A Technical Report On

Possibilities for Renewable Energy Systems

Within a Recreational Facility

[redacted]

Contents

1) Introduction

One of the main aims of this report has been to minimise the impact on the environment with regard to the reduction of Carbon Dioxide from a building such as a Recreational Facility, which by its nature is potentially a high consumer of energy.

This report can be seen as an example of how alternative technologies can be brought together in the design of a low carbon creating building of commercial size.

In carrying out this report a various design methods and principles have been used to enable the selection of technologies to integrate together to form the building services.

The report is limited however to the design of particular building services in respect of heating, ventilation and pre-cooling with underground air pipes.

The undertaking of this report has highlighted the complexity of building services currently utilised in commercial sized buildings. In particular the extent to which building services integrate and bind the building together.

Buildings should be designed holistically so that from the very beginning the building services concepts can be integrated as part of the building and not 'shoe horned' in as an afterthought. The new building regulations are geared towards this synergy and integration.

More and more buildings being built today do utilise passive, renewable and sustainable energy approaches, which require the early integration of building services, this report echoes the holistic approach and demonstrates innovation in several aspects building services design.

2) Aim

The aim of the report is to demonstrate that the author has the engineering knowledge, technical ability and professional competence to undertake the necessary research in detail of the various systems that can provide heating and ventilation via sustainable and renewable means.

Solar energy is the main renewable source utilised from Biomass (Photosynthesis) and solar collectors to provide ventilation specifically driven by solar power.
To achieve this various concepts have been considered in order to optimise zero carbon emissions with regard to fossil fuels, over the life of a commercial sized building.

During the consideration process several alternative building services concepts have been examined and considered. The inclusion of renewable and alternative technologies such as Solar Thermal, Solar PV, Natural ventilation, Stack ventilation, Biomass and passive building design should be included as it has been suggested that their inclusion would add 1 or 2% to a projects overall budget to.

However in this building design cost has been a low priority in favour of the conceptual nature and design and as such where possible the utilisation of low carbon and renewable energy technologies has been used.
The main starting point was to create a building with good thermal properties by selecting materials, which would enable low fabric heat losses to be achieved for a site located in the South of England in the UK.

Additionally the building materials have been selected to give a high thermal mass to the structure, this will give the building a high thermal capacitance and help prevent the effect of wide seasonal temperature swings by giving the building a slow thermal response.

However, that said, there are requirements for certain space functions to have relatively high ventilation rates, which will increase the heat loss through ventilation by a factor of nearly 5, (i.e. swimming pool area when ventilation calculated per wetted area of pool).

A tempered air supply will be included which the biomass boilers and the solar thermal system will serve supplemented by energy recovered from the ventilation stacks.

Mechanical ventilation plant has been avoided and in its place a method of stack ventilation adopted utilising solar thermal technology and underground air-pipe[1] supply ducting.

Under floor heating will be the main form of space heating throughout the building, with additional tempered air supply for areas with high ventilation rates. Modular biomass woodchip boilers will primarily provide heating during the winter months and during the summer months solar thermal energy will heat the pools and provide a seasonal hot water store to supplement the heating and hot water during autumn.

During hot periods in the summer some of the heat from the solar thermal system will be used to drive the stack ventilation system ensuring a temperature differential at all times to promote the required density differences of internal and external air.

3) **Background**

Set against the background of global warming and the requirement to reduce levels of CO_2 and other 'green house' gases, the building services industry has obligations to promote sustainable options for current building services practices and where possible to integrate renewable energy technologies into both new and existing buildings.

The provision of heating utilising renewable energy such as biomass and ventilation without the use of fan power is not a new concept and with new legislation in the form of the new building regulations part L2 of which the Energy Performance in Buildings Directive which is due for implementation in April 06, the use of renewable energy has become high profile and has to be seriously considered.

This study will concentrate on woodchip fuel utilising short rotation coppicing from woodland management schemes and Solar Thermal Energy.

Regarding the ventilation systems, research has revealed that the Victorians were experts in this field and many buildings visited during the research for this report demonstrated the use of stack ventilation on a commercial scale within such places as Prisons and Hospitals.

Similarly, others have researched the use of underground air pipe arrays to form supply air ducts and reference will be made to their work as appropriate throughout this report.

4) Technical Content and Description

The technical content of the report will focus on the development of renewable energy systems for space heating, hot water and ventilation purposes.

Schematic drawings demonstrating the technical detail and concepts employed will be supplemented by calculations and analysis where appropriate.

Spreadsheet calculations will also be used to assist the design aspects of the report and supplementary data and information will be listed and found in the appropriate appendices.

Detailed calculations for the boiler sizing, under floor heating, solar thermal system and stack ventilation system will be demonstrated and how the sizing of the heat exchangers has been analysed and determined.

The selection process for determining the type of solar collector and percentage to be installed will also be detailed.

To further promote the sustainable nature of the system the solar collectors will operate on a primary closed loop thermosyphon principle, which will allow circulation pumps to be omitted.

It is the intention of the report to illustrate the technical challenges and the analysis required to demonstrate how the system will function under varying external temperature conditions and modulates to maintain internal environmental conditions. The control of the heating and ventilation systems including energy recovery will also be examined and presented in the report.

4A) Stack ventilation system and swimming pool ventilation

The building has been designed with low energy in mind this is reflected in the fact that no mechanical ventilation is utilised. In its place a more innovative method of stack ventilation is used which is to a large extent built into the building from the early stages of construction.

The stack ventilation system will comprise of 5 stacks integral to the building that will be of varying height depending on the amount of spaces to be ventilated and the corresponding volume of air to be extracted.

The stacks will form the extract plenham ducts from which air will be extracted to atmosphere from the building spaces. Air will be induced through the different internal spaces due to the differential density of internal space air temperature and external outdoor temperature.

This method will be further enhanced during the summer months when the temperature differential is lower by the inclusion of solar thermal heating pipes coiled around the inner circumference of the stacks, the solar thermal aspect to this will operate on the thermosyphon principle. (See figure S1[6])

ENHANCED STACK VENTILATION SYSTEM[6]

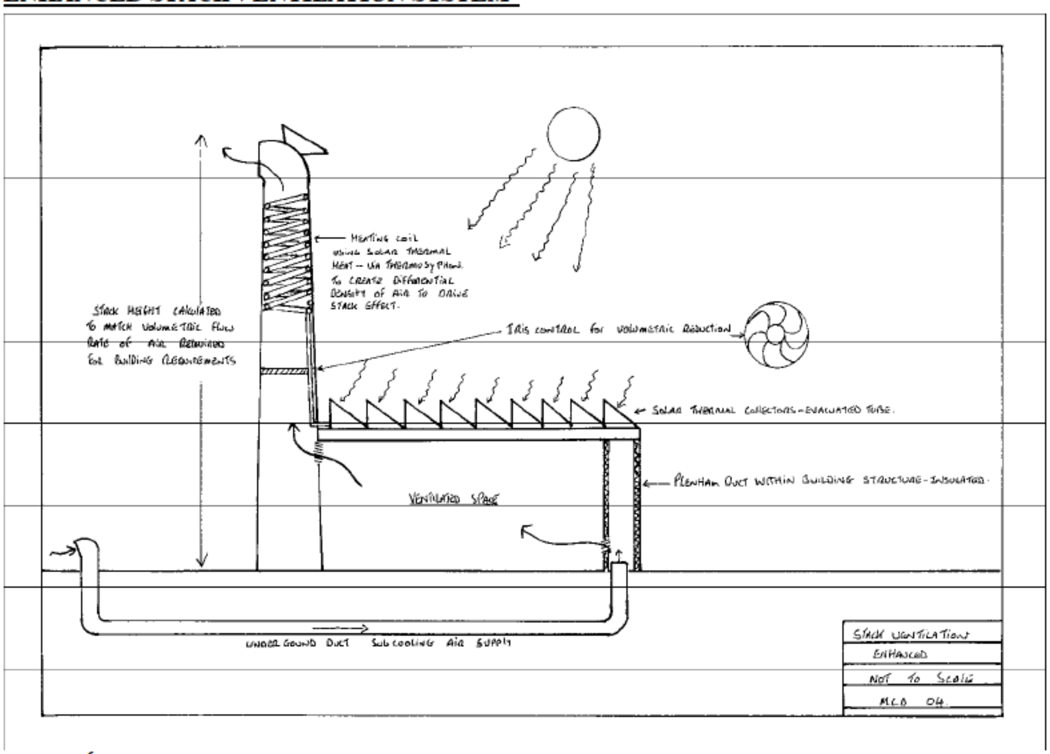


Figure S1[6]

Supply air will come into the building via an underground air pipe array system and into several supply plenham ducts before being displaced into the spaces served due to the stack effect.

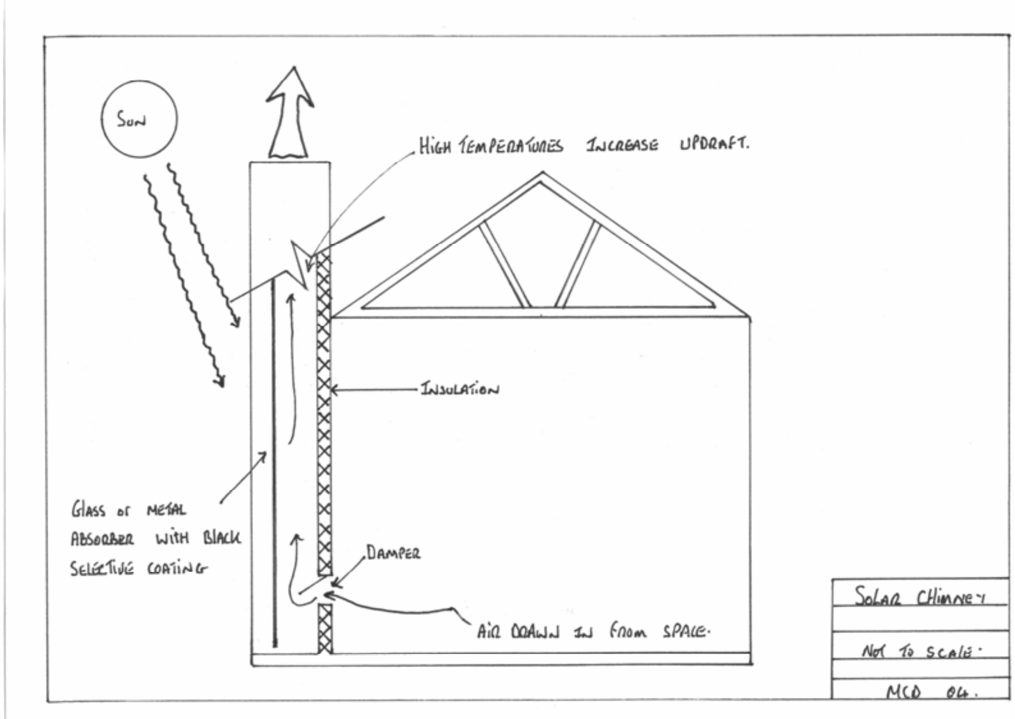
Volumetric airflow rate calculations for individual spaces have been carried out in the same manner, as a mechanical ventilation system would be. However the extract ducting takes the form of air passages constructed to the sizes required for volumetric airflow rates and a maintained velocity of between 2.5 to 3m/s[1], sizes for these and sizing of the stacks can be found in the analysis section on page 19.

