B.22]$$

Conductance or heat transfer global coefficient.

The global heat transfer coefficient shows the lost power per square meter and degree Celsius of difference that exist between the fluid inside the coil temperature and the environment **[4]**. Equivalent resistance is the sum of each thermal resistance which influences the heat transfer from the fluid to the outside of the solar collector.

$$R_{eq} = R_{conv,ip} + R_{pipe} + R_p + R_w + R_{cov} + R_o \quad [B.23]$$

- $R_{conv,ip}$ is the convection resistance inside the pipe.
- R_{pipe} is the pipe thermal resistance.
- R_p is the paint thermal resistance

- R_w is the thermal resistance in the space between the absorbers and the cover.
- R_{cov} is the cover thermal resistance.
- R_o is the convection thermal resistance on the panel surface.

When considering the thermal resistances, the area of each element has been included in its definition; so, the equivalent resistance of the collector is the product of global heat transfer coefficient and the collector area.

$$U \cdot A = \frac{1}{R_{eq}} \quad [B.24]$$

So, firstly, the value of each resistance will be calculated, then the value of equivalent resistance and finally, the global heat transfer coefficient. For this purpose, the following data will be needed:

- $L_{pipes}$is the total length of all the pipes ($L_{pipes} = 14 \cdot 23 = 322\ m$).
- A_c is the total collection area ($A_c = 14 \cdot 2.56 = 35.84\ m^2$).
- A_{ip} is the internal area of the pipes ($A_{ip} = \pi \cdot D_i \cdot L_{pipes} = \pi \cdot 7.5 \cdot 10^{-3} \cdot 322 = 7.58\ m^2$).
- t_p is the paint thickness ($t_p = 0.0004\ m$).
- t_w is the thickness of the space between the absorbers and the cover ($t_w = 0.03\ m$).
- t_{cov} is the cover thickness ($t_c = 0.004\ m$).
- k_{cu} is the copper thermal conductivity ($k_{cu} = 401\ \frac{W}{m \cdot K}$).
- k_{cov} is the cover thermal conductivity ($k_{cov} = 0.4\ \frac{W}{m \cdot K}$).

And the resistance of each element will be [TB.2]:

- Convection resistance inside the pipe:

$$R_{conv,ip} = \frac{1}{h_i \cdot A_{it}} = 5.1 \cdot 10^{-4}\ \frac{K}{W} \quad [B.25]$$

- The pipe thermal resistance will be:

$$R_{pipe} = \frac{ln\frac{D_e}{D_i}}{2 \cdot \pi \cdot L_{pipes} \cdot k_{cu}} = 7.9 \cdot 10^{-8}\ \frac{K}{W} \quad [B.26]$$

- The paint thermal resistance will be:

$$R_p = \frac{t_p}{k_{cu} \cdot A_c} = 2.7 \cdot 10^{-8} \frac{K}{W} \quad [B.27]$$

- The thermal resistance in the space between the absorbers and the cover will be:

$$R_w = \frac{t_w}{k_{wint} \cdot A_c} = 0.0348 \frac{K}{W} \quad [B.28]$$

- The cover thermal resistance will be:

$$R_{cov} = \frac{e_{cov}}{k_{cov} \cdot A_c} = 2.8 \cdot 10^{-4} \frac{K}{W} \quad [B.29]$$

- The convection thermal resistance on the panel surface will be:

$$R_o = \frac{1}{h_o \cdot A_c} = 0.004 \frac{K}{W} \quad [B.30]$$

Table B.0.2: Thermal resistances (K /W)

$R_{conv,ip}$	R_{pipe}	R_p	R_w	R_{cov}	R_o
$5.1 \cdot 10^{-4}$	$7.9 \cdot 10^{-8}$	$2.7 \cdot 10^{-8}$	0.0348	$2.8 \cdot 10^{-4}$	0.004

So, the global equivalent resistance will be:

$$R_{eq} = 0.0396 \frac{K}{W} \quad [B.31]$$

And the insulation provided by each element over the total insulation will be [TB.3]:

Table B.0.3: Insulation provided by each element over the total insulation

$\frac{R_{conv,ip}}{R_{eq}}$	$\frac{R_{pipe}}{R_{eq}}$	$\frac{R_p}{R_{eq}}$	$\frac{R_w}{R_{eq}}$	$\frac{R_{cov}}{R_{eq}}$	$\frac{R_o}{R_{eq}}$
0.0128	0.0000019	0.000000681	0.88	0.007	0.1

There is a big thermal resistance to the external convection but the greater insulation element is the air space between the cover and the collector plate.