The air pipe system providing the supply air to the building will be covered in the next chapter but in essence supply air is drawn into the building via underground ducts some distance from the building.

During the winter the air exiting the air pipes in the building will have picked up some heat from the ground, which will have raised the exiting temperature from the inlet. Similarly in summer the incoming air will be cooled by the same process this is dependant on the type of shade and ground cover around the building.

The control of the ventilation system will be via the internal and external air temperatures and in addition Iris air control dampers will be installed in the stacks to control the volumetric flow of air during normal use and as a safety feature the Iris control dampers fail in the closed position. This is an extra precaution should a possible fire occur within the building. (See figure S1[6])

Figure S2[6] ***SOLAR CHIMNEY***



Consideration has been given to the solar chimney[16] method of ventilating buildings and the effect on air within a stack when an absorber plate is heated through glazing along the height of the ventilation chimney.

However although this method is effective it does have one main drawback and that is the orientation of the stacks and the building must always be facing due south to benefit from this method, this may not always be feasible. In comparison having coils within the stack heated by solar thermal collectors means the collectors could be remotely place to get maximum irradiance from the sun.(See fig S2[6]).

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VENTILATION DUCT LAYOUT FROM MAIN STACKS[6]

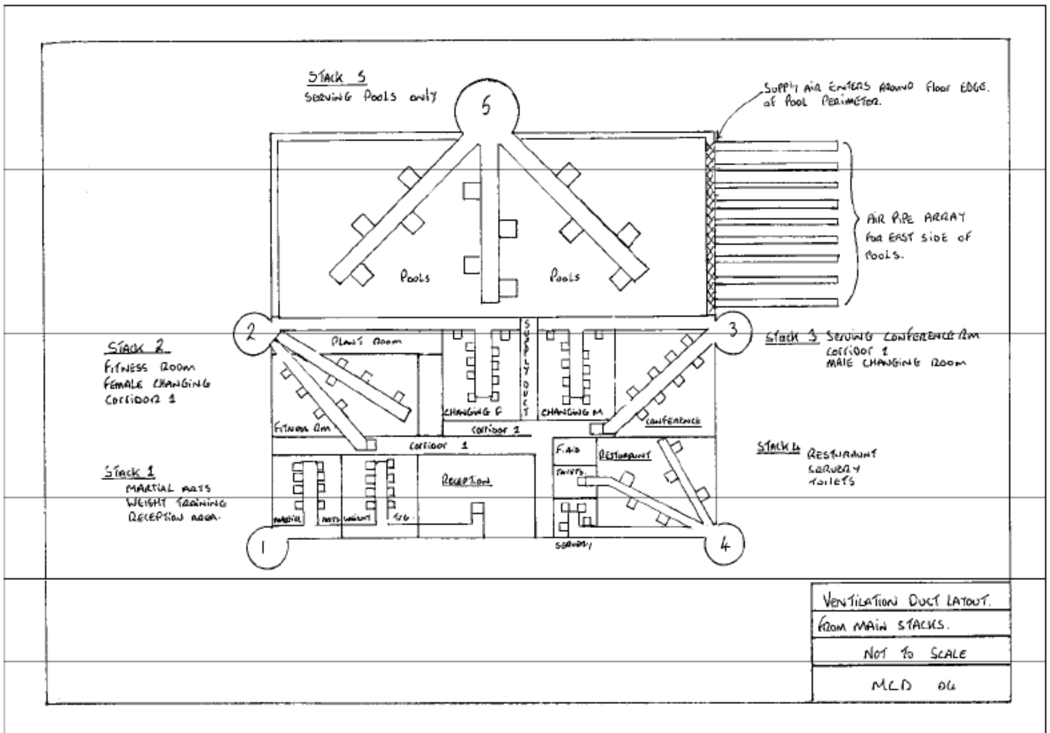


Figure S3[6]

It can been seen from figure S3 that the building is split into 5 ventilation zones and it can also be noted the rather unorthodox detail of the ducts, which take a more direct route to the spaces to be extracted this is to eliminate unnecessary bends and branches which would create additional resistances to air flow.

4B) Underground air pipe supply

The supply air pipe ducting is only detailed on the east side of the Swimming pool area there are approximately 45 to be installed for the swimming pool area with 0.5m spacing between centres[1].

The rest of the air pipe system for the building at this stage has not been calculated, as the pool area only will form the analysis.
The supply air pipes for the pool area will enter the building around the perimeter of the walls forming a raised grille along the whole perimeter of the swimming pool area.

The grille arrangement has to be raised to prevent the ingress of water should the floor area become wet as is usually the case in swimming pool areas.

In other areas for instance the changing/shower areas have a supply plenham duct between male and female where the air pipe system will enter before discharging into the changing areas. (See figure S4[6]).

PLENHAM SUPPLY AND AIR PIPES

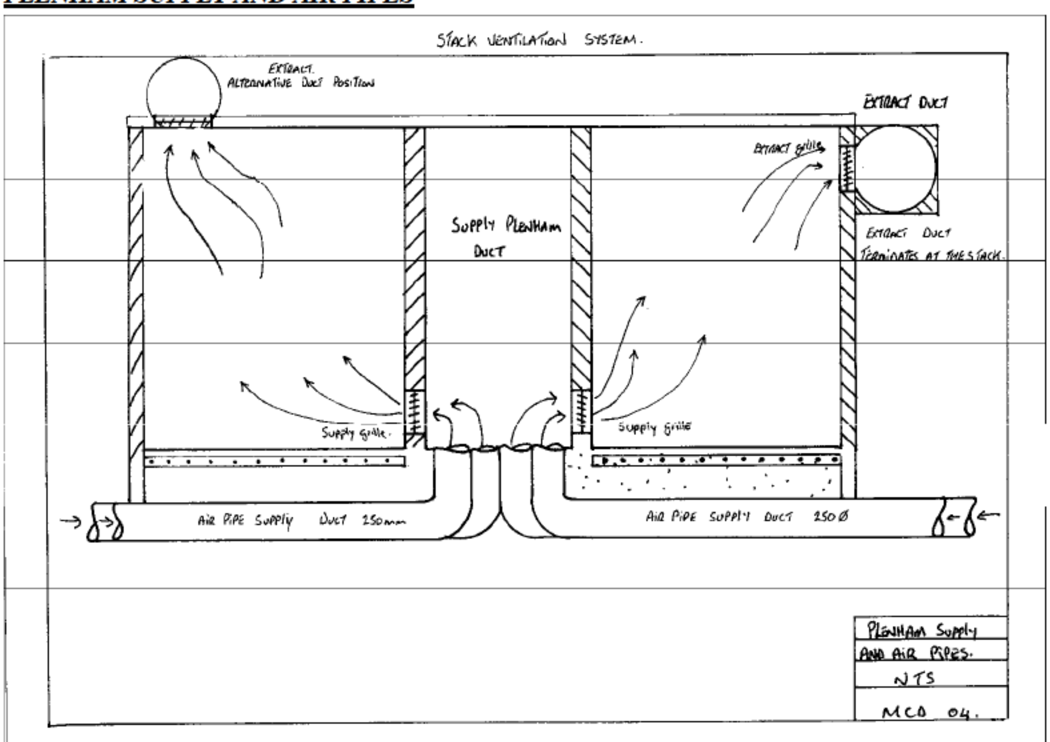


Figure S4[6]

Once the air has circulated in the changing/ shower areas it will be extracted via the high level grilles and be conveyed to the stack before being discharged to atmosphere.

The amount of air pipes is dependent upon the volumetric airflow rate required for the space, calculations for the quantity of air pipes is included in the air pipe section and in the appendices at the rear.

The dimensions of the interior plenham ducts in general are larger for ease of construction during the building phase.

The design of the air pipe arrays, which will serve the building, has been done via information gathered by others after an in-depth thesis on the subject was carried out in 1999[1].

Guidance values for the optimum lengths depths and diameters have been summarized in the table S5 below from the thesis data[1].

Pipe length m	**10 m**
Pipe burial depth m	**4m**
Pipe radius m	**0.125m**
Air velocity m/s	**3-4m/s**
Pipe array spacing	**0.5m**

Table S5[1]

From the research information it is generally considered that multiple pipe installations give a better output than single large diameter pipes or even tunnels. Soil temperatures[1] at approx 4m remain constant and are equal to the yearly sol-air[3] temperature recorded at the earth's surface.

During the winter the air pipe system will increase the incoming air from -5^0Cdb to 1^0C alternatively during the summer months from an outside design of 28^0C db the drop will be around 22^0C[1].

For the design of the ventilation system a temperature of 6K[6] has been used as the temperature increase, this temperature parameter has been taken from the research information.

For this specific reason the area of ground above the air-pipe system for optimum air-pipe performance would need to be wet and shaded during the summer and during winter blackened and sunlit to provide a level of pre heating[1].

To help create the conditions for a shaded area in summer and sunlit in the winter trees are planned to be planted around the perimeter of the building in depth (planting boarder) these trees will be deciduous and of a variety which will not grow too high but provide good shade during the summer.

This will also help keep the building heat gains down during the summer by providing shading from the sun in the from of a break between glazing and the sun see figure S6[1]

SHADING THE BUILDING AND GROUND FROM THE SUN.

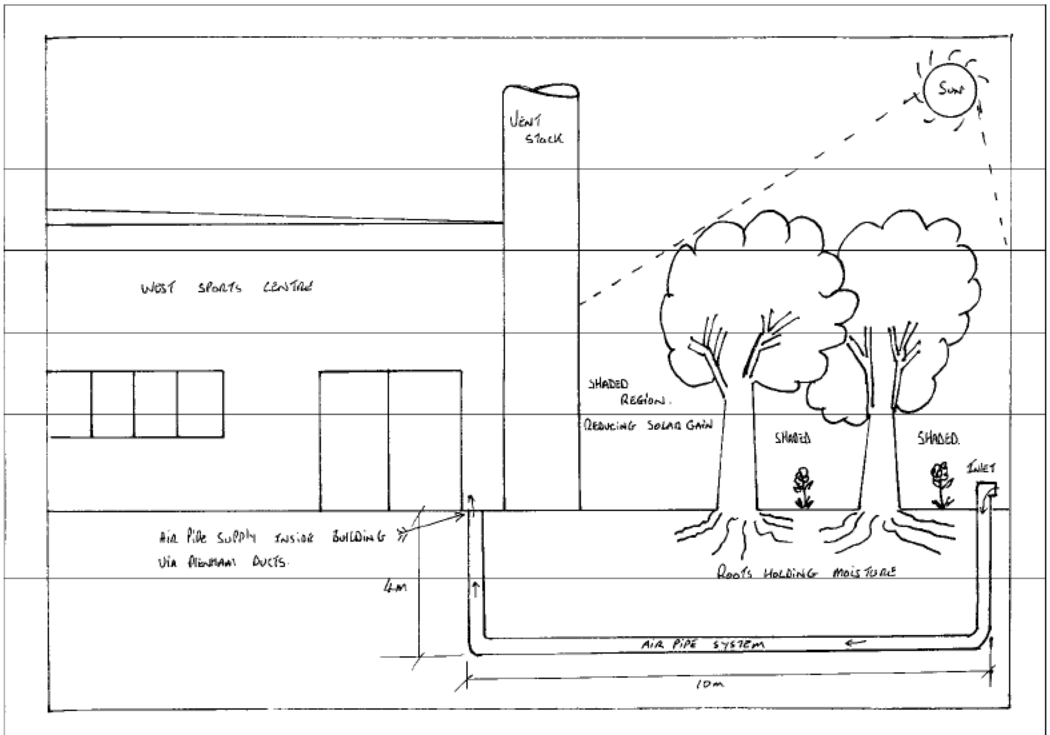


Figure S6[1] (Drawing and design by M C Durkin)[6]

The air pipe system has to be on a slight fall to enable condensation and moisture to collect at one point to drain via a soak away, which is not shown in the above diagram, also access for cleaning has to be considered and this function is simple to carry out with the correct equipment or cleaning company[1].

The amount of supply air is regulated by the pull on the stack system and the air tightness of the building volume control dampers in the stack system can close right down which will stop the air being drawn through the air pipe system.

4C) Swimming pool area design overview

The pool area should be 1^0C above the pool water temperature and the pool temperature should be 25-28^0C for recreation and training pools, between 28-30^0C for learner pools[18]and 30-35^0C for Hydrotherapy pools.[18]

Ventilation for swimming pools has to be done on a 24hr basis due to the build up of chlorine within the atmosphere of the pool areas[18].

The ventilation rate for the pool area during non occupied times can be substantially lowered due to moisture content release being reduced[18].

The CIBSE gives guidelines on the amount of ventilation air required for pool areas and is 10-15 l/s per m^2 of pool-wetted area[18].

The ventilation of the pool will be balanced to aid in the drying of pool surrounds under floor heating is included around the pool surrounding areas to help achieve this[18].

The pool design allows for the pools to be heated over a period of 72 hours from a cold-water temperature of 10^0C cold to a design temperature of 27^0C for the main pool and 35^0C[18] for the hydrotherapy/learner pool.

The main pool holds 825,000ltrs[6] of water and the hydrotherapy pool holds 150,000ltrs of water the following calculations for the pool boiler plant sizing are based on the individual masses of water for each pool as the temperature difference for each pool is different.

Main pool

Using the equation, [14]

$$Q = \frac{M\, Cp\, \Delta t}{\theta \times 3600}$$

Where;

Q = heat required in kW
M = Mass of water to be heated Kg
Cp = Specific heat capacity of water KJ/KgK
Δt = Temperature difference ^{0}C
θ = Time period in hours

[14]

$$\frac{825000 \times 4.19 \times (27-10)}{72 \times 3600} = \mathbf{145\ kW}$$

Hydrotherapy pool

$$\frac{150000 \times 4.19 \times (35-10)}{72 \times 3600} = \mathbf{61Kw}$$

The pool space area will be heated to a temperature of 28^0C[18] regardless of the hydrotherapy pool having an elevated temperature of 35^0C it is considered unrealistic to be heating the pool area to 36^0C as this would affect the comfort conditions and create a situation where the main pool at 27^0C would appear to feel cold to the main pool users.

The heating as described in earlier sections will comprise of Solar Thermal and Biomass wood chip boilers, which will heat the swimming water via plate heat exchangers controlled by water temperature sensors and diverting valves.

The pools have been designed to have total independence of filtration plant and heat exchangers to cater for the differing operating times and temperatures of the pools, in addition the plant can be maintained without the loss of function of one of the pools.

Below in diagram P1[6] is the general schematic of the pool filtration plant and heating arrangements via plate heat exchangers.

The balance tank arrangement is provided to maintain the required pool water levels by injecting make up water at the dictates of the level indicator.

The sand filters are a typical arrangement and provide fine filtration and backwashing to clean and flush debris from the sand at maintenance intervals.

<u>SWIMMING POOL FILTTRATION SYSTEMS</u>

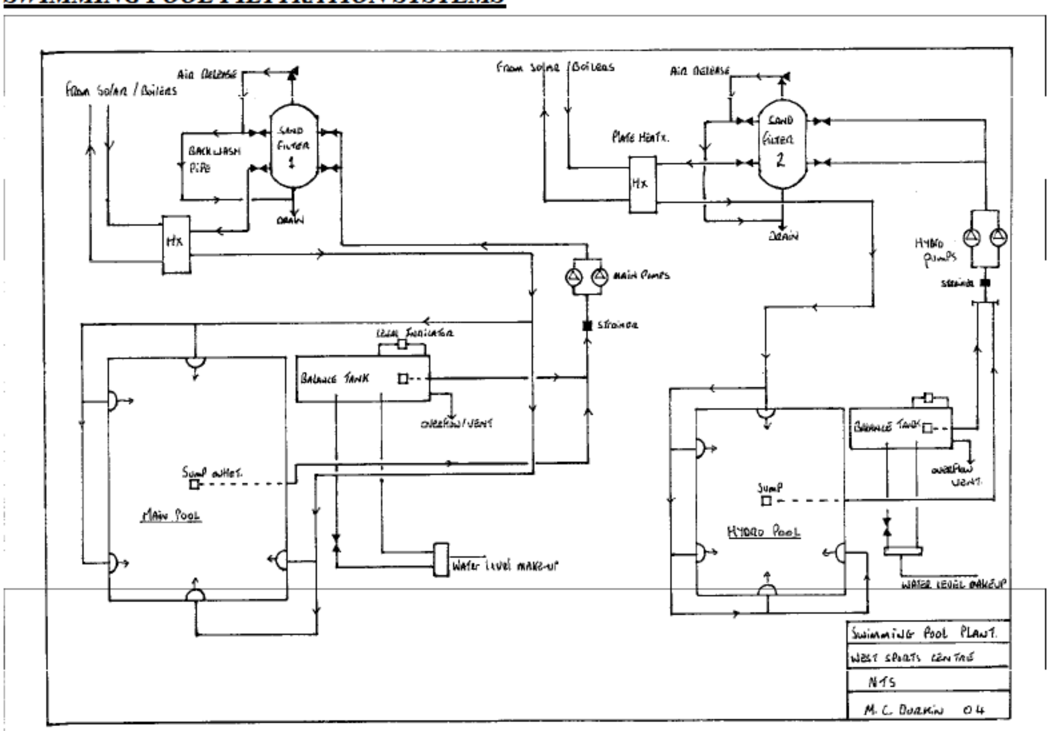


Diagram P1[6]

4D) Swimming pool ventilation systems

As described in earlier sections the ventilation is provided via 'Enhanced Solar Stack[6]' as designed by the author where basically air is moved through the pool area via the differential densities of warm and cool air.

The hot air will rise up and through the grilles and ducting to the stack where the velocity is maintained at least at 3m/s[16] if not more during the summer months when the ambient temperature outside is high.

The volumetric airflow rate for the pool area is based on CIBSE recommendations of 12ltr/sec per m^2 of wetted pool area[18]. It has been calculated that the wetted pool area will be $525m^2$ and the volume of the pool area is $6000m^3$ this gives a volumetric airflow rate of $6.67m^3/s$ or roughly 4 air changes per hour extract.

In addition to the air pipe system raising the outdoor temperature by some 6K[6], the tempered air supply to the building will be provided by heater batteries and recovery coils. They will be positioned in plenham ducts between outlet diffusers in the spaces to be heated and air inlet pipes.

Biomass woodchip boilers will provide supplementary heat energy for the heater batteries to maintain the required space temperature during winter months. A sketch of the general arrangement for the supply of tempered air can be seen in figure P2[6] this being the swimming pool tempered air supply, note that the recovery coil will be part of a run around energy recovery system from the ventilation stack serving the swimming pool area.

POOL SUPPLY AIR SYSTEM[6]

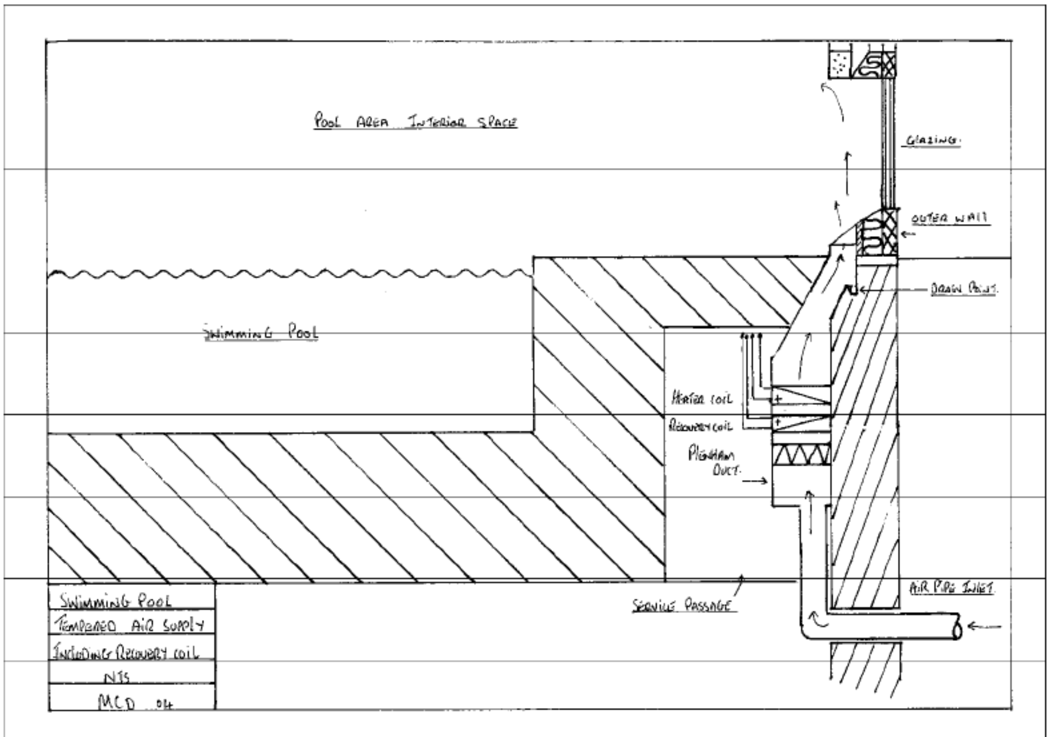


Figure P2[6]

4E) <u>Energy recovery system</u>

The recovery of waste heat energy due to high ventilation rates is paramount in the interests of energy conservation. The total heat loss from the building is approximately 105Kw[6](not including the ventilation tempered air supply) however due to the high ventilation rates of the swimming pool area, the restaurant and various gymnasia the ventilation loss is much greater.