And finally, with all the thermal resistances, the global heat transfer coefficient can be calculated:

$$U \cdot A = \frac{1}{R_{eq}} = \frac{1}{0.0396} = 25.25\ \frac{W}{K} \quad [B.32]$$

All these values are taken in to account just the average data for a year but when calculating the monthly solar fraction it will be also necessary the monthly data.

LOSSES IN PIPELINES.

A pipe is a tube formed by tubes which allow the passage of liquids or gases. Pipes are the fundamental element of the installation of the water. The installation or distribution of water is just a group of piping connections with fitting elements. Pipes can be made from different materials, some of them are **[9]**:

- Lead: they are not used anymore but they are quite soft and they can be easily cut with hacksaws.
- Iron: they have replaced lead pipes, especially in hot water systems. They are quite hard and therefore difficult to handle but they can also cut with hacksaws.
- Copper: this is the most common material used today. They are strong and malleable. Can be cut with fine tooth saw or short-copper. These pipes must be surrounded by an insulating material to prevent corrosion and damage from the plaster or cement.
- Plastic: there are PVC with are different sizes and with many accessories and threads. It is relatively easy to cut them with saws.

The most proper material for the pipes of this installation will be copper, because comparing with iron, lead or PVC pipes, lead pipes do not suffer impairment, and they have a great resistance against heat, pressure and oxidation.

The height of the building from the basement, where the heater room, heat exchanger and accumulators will be installed, to the roof is 51 meters (2 basement floors of 4 meters each one, an entrance roof of 4 meters and 13 regular floors of 3 meter each one).

The pipes will be surrounded by a thermal insulator that is a material used in construction characterized by high thermal resistance. It establishes a barrier to the heat transfer between two different materials that are at different temperatures **[13] [4]**.

Due to their low thermal conductivity and low radiation absorption coefficient, air and some other gases are the materials most resistant to heat transmission. However, on the airs chambers there is a convection phenomena that increases heat transfer capability; for this reason, fibrous or porous materials are used as thermal insulation. They are capable to restrain the air confined within the stagnant cells. Combined solid materials and gases are typically used as insulation, some of them are fiberglass, glass wool, expanded glass, expanded polystyrene, polyurethane foam, polyethylene foam, cork agglomerates, etc. In most cases the locked gas is air.

For this installation, the most proper insulation will be polyethylene foam, because, although its thermal efficiency is medium, it is water-resistant, easy to install and economic. It has a conductivity thermal coefficient between 0.035 and 0.045$\frac{W}{m \cdot K}$. The chosen insulation for this system will be an insulation of 6 mm thickness and a conductivity thermal coefficient of 0.037 $\frac{W}{m \cdot K}$ manufactured by Salvador Escoda [TB.4].

Table B.0.4: Insulation

Insulation	
Manufacturer	Salvador Escoda
Model	K-Flex ST
Thickness	6 mm
Conductivity thermal coefficient	0.037 $\frac{W}{m \cdot K}$
Length	Pipes length

When calculating the pipe losses, firstly, it must be differentiated the losses on the pipes depending on where they are located:

- For the pipes that are outside the building, it must be taken into account the wind speed, which means the generation of forced convection.
- For pipes that are inside the building, where there is no wind, the losses are calculated depending on the natural convection due to air that is around them, these losses are really small and therefore, they are negligible.