This figure can be offset by the inclusion of energy recovery in the form of recuperation. For the swimming pool areas a run-round system has been selected. See figure P3[9]

SWIMMING POOL HEAT RECOVERY SYSTEM

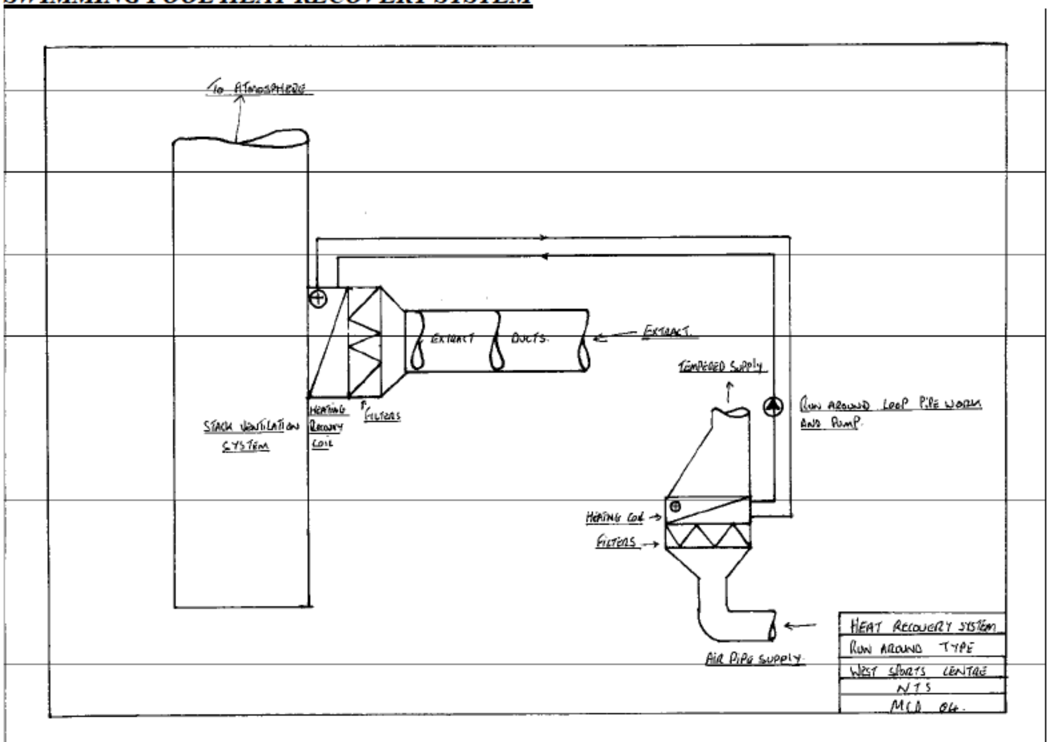


Figure P3[9]

Such a system can recover a substantial amount of energy, which can be reused to offset the energy required to heat the tempered air supply and is known as a recuperative heat exchange process.

The method shown in schematic in figure P3[9] is used because the two fluids required to exchange heat, are quite far apart. In addition the primary fluid i.e. the extract air exchanges heat indirectly through heat transfer to a water system then from the water system to the supply air stream.

For this system to work effectively good insulation is required to prevent heat losses from the water system pipe work.

The size of the run-round coil system can be seen from the data below a typical heat transfer rate for a run-round coil of10kW/K (for example).[9,22] has been selected. Other typical figures used for the specific heat capacity of both streams of air is 1.005kJ/kg/K[19] for both supply and extract air and the specific heat capacity of water is 4.19kJ/kg/K[19], then using the following formula for heat recovery.

[9]

$$\mathbf{Q = \frac{\underline{(UA)_H\,(t_{H1} - t_{C2})}}{2 + ((UA)_H/(mc)c)}}$$

Where,

Q = heat recovered Kw
UA_H = heat transfer coefficient Kw/K
(mc)c = thermal capacity Kw/K
t_{H1} = extract air temperature from space ^{0}C
t_{C2} = outside air entering the system

From a volumetric flow rate for the swimming pool area of 6.67m^3/s x density of air 1.2kg/m^3 = **8.0kg/s**[6] mass flow rate of air.

Thermal capacity = 8kg/s x cp of air 1.005kJ/kg/K = **8.04Kw/K**

Therefore the heat recovered from the formula above is

9

$$\frac{\mathbf{10 \times (28\text{-}1)}}{\mathbf{2 + (10/8.04)}} = \underline{\mathbf{83.25kW}}$$

The temperature of supply air leaving the recovery coil can now be calculated from the formula,

9

$\mathbf{t_{C1} = t_{C2} + Q / (mc)c}$ = 1+ 83.25 / 8.04 = **11.35^{0}C**

The temperature of the secondary fluid $\mathbf{ts_1}$, which is conveying the recovered energy to the heat exchanger in the supply air steam, is then calculated from the expression,

$\mathbf{ts_1 = (th_1+tc_1)/2}$ = (28+11.35)/2 = **19.67^0 C**

Because the specific heat capacity of the supply and extract air steams are the same the temperature lines in this analysis are parallel. With this being the case the interval difference between the temperatures $\mathbf{th_1}$, $\mathbf{ts_1}$ and $\mathbf{tc_1}$ will be the same for $\mathbf{th_2}$, $\mathbf{ts_2}$ and $\mathbf{tc_2}$ this can be seen by observing table **R2** below

Therefore, $\mathbf{th_2 = (th_1 - tc_1) + tc_2}$ = (28 – 11.35) + 1 = **17.65^{0}C**

Finally to find ts2 the expression used to find ts1 is rearranged thus,

$\mathbf{Ts_2 = (th_2 + tc_2)/2}$ =(17.65+1)/2 = **9.32^{0}C**

The percentage saving of energy would then be the temperature difference of the system with heat recovery divided by the system without heat recovery.

9

$$\frac{(11.35\text{-}1)}{(28\text{-}1)} \times 100 = \underline{\mathbf{38\%}}$$

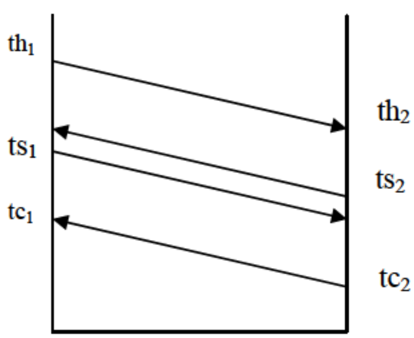


Table R2

HEAT RECOVERY CALCULATION DATA

tc2	th1	sup temp	Ht battery 1	kg/s	(UA)h	ts1	ts2	tc1	th2	Q	Ht Battery 2	Saving %	Stack delT
1	28	30	233.16	8	10	19.7	9.3	11.4	17.6	83.2	149.9	38	17
2	28	30	225.12	8	10	20.0	10.0	12.0	18.0	80.2	145.0	38	16
3	28	30	217.08	8	10	20.3	10.7	12.6	18.4	77.1	140.0	38	15
4	28	30	209.04	8	10	20.6	11.4	13.2	18.8	74.0	135.1	38	15
5	28	30	201	8	10	20.9	12.1	13.8	19.2	70.9	130.1	38	14
6	28	30	192.96	8	10	21.2	12.8	14.4	19.6	67.8	125.1	38	14
7	28	30	184.92	8	10	21.5	13.5	15.1	19.9	64.7	120.2	38	13
8	28	30	176.88	8	10	21.8	14.2	15.7	20.3	61.7	115.2	38	12
9	28	30	168.84	8	10	22.1	14.9	16.3	20.7	58.6	110.3	38	12
10	28	30	160.8	8	10	22.5	15.5	16.9	21.1	55.5	105.3	38	11
11	28	30	152.76	8	10	22.8	16.2	17.5	21.5	52.4	100.4	38	10
12	28	30	144.72	8	10	23.1	16.9	18.1	21.9	49.3	95.4	38	10
13	28	30	136.68	8	10	23.4	17.6	18.8	22.2	46.2	90.4	38	9
14	28	30	128.64	8	10	23.7	18.3	19.4	22.6	43.2	85.5	38	9
15	28	30	120.6	8	10	24.0	19.0	20.0	23.0	40.1	80.5	38	8
16	28	30	112.56	8	10	24.3	19.7	20.6	23.4	37.0	75.6	38	7

Recuperative Energy Recovery System

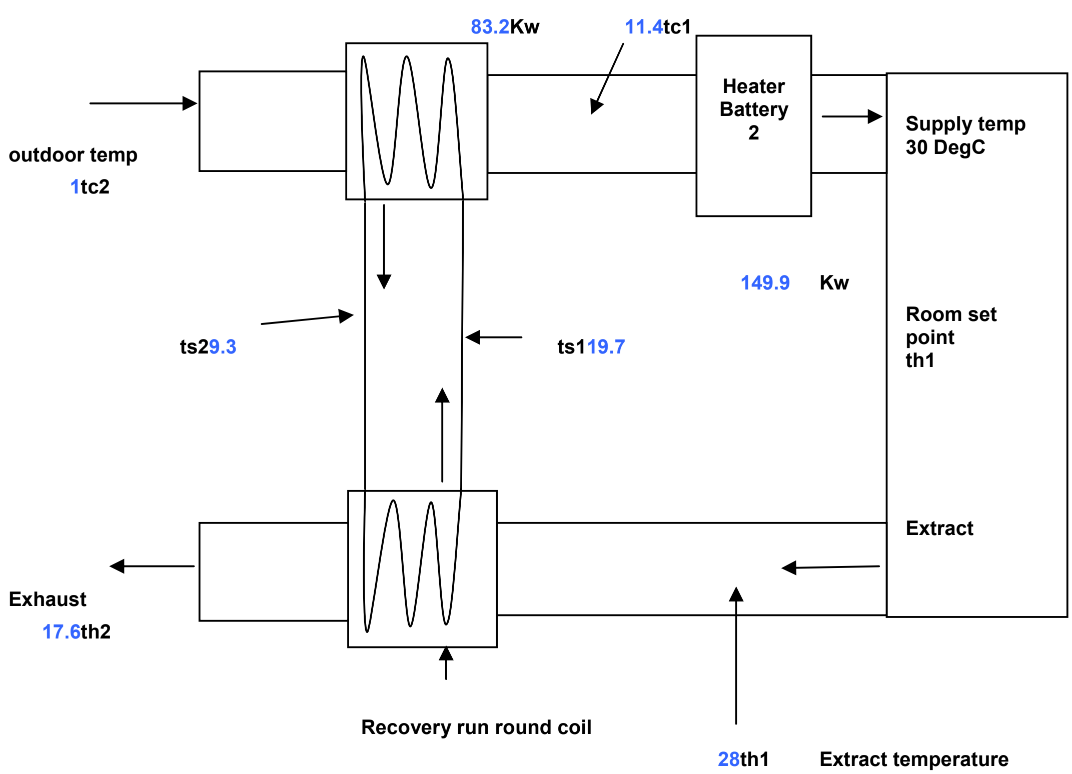


Figure P4[9]

The heater battery size without recovery can be seen above as heater battery 1 at 233kW and with the inclusion of the air pipe system and the energy recovered from the run-round coil system the heater battery output will need to be 149kW this being produced by the Biomass boilers.

4F) **<u>Stack ventilation system analysis</u>**

In order for the stack system to function the difference in density of the inside air of the building and ventilation stack and the outside ambient air has to be maintained.

The temperature difference, which has been selected to maintain the inside/outside differential is 6K, and a velocity of 3m/s, which is a recommended figure[13] for natural ventilation systems.
The entry temperature to the stack is 28 ^{0}C, which is the pool area space temperature, and the exit temperature from the stack with a 6k differential is 34^{0}C.

In practical terms this means the outside air temperature would need to be above 40 ^{0}C before the system would fail to maintain the velocity of 3m/s.

This would only be the case on a very hot summer's day in July but not quite yet in the UK, typical summer peak temperatures are 28to30^{0}C.

And by using the following formula for the calculation of air velocity (due to natural draught[14])

$$\mathbf{V = 4.43\ \sqrt{}h(tc\text{-}to)/273+to}$$

Where;

V = velocity m/s (1b:1)
4.43= constant
h = height in meters
tc= temperature inside column ^{0}C
to= temperature outside air ^{0}C

$$\mathbf{Draught\ Pa = h\ (\rho0\text{-}\rho1)g}$$

Where;

ρ_0 = Density of outside air (1b:2)
ρ_1 = Density of inside air
h = Height in meters
g = Acceleration due to gravity.

Or by utilising temperature differentials the above becomes

$$\mathbf{Draught\ Pa = 0.0465\ x\ dt\ x\ h}$$

Where;

dt = temperature difference (1b:3)
h = Height in meters

Therefore $\mathbf{0.0465\ x\ (34 - 28)x\ h}$

However we need to find h and to calculate the draught therefore if we transpose formula (1b:1) to make h the subject and we insert 3m/s as the required velocity we have,

$$\mathbf{h = \frac{\underline{273 + to \times V^2}}{(tc-to)\ 4.43^2}} \qquad (1b:4)$$

$$\mathbf{\frac{\underline{273+28 \times 3^2}}{(34-28) \times 4.43^2} = \underline{23meters}}$$

As a final check if we put the result of 23 meters in formula (1b:3) the draught in Pascals is

$$\mathbf{0.0465 \times (34-38) \times 23 = 6.417\ Pa} \qquad (1b:5)$$

where 6.417 Pa = 6.417N/m^2

Therefore to calculate the velocity from the formula

$$\mathbf{V = \sqrt{p2/\rho}} \qquad (1b:6)$$

Where V = velocity m/s
P = pressure in Pa
2 = constant
ρ = density of air kg/m^3

$$\mathbf{\sqrt{6.417 \times 2/1.2} = \underline{3m/s}} \qquad (1b:7)$$

So far we have the height of the stack to maintain a given velocity of 3m/s however the volumetric flow rate of air is required before the area of the stack and diameter can be determined.

The volumetric air flow rate for this example has been calculate from the ventilation requirements for the swimming pool area which is 6.67m^3/s (calculation details can be seen further in this section).

To find the cross sectional area the continuity equation is used and is expressed as

$$\mathbf{Q = V \times A} \qquad (1b:8)$$

Where;
Q= Volumetric flow rate m^3/s
V = Velocity m/s
A = Area m^2

Transposed for **A = Q/V** (1b:9)

6.67/3 = **__2.22m^2__**

To calculate the diameter from A = π r^2 transposed for the radius

r =2√2.2/π = __1.67m__ (1b:10)

To summarise so far, a stack ventilation system designed to extract a volumetric air flow rate of 6.67m^3/s at a velocity of 3m/s requires a stack height of 23m and an internal diameter 1.67m
To maintain the velocity of 3m/s a temperature differential of 6K is required to achieve internal stack buoyancy when the outside air temperature is in excess of 28^{0}C.

__Supplementary heating__

There will therefore be a requirement for supplementary heating for the inside of the stack to maintain 3m/s and the 6K differential when the outside air is in excess of 28^{0}C.

To calculate the heat flux required the mass flow rate is first determined
(ρ = density @ 28^{0}C =1.2kg/m^3)

6.67m^3/s x 1.2kg/m^3 = __8 kg/s__ (1b:11)

The heat flux to maintain 34^{0}C internal stack temperature with external ambient temperature of 28^{0}C,

Q=M Cp Δt (1b:12)

Where
Cp = specific heat capacity of air 1.005 kJ/kg/k
M = mass flow rate of air 8kg/s
Δt = Temperature difference ^{0}C

8 x 1.005 x (34-28) = __50Kw__ (1b:13)

Supplementary primary heat (Solar Collectors)

The heat energy to maintain high efficiency solar collectors will generate the temperature differential. The analysis of which will be described in a following chapter.

In order to proceed with the heat exchanger analysis the flow and return temperatures of the solar collectors are taken as 120^0C[23] flow and 105^0C[23] return.

The mass flow rate passing through the solar collectors is therefore found from the expression,

$$\mathbf{M = \frac{Q}{Cp\ \Delta t}} \qquad (1b:14)$$

Where;

M = mass flow rate kg/s
Q = Heat flux Kw
Δt = Temperature difference (flow and return)

50/4.2 x (120-105) = **0.793 kg/s**

4G) Heat Transfer analysis

The heat transfer from the solar collectors will take place via a single coiled copper tube inside the stack with a notional diameter of 42mm to promote gravity flow.

To calculate the heat exchanger surface area and length the internal and external heat transfer coefficients along with the heat transfer coefficient through the tube need to be analysed.

The expression for calculating the heat transfer rate in Kw is found from the formula;

$$\mathbf{Q = U\ A\ \Delta t\ (lmtd)} \qquad (1b:15)$$

Where;

Q = Heat transfer rate in Kw
U = Overall heat transfer coefficient W/m^2K
A = Area of the heat transfer surface m^2
Δt lmtd = Log mean temperature difference

and the overall heat transfer coefficient 'U' is found from the expression;

$$\mathbf{U = 1/1/hi + (ln\ (r2/r1/2\pi kl)) + 1/ho} \qquad (1b:16)$$

Where;

hi = inside heat transfer coefficient W/m^2k
ho = outside heat transfer coefficient W/m^2k
k = Thermal conductivity of copper kW/mK
l = Length m
r1 = internal radius m
r2 = external radius m

In order to calculate the overall heat transfer coefficient the inside and outside heat transfer coefficients need to be ascertained first, these coefficients require the correct application of fluid flow from the dimensionless numbers Reynolds, Prandtl and Nusselt.
They are derived as follows;

For turbulent flow inside tubes;

$$\mathbf{Nu = 0.023(Re)^{0.8}(Pr)^{0.4}} \qquad (1b{:}17)$$

For forced convection over pipes;

$$\mathbf{Nu = 0.193(Re)^{0.618}(Pr)^{1/2}} \qquad (1b{:}18)$$

The Reynolds number is found from;

$$\mathbf{Re = \rho\ v\ D/\mu} \qquad (1b{:}19)$$

Where;

Re = Reynolds number (dimensionless)
ρ = Density kg/m3
v = Velocity m/s
D = diameter
μ = Kinematic viscosity

Data

	^{0}C	**Kg/m3**	**M/s**	**Kg/ms**	**Kw/mk**
	Temp	**Density**	**Velocity**	**Viscosity**	**Conductivity**
Water	120	1000	0.55	$230\text{x}10^{-6}$	$687\text{x}10^{-6}$
Air	28	1.2	3	$1.846\text{x}10^{-5}$	$2.642\text{x}10^{-5}$

Table HT1

Dimensionless numbers results table

Prantle No	**Re No**	**Nusselt No**
1.42	100434	**265**
0.707	8033	**44.52**

Table HT2

From the above data in table HT1 the dimensionless Reynolds number is calculated for both water and air the two values are then used to calculate the Nusselt numbers in table HT2.

In both cases the Reynolds number is greater than 2000, which places the flow pattern in the turbulent region
This being the case the formulas for the calculation of the Nusselt values in (1b:16) and (1b:17) are confirmed as being correct.

The inside and outside heat transfer coefficients can now be calculated by using the expression,

$$\mathbf{hi = Nu.k/d} \quad \text{for the inside of the tube} \qquad (1b:20)$$

and

$$\mathbf{ho = Nu.k/d} \quad \text{for the outside of the tube.} \qquad (1b:21)$$

Where,

hi= inside heat transfer coefficient kW/m^2K
ho=outside heat transfer coefficient kW/m^2K
Nu= Nusselt number
k= thermal conductivity kW/mK
d= diameter of tube

The value of the overall heat transfer coefficient can now be calculated but first the wall thickness resistance needs to be calculated, however due to the small wall thickness in ratio to the pipe diameter this value can be negated.

For the purposes of investigation this value has been included and was derived from the expression,

$$\mathbf{Rw = \ln(ro/ri)/(2\pi.k\ l)} \qquad (1b:22)$$

Where,

Rw = wall resistance k/W
ro = outside radius m
ri = inside radius m
k = thermal conductivity of material Kw/Mk
l = length m

For the 42mm pipe diameter initially analysed this value is,

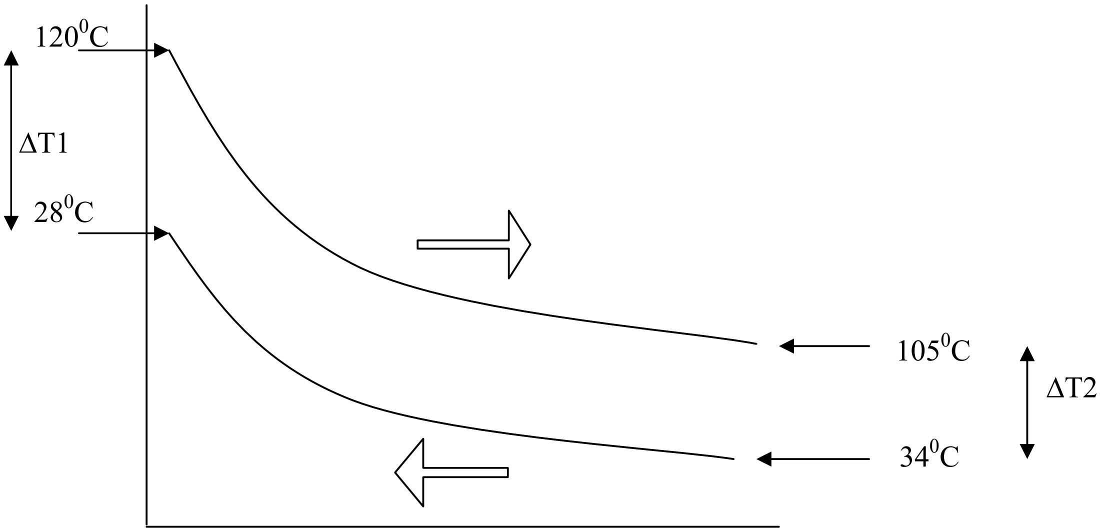
$$\boxed{\mathbf{3.88\ x10^{-5}\ K/W}} \qquad (1b:23)$$