In this building, there is a preinstalled hollow from the basement to the roof for the installation of the water pipes; it can be also used for the pipes of the primary circuit. Therefore, it will be just consider the losses of the pipes located in the roof. These pipes are 40 meters long and as the hollow is in the middle of the building and the collectors will be located in a symmetric disposition, the pipes will be also installed symmetric.

And the total length of the pipes made from copper will be 142 m and its conductivity thermal coefficient will be 401 $\frac{W}{m \cdot K}$.

Finally, before calculating the losses, it is necessary to calculate the diameter of the pipes that will be determinate by the flow of each collector and the number of collectors.

$$D = \sqrt{\frac{4 \cdot Q}{\pi \cdot v}} \quad [B.33]$$

- $Q$is the total flow through the primary circuit ($\frac{l}{h}$).

$$Q = Q_{collector} \cdot A \cdot N \quad [B.34]$$

- A is the area of each collector ($A = 2.78\ m^2$).
- N is the number of collectors (N= 14).
- $Q_{collector}$ is the flow through one collector ($Q_{collector}$=30 $\frac{\mathrm{l}}{\mathrm{h} \cdot m^2}$).
- v is the velocity of the fluid within the collector.

$$Q = Q_{collector} \cdot A \cdot N = 1167.6\ \frac{\mathrm{l}}{\mathrm{h}} = 0.32\ \cdot 10^{-3}\,\frac{m^3}{s} \quad [B.35]$$

According to legislation, the velocity should between 0.5 and 2 $\frac{m}{s}$; the velocity supposed for the calculation in this installation will be 1 $\frac{m}{s}$.

Therefore, the diameter will be:

$$D = \sqrt{\frac{4 \cdot Q}{\pi \cdot v}} = 19.18\,mm \quad [B.36]$$

An according to Spanish regulation UNE-EN 1057 (standard sizes for pipes), the outside diameter is 22 mm and the inside diameter is 20 mm; so, thickness will be 1 mm. The chosen model will be the standard a rigid copper pipe 1 1 / 2 " model [TB.5].

Table B.0.5: Pipes

Pipes	
Internal diameter	20 mm
External diameter	22 mm
Thickness	1 mm
Conductivity thermal coefficient	$401\ \frac{W}{m \cdot K}$
Length	Pipes length

From inside the pipe to the thermal environment, there are 4 resistances whose outline is as follows [FB.3]:

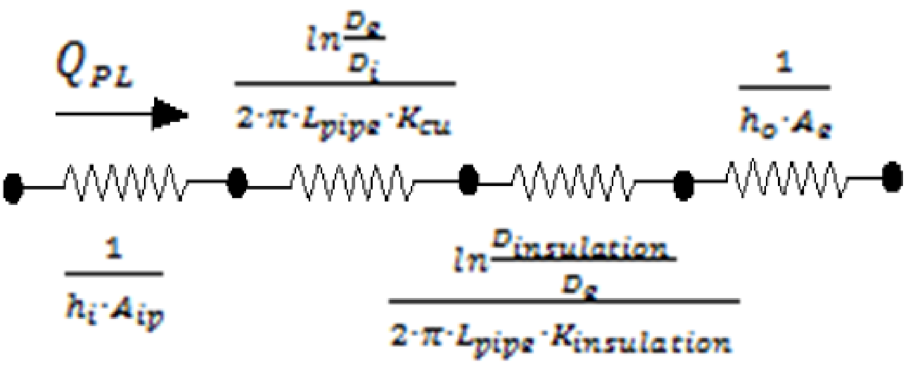


Figure B-0.3 : Thermal resistances (II)

- Q_{PL} is the heat lost by the fluid from inside the pipe to the environment.
- $R_{conv,ip} = \frac{1}{h_i \cdot A_{ip}}$ is the convection thermal resistance inside the pipe.