The heat transfer equation, which is being employed,

$$\mathbf{Q = U.A.\Delta t(lmtd)} \qquad (1b:24)$$

Can be utilised once the log mean temperature difference has been calculated. The log mean temperature difference is a more accurate method of ascertaining the temperature difference across the heat exchange process and is derived as follows

Log mean temperature difference



The log mean temperature difference can be calculated by using the data from the above graph in the following expression,

$$\frac{\Delta T1-\Delta T2}{\ln \frac{\Delta T1}{\Delta T2}}$$

$$\frac{(120-28)-(105-34)}{\ln \frac{(120-28)}{(105-34)}}$$

$$\mathbf{Lmtd = 82^{0}C} \qquad (1b:25)$$

Now the log mean temperature difference has been calculated the area of heat transfer can be found by transposition of the heat exchange rate formula ie,

$$\mathbf{A = Q/U.Lmtd} \qquad (1b:26)$$

And from the area the tube length in meters can be calculated

$$\mathbf{L = A/\pi.D} \qquad (1b:27)$$

Where; L = length m
A = Area m^2
D = Diameter m

The table overleaf is a summary of the calculations for differing diameters of pipe, calculated heat transfer and length.
Table HT3 below gives the calculation results for a range of pipe diameters and pipe wall thickness. Note the varying values of **hi** and **ho** which effect the overall **U'** the 42mm diameter calculation line has been highlighted

	R out	R in	x	x/k	Re w	Re a	Nu w	Nu a	hi	ho	U'	mtd	area	ength	Q
15	0.0075	0.007	0.0005	5.49E-05	35869.57	2869.18	116.53	23.56	5336.9	41.22	40.81	82	14.94	317.03	50000
20	0.01	0.0093	0.0007	5.77E-05	47826.09	3825.57	146.68	28.15	5038.5	36.93	36.59	82	16.67	265.25	50000
25	0.0125	0.0115	0.001	6.64E-05	59782.61	4781.96	175.35	32.31	4818.6	33.91	33.60	82	18.15	231.04	50000
35	0.0175	0.0163	0.0012	5.65E-05	83695.65	6694.75	229.51	39.78	4505.0	29.82	29.58	82	20.61	187.48	50000
42	**0.021**	**0.0195**	**0.0015**	**5.9E-05**	**100434.78**	**8033.69**	**265.55**	**44.53**	**4343.7**	**27.82**	**27.60**	**82**	**22.10**	**167.46**	**50000**
50	0.025	0.023	0.002	6.64E-05	119565.22	9563.92	305.30	49.59	4194.8	26.03	25.82	82	23.62	150.34	50000
65	0.0325	0.03	0.0025	6.37E-05	155434.78	12433.10	376.60	58.32	3980.4	23.54	23.37	82	26.09	127.77	50000
80	0.04	0.037	0.003	6.2E-05	191304.35	15302.28	444.65	66.31	3818.5	21.75	21.60	82	28.23	112.34	50000

Table HT3

The graphical data in table HT4 below shows the near straight line relationship between overall heat transfer and pipe length

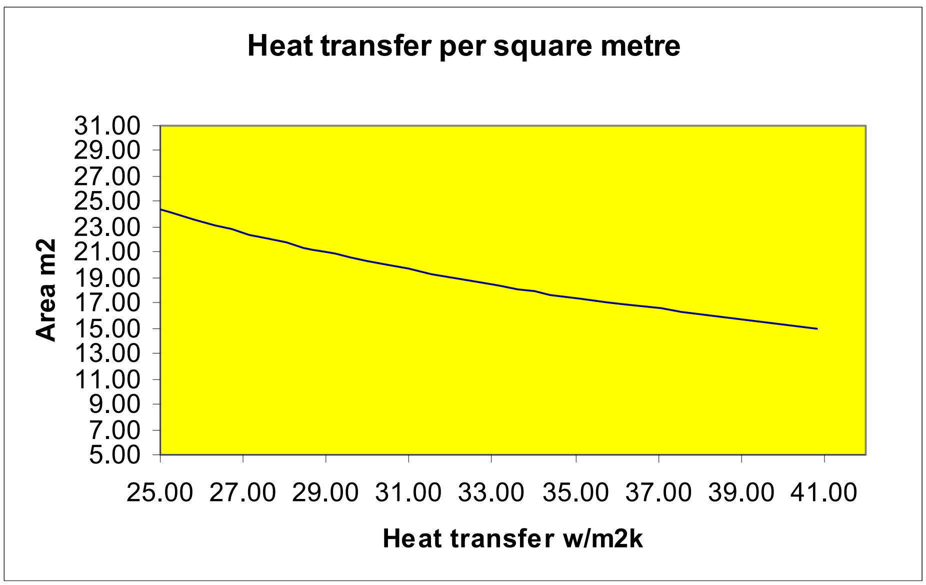


Table HT4

In conclusion of this analysis it can be seen by calculation that the heat transfer rate and the area of the pipe material are linked in an almost linear relationship.

After further analysis of the hydraulic area and thickness area ratios of the range of pipe diameters it appears there is no optimum diameter.
Throughout the whole range the average ratio is 6:1 which indicates that as the diameter increases the wall thickness increases in the ratio of 6:1.
This maintains the integrity of the pipe as its magnitude increases throughout the whole range of diameters.

The practical selection of the pipe size should therefore be based on the ease of forming the coil shape and the ability to occupy the interior of the whole 23m of height to maintain the heat transfer for buoyancy. The diameter of 42mm has been selected as being a manageable and cost effective.

4H) Solar collector analysis

Solar fraction

The solar fraction is the amount of available heat supplied to the system by the solar installation when compared to other heating systems, which are, integrated for example the biomass system.
It is important to fraction the amount of solar energy so that during periods of high solar irradiance the systems do not overheat and cause dangerously high temperatures or damage.
The use of a thermal store to dump excess heat is a requirement and the proposed system will be utilising a storage capacity based upon data from Viessman Ltd[5] which suggests 120-150ltrs[5] of storage per m^2 of collector area details of which follow below.

System dimensioning

The factors affecting the size of the solar thermal system to be employed are,

1) The local climate conditions (available solar radiation)
2) Heating/hot water demand
3) The solar fraction required
4) The efficiency of the system used

The system to be designed for the sports centre has an available roof area for collectors of $2500m^2$ and would be classified as a large solar installation. With a collector efficiency of 85%[5] the maximum heat output will be approximately 2120kw for the total roof area.

Table ST1 is a list of heat requirements and collector sizes for the whole building

Location of system	Heat output	Solar fraction	Solar output	Efficiency %	Collector area	Total area
Pool area	***400kw***	***60%***	***240kw***	***85%***	***$282m^2$***	***$282m^2$***
U F heating	***105kw***	***40%***	***42kw***	***85%***	***$48m^2$***	***$330m^2$***
Stack system	***164kw***	***100%***	***164kw***	***85%***	***$192m^2$***	***$522m^2$***
Hot water	***98kw***	***60%***	***58.8kw***	***85%***	***$69m^2$***	***$591m^2$***

Table ST1

From the table ST1 it can be seen that a total collector area of $590m^2$ would be required to achieve the solar heat outputs desired and over the range of solar fractions this gives an average solar fraction of 53%.

The size of the solar storage tank for inter-seasonal use is based on the area of collectors, which is 591m^2 (minus the stack system, which will operate independently,) that gives a collector area of 398m^2.

Table ST2 below gives details of the size of the solar storage vessels, which will be split into multiple units to keep thermal losses to a minimum.

Collector area m^2	**Storage factor ltr/m^2**	**Total storage ltrs**	**Volume m^3**	**Cylinder vol m^3**	**Dimensions m**	**Number**
398	***130***	***76700***	***77***	***25.73***	***3.2dia x 3.2high***	***3***

Table ST2

The storage cylinders will be heavily insulated to prevent losses and help to provide a buffer for days when the solar irradiance is low for protracted periods of time.

Evacuated tube collectors

Evacuated tube collectors are of a single tube construction, which houses a coaxial heat exchanger pipe, containing an absorber plate of coated copper.
Between the coaxial heat exchanger pipe and the outer tube exists a sealed vacuum, (like a thermos flask) this eliminates the convection losses between the glass tube and absorber and give the evacuated tube an efficiency of 85%[5] with very low heat losses. Typically evacuated tubes are arranged in arrays which form a given output per m^2, the tubes can be replaced individually if damaged.

System integration

There are 4 main areas, which make up the solar systems and they are as described previously.

a) The swimming pools

b) Domestic hot water supply

c) Under floor heating

d) Ventilation stacks

The general integration of the systems can be seen from diagram ST4[6]

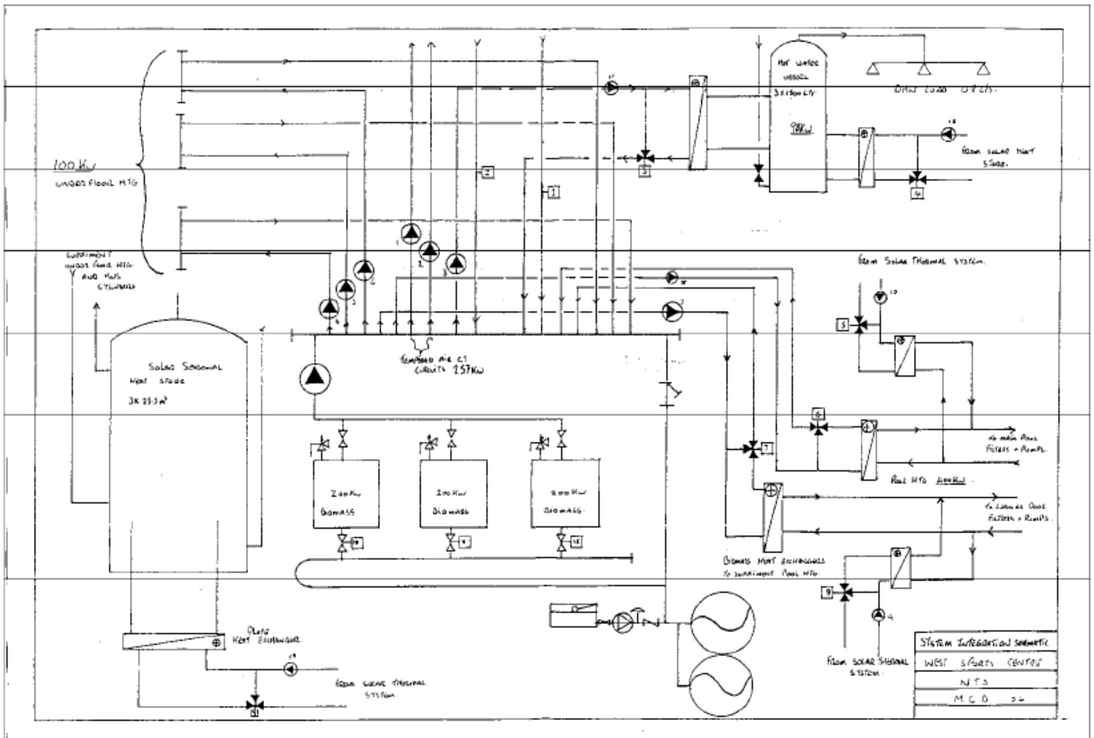


Diagram ST4[6]

Solar Tube Efficiency analysis

The solar panels, which, have been selected, are of the evacuated tube type and in principle have a very low convective almost negligible heat loss.