- $R_{pipe} = \frac{ln\frac{D_e}{D_i}}{2 \cdot \pi \cdot L_{pipe} \cdot k_{cu}}$ is the copper pipes thermal resistance.
- $R_{insulation} = \frac{ln\frac{D_{insulation}}{D_e}}{2 \cdot \pi \cdot L_{pipe} \cdot k_{insulation}}$ is the insulation thermal resistance for conduction transmission.
- $R_o = \frac{1}{h_o \cdot A_e}$ is the convection thermal resistance from the insulation to the environment.

When calculating the losses of the pipes exposed to the wind, it is important also to make a distinction depending on the temperature of the water that flows through it. However, the length of both pipes will be the same:

- $L_{hw,pipe} = 20\ m.$
- $L_{cw,pipe} = 20\ m.$

Therefore, the fluid properties have changed because they are not at the same temperature inside the cold and hot water pipes [TB.6].

Table B.0.6: Properties of the fluid inside the cold and hot water temperature

Pipes	Heat exchanger fluid: Propylene Glycol					
	μ($\frac{Kg}{m \cdot s}$)	υ($\frac{m^3}{s}$)	ρ($\frac{Kg}{m^3}$)	k($\frac{W}{m \cdot K}$)	Pr	Cp($\frac{KJ}{kg \cdot K}$)
Cold	$2.372 \cdot 10^{-6}$	$2.328 \cdot 10^{-3}$	1019	0.433	20.731	3781
Hot	$1.676 \cdot 10^{-6}$	$1.657 \cdot 10^{-3}$	1011	0.437	14.587	3801

<u>Convection inside the pipes</u>

As was made for collectors, firstly, it is necessary to calculate the Reynolds number of the fluid flowing inside the pipe to determine the flow regime in which the fluid is in order to choose the most appropriate calculation method most appropriate:

$$Re = \frac{u \cdot D_i}{\upsilon} \quad [B.37]$$

$$Re < 2300\ \rightarrow Laminar\ flux. \quad [B.38]$$

$$Re > 2300 \ \rightarrow Turbulent\ flux. \quad [B.39]$$

- u is the fluid velocity.

$$u_c = \ u = \frac{4 \cdot m_1}{\pi \cdot \rho_c \cdot {D_i}^2} = 1.037 \frac{m}{s} \quad [B.40]$$

$$u_h = \ u = \frac{4 \cdot m_1}{\pi \cdot \rho_{\square} \cdot {D_i}^2} = 1.045 \frac{m}{s} \quad [B.41]$$

- D_i is the internal diameter ($D_i = 20\ mm$).
- υ_i is the air cinematic viscosity ($\frac{m^3}{s}$).

$$\upsilon_c = 2.328 \cdot 10^{-6}\ \frac{m^3}{s} \quad [B.42]$$

$$\upsilon_h = 1.657 \cdot 10^{-6}\ \frac{m^3}{s} \quad [B.43]$$

So, Reynolds numbers will be:

$$Re_c = \frac{1.037 \cdot 0.02}{2.328 \cdot 10^{-6}} = 8908.9 > 2300 \ \rightarrow Turbulent\ flux. \quad [B.44]$$

$$Re_h = \frac{1.045 \cdot 0.02}{1.657 \cdot 10^{-6}} = 12613.15 > 2300 \ \rightarrow Turbulent\ flux. \quad [B.45]$$

Moreover, the turbulent flow is totally developed, so the Dittus-Boelter equation can be applied. It defines the Nusselt number in pipes based on Prandtl and Reynolds numbers.

$$Nu = 0.023 \cdot Re^{4/5} \cdot Pr^{0.4} \quad [B.46]$$

And the values must be in the following ranges:

$$\frac{L}{D} > 10 \quad [B.47]$$

$$10000 < Re < 10^6 \quad [B.48]$$

$$0.7 < Pr < 150 \quad [B.49]$$

And the convection coefficient is calculated by the following formula:

$$\begin{Bmatrix} Re_c = 8908.9 \\ Pr_c = 20.731 \end{Bmatrix} \rightarrow Nu_c = 111.74 \quad [B.50]$$

$$\begin{Bmatrix} Re_h = 12613.15 \\ Pr_h = 14.587 \end{Bmatrix} \rightarrow Nu_h = 128.22 \quad [B.51]$$