The manufacturers of these collectors have through bench testing provided empirical data to support the collector efficiency possible, the analysis of which is provided below.

Instantaneous Energy Collection formula;

$$X = \frac{(Tin + Tout)/2 - Ta}{1h} \quad (S1:6)$$

$$Y = 0.86 - 2.32 \times X \quad (S1:7)$$

Where,

Y = instantaneous energy collection efficiency
Tin = Collector inlet temperature ^0C
Tout = Collector outlet temperature ^0C
Ta = Ambient temperature ^0C
1h = Solar radiation of an incline with an absorber tilt angle Kw/m^2 hr

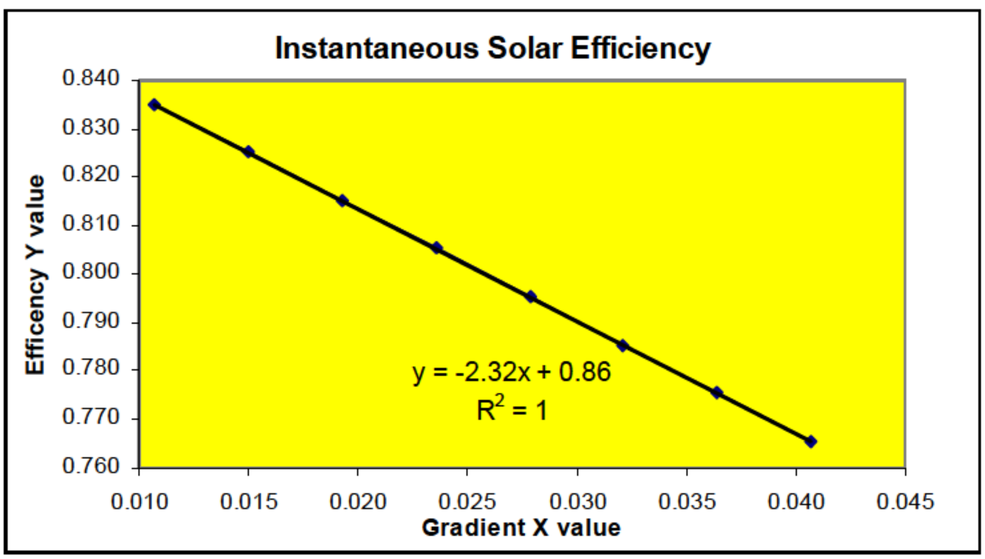


Chart ST5

Tin	Tout	Tin+Tout	Average	Ambient	Solar Int	min amb	X val	Y val
12	27	39	19.5	12	700	7.5	0.01	0.84
17	32	49	24.5	14	700	10.5	0.02	0.83
22	37	59	29.5	16	700	13.5	0.02	0.82
27	42	69	34.5	18	700	16.5	0.02	0.81
32	47	79	39.5	20	700	19.5	0.03	0.80
37	52	89	44.5	22	700	22.5	0.03	0.79
42	57	99	49.5	24	700	25.5	0.04	0.78
47	62	109	54.5	26	700	28.5	0.04	0.77

Table ST6

From the graph and the results table on the previous page a straight-line relationship exists. The values of the X-axis on the graph are derived from the first expression under the instantaneous energy collection formula.

The value of X is then used in the straight-line equation formula to give the results, the value of X is the gradient of the line in the graph.

The load value to the collectors being the temperature difference between the inlet temperatures and outlet temperatures does not have a bearing on the tube efficiency as a load of 15Deg C has been used throughout the complete range and the tube efficiency still falls away at the higher ambient temperatures.

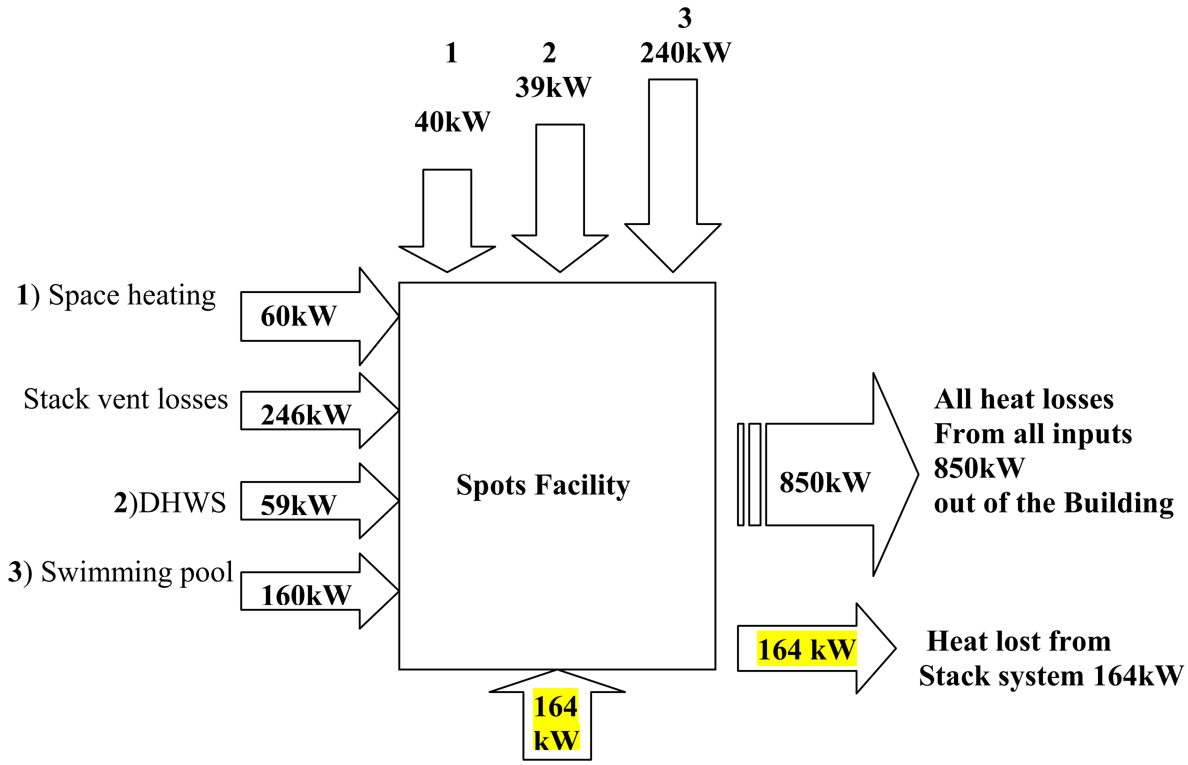
This shows that as the ambient temperature increases there is a slight decrease in the solar tube efficiency. This is principally due to the lower heat transfer rates at higher ambient temperatures.

4I) Building thermal response and boiler sizing

Building energy balance of heat producing equipment

Solar Inputs

1= Space heating
2= DHWS
3= Swimming pools



Total 1013Kw

Stack ventilation coil heating

Location	Heat input kW	Solar %	Solar output kW	Efficiency %	Collector area m^2	Biomass output kW
Heating	*105*	*40*	*42*	*85*	*48*	*60*
Tempered air	*246*	*Nil*	*Nil*	*85*	*Nil*	*246*
Swimming pools	*400*	*60*	*240*	*85*	*282*	*160*
DHWS	*98*	*60*	*59*	*85*	*69*	*59*
Stack system	*164*	*100*	*164*	*85*	*192*	*Nil*
Totals	**1013**		**505**		**591**	**528**

The Stack Ventilation heating system has not been included in CO2 mitigation for this project and has therefore not been included in the calculations and has been omitted from being classed as a heat loss from the building.

The total heat loss figure used in calculations is therefore 850kW

There are several loads, which contribute to the overall boiler plant size and these are,

- Heating………………105Kw – 40%solar contribution = **60kW**
- Tempered air…………246kW……………………….. = **246kW**
- Hot water……………98kw – 60% solar contribution = **59kW**
- Swimming pools……..400kw -60% solar contribution = **160 kW**

(Without solar thermal) Total **850 kW** (With solar thermal) Total= **528 kW**

The above areas without a solar contribution would give a plant size of 850Kw the 525Kw represents 62% of which the solar collectors contribute 38% overall.

To assess the buildings thermal response with respect to temperature decay from plant shutdown to start up an analysis of the cooling of the building and thermal capacity needs to be examined.

Thermal Response

The period of time the plant is shut down is termed as the unoccupied period. The required temperature when the building is to be occupied should be the same as the heating set point and the amount of energy expended in reaching this set point when the plant is started and the time this takes needs to be ascertained.

The design heat loss has been calculated from the heat loss spread sheets and found to be;

Fabric/Infiltration
105kW

The tempered air system has not been included in the following analysis. The ventilation system would be non operational during the unoccupied time with the exception of the pool system which would operate at 10% of its maximum.

In order to determine the optimum installed plant size an analysis of plant ratio size against re-heat energy will be ascertained.
The life cycle costs of the plant selection can be placed against the savings made in terms of kWh.
There are possible energy savings for a larger plant size and a shorter pre-heat period but this depends upon the buildings thermal capacitance.

Thermal mass of a building can have a great effect on the energy performance. For instance undersized sized plant will run longer, whereas oversized plant may have poor part load performance for much of its life.

Thermal capacitance 'C'

The calculation of the parameter for thermal capacitance is described in BS EN 13790. 2004.
For the calculation process the first 30mm of the material structure of the inner leaf elements of the building ie walls, floors, roof slabs etc in the same way that admittance is calculated. The results of the calculation are summarised in ***table B1a***

$$\boxed{\mathbf{C = \Sigma Cp\ x\ \rho\ x\ V}} \qquad \text{(T:1)}$$

Where; C = Thermal capacitance kJ/K
Cp = Elemental thermal capacitance kJ/kg/K
ρ = Density of elemental mass kg/m^3
V = Volume of elemental structure m^3

Overall heat transfer coefficient 'U'

The calculation of this parameter is derived from the heat loss elements including the ventilation losses. The results again can be found in ***table B1a.***

$$\boxed{\mathbf{'U' = \Sigma AU + 0.33NV/1000}} \qquad \text{(T:2)}$$

Where;
'U' = Overall heat transfer coefficient KwK
ΣAU = Summation of areas and Uvalues W/mK
0.33NV = Ventilation loss W/mK

Time constant 'τ'

The building time constant can be derived from the division of the building thermal capacitance 'C' by the overall heat transfer coefficient 'U' this value is in hours and is indicative of the buildings thermal response.

$$\boxed{\mathbf{\tau = C/'U' x\ 3600}} \qquad \text{(T:3)}$$

Where; τ = time constant hrs
'U' = Overall heat transfer coefficient KwK
C = Thermal capacitance kJ/K

To calculate the buildings thermal response the following data has been calculated and is summarised below in table b1a

	Parameter	Formula/Calculation	Value
1	Set point to be maintained		**20^0C**
2	Hours of occupation	06:00hrs to 22:00hrs	**16hrs**
3	Unoccupied hrs	24-16	**8hrs**
4	Thermal capacitance 'C'	(ΣCp x ρ x V)	**6.7×10^5kJ/K**
5	Overall heat transfer coefficient U'	ΣAU+0.33NV/1000	**5kW/m^2K**
6	Building time constant 'τ'	C/'U'x3600 $6.7\text{x}10^5$/5x3600	**37.22hrs**

Table B1a

Intermittent heating and Newtonian Cooling

In order to complete an analysis of the buildings thermal response and to ascertain the energy required bringing the building back up to the occupancy set point temperature the rate at which the building cools needs to be calculated.