And the convection coefficient will be:

$$Nu = \frac{h_i \cdot D_i}{k_1} \rightarrow h_i = \frac{Nu \cdot k_1}{D_i} \quad [B.52]$$

$$h_{ic} = 2419.17 \frac{W}{m^2 \cdot K} \quad [B.53]$$

$$h_{ih} = 2801.6 \frac{W}{m^2 \cdot K} \quad [B.54]$$

Convection outside the insulation

As previously, before proceeding, it is necessary to check whether the flow on the panels is laminar or turbulent, which is defined by the Reynolds number:

$$Re < 5 \cdot 10^5 \rightarrow Laminar\ flux. \quad [B.55]$$

$$Re > 5 \cdot 10^5 \rightarrow Turbulent\ flux. \quad [B.56]$$

$$Re = \frac{\rho_{wind} \cdot v_{wind} \cdot L_c}{\mu_{wind}} \quad [B.57]$$

- ρ_{wind} is the air density ($\rho_{wind} = 1.184 \frac{kg}{m^3}$).
- v_{wind} is the average wind velocity (3.1 m/s).
- L_c is the characteristic length ($L_c = D_{insulation} = D_e + 2 \cdot L_{\text{thickness}} = 0.22 + 0.006 \cdot 2 = 0.232$ m).
- μ_{wind} is the dynamic viscosity of the fluid ($\mu_{wind} = 1{,}82 \cdot 10^{-5}$ Pa · s).

$$Re = Re_c = Re_h = \frac{\rho_{wind} \cdot v_{wind} \cdot L_c}{\mu_{wind}} = 46787.51 \quad [B.58]$$

So, $Re < 5 \cdot 10^5$ and the flux will be laminar. And the Nusselt number will be:

$$Nu = 0.193 \cdot Re^{0.618} \cdot Pr^{1/3} \quad [B.59]$$

- Pr is the Prandtl number.

$$Pr = \frac{\mu_{wind} \cdot C_{pwind}}{k_{wind}} \quad [B.60]$$

- k_{wind} is the air thermal conductivity and whose value is estimated in ($k_{wind} = 0.024 \frac{W}{mK}$).
- C_{pwind} is the wind specific heat ($C_{pwind} = 1007 \frac{J}{kg \cdot K}$).

$$Pr = 0.76 \quad [B.61]$$

So, Nusselt number will be:

$$Nu = 0.193 \cdot Re^{0.618} \cdot Pr^{1/3} = Nu_c = Nu_h = 135.5 \quad [B.62]$$

$$Nu = \frac{h_o \cdot L_c}{K_{wind}} \rightarrow h_o = \frac{k_{wind} \cdot Nu}{L_c} \quad [B.63]$$

$$h_{oc} = h_{oh} = 14.01 \frac{W}{m^2K} \quad [B.64]$$

<u>Conductance or heat transfer global coefficient.</u>

Equivalent resistance is the sum of each thermal resistance which influences the heat transfer from the fluid to the outside of the solar collector.

$$R_{eq} = R_{conv,ip} + R_{pipe} + R_{insulation} + R_o \quad [B.65]$$

- $R_{conv,ip}$ is the convection resistance inside the pipe.
- R_{pipe} is the pipe thermal resistance.
- $R_{insulation}$ is the insulation thermal resistance.
- R_o is the convection thermal resistance on the external insulation pipe.

When considering the thermal resistances, the area of each element has been included in its definition; so, the equivalent resistance of the collector is the product of global heat transfer coefficient and the collector area.

$$U \cdot A = \frac{1}{R_{eq}} \quad [B.66]$$

So, firstly, the value of each resistance will be calculated, then the value of equivalent resistance and finally, the global heat transfer coefficient. For this purpose, the following data will be needed:

- $L_{cp}$is the total length of the cold water pipes ($L_{cp} = 20\ m$).
- $L_{hp}$is the total length of the cold water pipes ($L_{hp} = 20\ m$).
- A_c is the area of the cold water pipes ($A_c = \pi \cdot D_i \cdot L_{cp} = 1.256\ m^2$).
- A_h is the area of the hot water pipes ($A_c = \pi \cdot D_i \cdot L_{hp} = 1.256\ m^2$).
- $A_{insulation}$ is the area of the insulation cover ($A_{insulation} = \pi \cdot D_{insulation} \cdot L_{cp} = \pi \cdot (D_e + 2 \cdot L_{\text{thickness}}) \cdot L_{cp} = 0.915\ m^2$).
- k_{cu} is the copper thermal conductivity ($k_{cu} = 401\ \frac{W}{mK}$).
- $k_{insulation}$ is the cover thermal conductivity ($k_{insulation} = 0.037\ \frac{W}{m \cdot K}$).

And the resistance of each element will be:

- Convection resistance inside the pipe:

$$R_{conv,ip} = \frac{1}{h_i \cdot A_p} \quad [B.67]$$

- The pipe thermal resistance will be:

$$R_{pipe} = \frac{ln \frac{D_e}{D_i}}{2 \cdot \pi \cdot L_p \cdot k_{cu}} \quad [B.68]$$

- The insulation thermal resistance will be:

$$R_{insulation} = \frac{ln \frac{D_{insulation}}{D_e}}{2 \cdot \pi \cdot L_p \cdot k_{insulation}} \quad [B.69]$$

- The convection thermal resistance on the external insulation pipe will be:

$$R_o = \frac{1}{h_o \cdot A_{insulation}} \quad [B.70]$$

So, the resistances in the cold water pipe will be [TB.7]:

$$R_{conv,ip,c} = 3.3 \cdot 10^{-4} \frac{W}{K} \quad [B.71]$$

$$R_{pipe,c} = 1.9 \cdot 10^{-6} \frac{W}{K} \quad [B.72]$$

$$R_{insulation,c} = 0.0114\frac{W}{K} \qquad [B.73]$$

$$R_{o,c} = 0.078\frac{W}{K} \qquad [B.74]$$

And the resistances in the hot water pipe will be [TB.7]:

$$R_{conv,ip,h} = 2.8 \cdot 10^{-4}\frac{W}{K} \qquad [B.75]$$

$$R_{pipe,h} = 1.9 \cdot 10^{-6}\frac{W}{K} \qquad [B.76]$$

$$R_{insulation,h} = 0.0114\frac{W}{K} \qquad [B.77]$$

$$R_{o,h} = 0.034\frac{W}{K} \qquad [B.78]$$

Table B.0.7: Thermal resistances (K /W)

Pipe	$R_{conv,ip}$	R_{pipe}	$R_{insulation}$	R_o
Cold water	$3.3 \cdot 10^{-4}$	$1.9 \cdot 10^{-6}$	0.0114	0.078
Hot water	$2.8 \cdot 10^{-4}$	$1.9 \cdot 10^{-6}$	0.0114	0.078

So, the global equivalent resistance will be:

$$Cold\ water\ pipe \rightarrow R_{eq,c} = 0.0897\frac{K}{W} \qquad [B.79]$$

$$Hot\ water\ pipe \rightarrow R_{eq,h} = 0.0896\frac{K}{W} \qquad [B.80]$$

And finally, with all the thermal resistances, the global heat transfer coefficient can be calculated:

$$U \cdot A = \frac{1}{R_{eq}} \qquad [B.81]$$

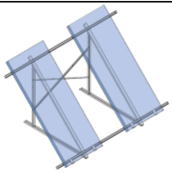
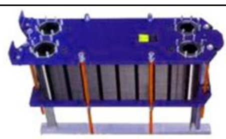
$$Cold\ water\ pipe \rightarrow (U \cdot A)_c = 11.144\frac{W}{K} \qquad [B.82]$$

$$Hot\ water\ pipe \rightarrow (U \cdot A)_{\ h} = 11.16\frac{W}{K} \quad [B.83]$$

All these values are taken in to account just the average data for a year but when calculating the monthly solar fraction it will be also necessary the monthly data.

APPENDIX C: LIST OF ELEMENTS

Table C.1: List of elements

Element	Manufacturer	Model	Units	Refer-ence	
Solar collector	Salvador Es-coda S.A.	SOL 2800 selective	14	**[23][24]**	
Bracket	Salvador Es-coda S.A.	SOL 2800	14	**[23][24]**	
Expansion tank	Zilmet	Solar Plus 50	1	**[26]**	
Pump	Grundfos	Solar 15-65	2	**[25]**	
Pipes		Copper pipe 1 1 / 2 "	142 m		
Insulation	Salvador Escoda S.A	K-Flex ST	142 m	**[23]**	
Heat exchanger fluid	Sonnerkraft	Propylene glycol (39.5%)	40 l	**[2][30]**	
Heat exchanger	Alfa Laval	M3-FM	1	**[27]**	
Accumulator	Acro ISO	PS Series 2500	1	**[28]**	
Accumulator	Acro ISO	PS Series 500	1	**[28]**	
Boiler	Junkers	ZW420	1	**[29]**	

APPENDIX D: BUILDING FLOOR PLANS

Below, there are the plans of one of the floors of the the building and a roof overview, that is where the facility will be installed.

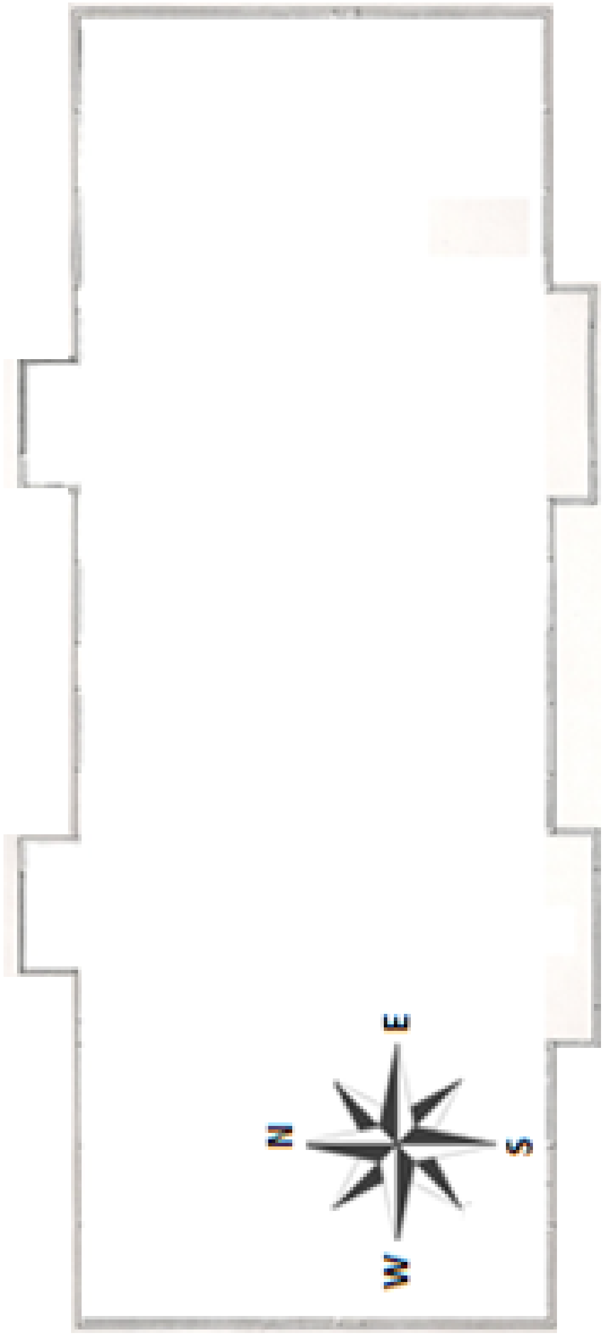


Figure D.1: Roof overview

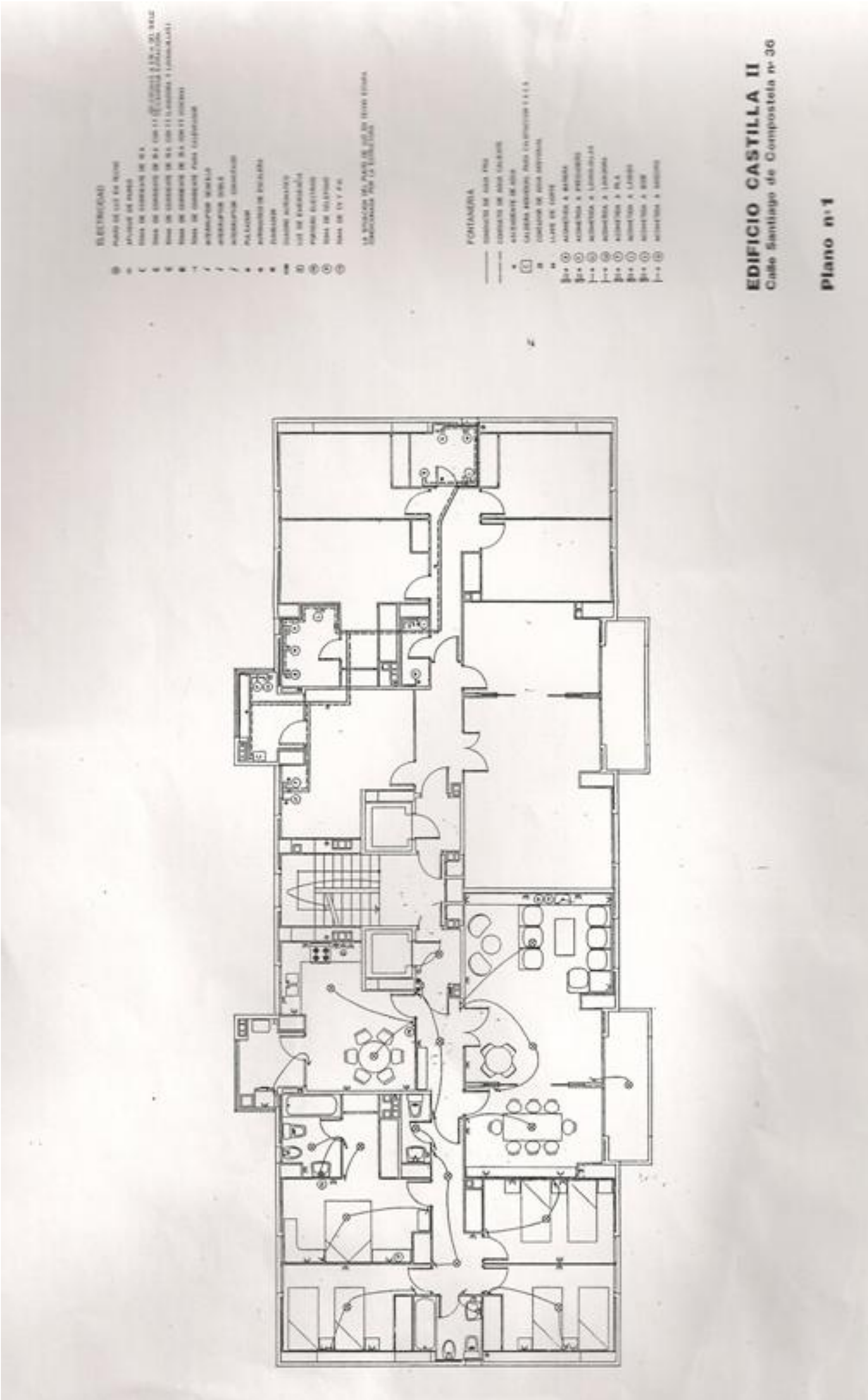


Figure D.2: Floor overview

APPENDIX E: LIST OF TABLES

APPENDIX F: LIST OF FIGURES