The rate at which a body or a building cools with time is expressed as

$$\boxed{C\, d\theta i/dt = U'(\theta i - \theta o)} \qquad \text{(T:4)}$$

Where;

C = Heat capacity of the structure
U'= Heat loss coefficient

This can be more conveniently expressed after integrating over time as;

$$\boxed{t2 - t1 = -C/U'\ln[(\theta_2 - \theta ao)/(\theta_1 - \theta ao)]} \qquad \text{(T:5)}$$

Where;

θ_2 = plant switch on time
θao= Outside air temperature
θ_1= Set point temperature

From which the temperature of the structure can be determined for any time interval. It will be therefore found that when an object cools at constant external temperature, the drop in temperature during the time interval equal to the time constant is 63% of the full final drop.

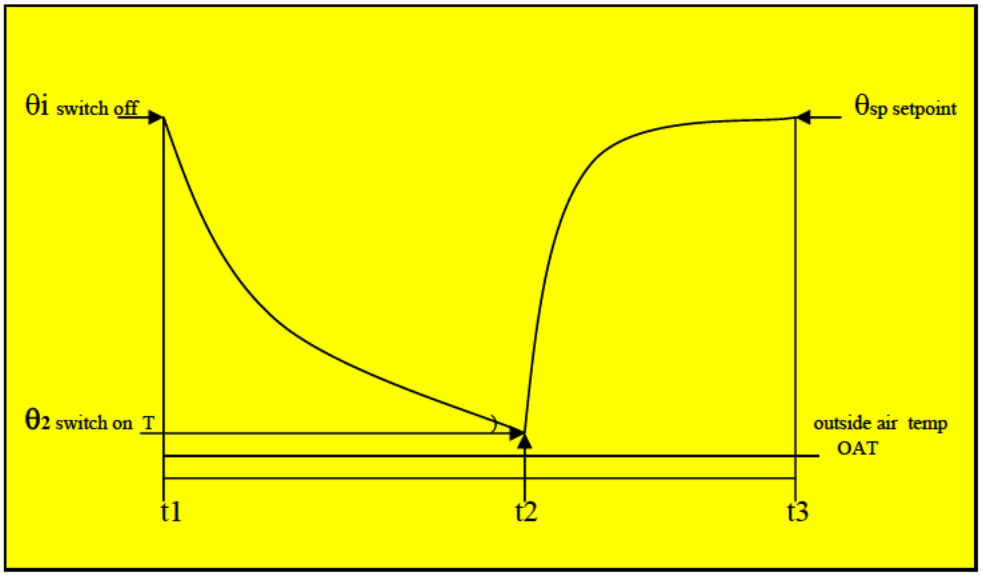
The equation, which will be used to access the rate of cooling, will therefore be;

(T:6)

$$\boxed{t2 - t1 = -C/U'\ln[(\theta_2-\theta ao)/ (\theta_1-\theta ao)]}$$

However to calculate this the value of θ_2 needs to be found first which is the temperature at which the cooling rate has reached a level where in order to reach the set point within the time period left until occupancy the plant will need to be switched on.

Cooling curve and pre-heat curve of the building



DWG B1

The following expressions will be used to analyse the optimum start for the building that, will then give the basis for the pre-heat energy required and ultimately the selection of size of plant.

The calculation of θ_2 is the first stage of the calculation process to enable the switch on temperature point at time t2 to be ascertained. Once this factor has been calculated both the cooling curve and pr-heat curve can be ascertained.

θ_i °C	θ_o °C	Qp kW	U' kW/K	C kJ/K	C/U' hours	Unoccupied period hours
20	-1	126	5	670000	37.2	8
	Times 1.5	158				
	Times 2	210				

Data table 2

(T:7)

Calculation of θ_2 = **e^-(unocc/τ)QP(θsp-θao)/QP-U'(θsp-θao)[1-e^-(unocc/τ)] +θao**

Calculation of θ_2 = $$\frac{e^{-(8/37.2)} \times 126 \times (20-(-1))}{126 - 5\,(20-(-1)) \times [1-e^{-(8/37.2)}]} + -1 = \underline{19.2^{0}C}$$

Cooling t2-t1 = **$t_2 - t_1 = -C/U' \ln[(\theta_2-\theta ao)/(\theta_1-\theta ao)]$** (T:8)

$-37.2 \ln[(19.2-(-1))/(20-(-1))] = \underline{1.4hrs}$

Pre –heat t3-t2 = **$t_3-t_2 = -C/U' \ln[QP-U'(\theta_1-\theta ao)/ QP-U'(\theta_2-\theta ao)]$**

$-37.2\ln[126-5(20-(-1))/126-5(19.2-(-1))] = \underline{6.6hrs}$

The design heat loss of 105Kw has been increased by a factor of 1.2,1.5.and 2 to give the figures in column 3. The figures in the other columns are the parameters as previously calculated

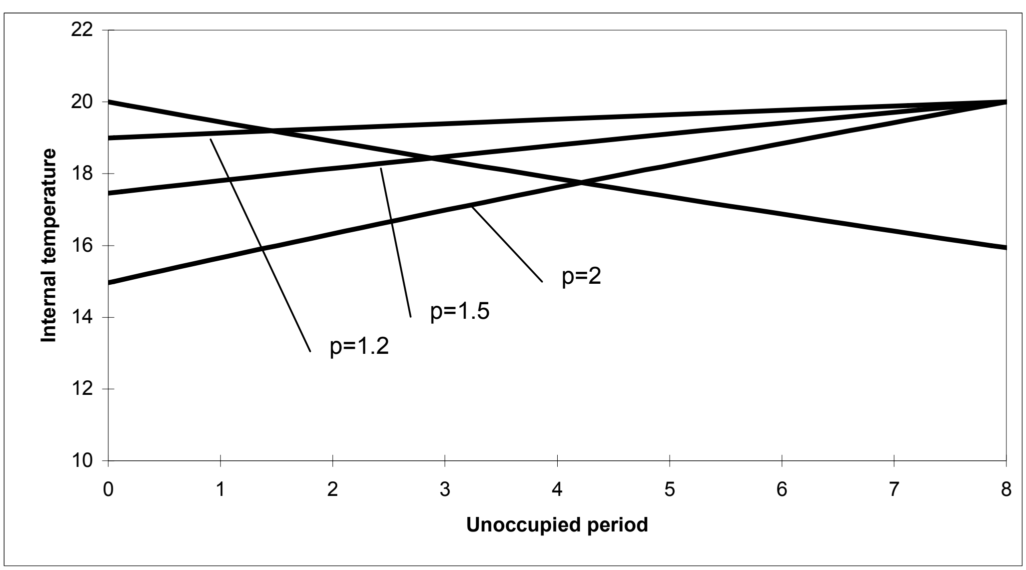


Chart 1b3

θ	t3-t2	t2-t1
19.2	***6.6***	***1.4***
18.4	***4.3***	***3.7***
17.8	***3.9***	***4.1***

Table 1b4 Summary

In chart 1b3 it can be observed that the building will cool from the set point over an 8-hour period to a level of 16^0C without any heating.

The intersection point where line P=1.2 crosses the cooling curve is the switch on temperature point when the plant is increased from the design heat loss by 20%.

The points P=1.5 and P=2 are plant size increases of both 50% and 100% respectively, the results of which are summarised in table 1b4.

It can be seen in table 1b4 that the switch on temperature (θ) reduces as the plant ratio increases enabling a longer plant shut down period with a quicker pre-heat period up until the occupation time set point.

To summarise table 1b4 when the plant ratio is increased by 20% above the design heat loss, the switch on temperature after plant shut down at a set point of 20^0C is 19.2^0C and this will be 1.4 hrs after plant shut down and take 6.6hours to reach set point temperature at time of occupation.

Alternatively when the plant is 100% above the design heat loss the switch on temperature is lower at 17.8^0C and will be 4.1hrs after plant shutdown and take 3.9 hrs to reach set point temperature at time of occupation.

4J) Boiler sizing financial savings

To consider cost savings the capital and marginal cost of the boiler plant, the discount % rate over the life of the plant, fuel costs and length of heating season need to be ascertained and analysed to indicate the optimum plant size or plant ratio against net discounted savings.

Design heat loss kW	No. of heating days	Discount rate	Cost of gas p/kWh	Years	Marginal Cost of plant £/kW
105	**200**	**10%**	**2**	**10**	**7**

Table 1b5

The figures used in table 1b5 have been calculated or are typical figures at the time of writing this report.

For example the cost of gas is currently 2p/kWh the highest ever price for gas in the UK. The discount period of 10 years is a typical figure for boiler plant as is the marginal cost of plant. The rate of discount at 10% is a conservative figure and the number of heating days matches the building use pattern.

θι	θo	U'	q	q/U'	Unoccupied period	Design θo
°C	°C	kW/K	kJ/K	hours	hours	°C
20	-1	5	670000	37.2	8	-1

Table 1b6

Table 1b6 comprises of the data, which will be used to calculate the optimum plant ratio which is presented in tabular and graphical form below in table 1b7 and chart 1b8 a.

H	Plant ratio	θopt	t2-t1 opt	t3-t2	Preheat energy
kW					kWh
105	1.00	20.0	0.0	8.0	840
125	1.19	19.2	1.4	6.6	825
145	1.38	18.7	2.4	5.6	814
165	1.57	18.3	3.1	4.9	807
185	***1.76***	***18.0***	***3.7***	***4.3***	***801***
205	1.95	17.8	4.1	3.9	796
225	2.14	17.6	4.5	3.5	792
245	2.33	17.5	4.8	3.2	789
265	2.52	17.3	5.0	3.0	786
285	2.71	17.2	5.2	2.8	784
305	2.90	17.1	5.4	2.6	782
325	3.10	17.1	5.6	2.4	781
345	3.29	17.0	5.7	2.3	779

Table 1b7

Table 1b7 indicates that as the plant size increases the pre heat energy reduces over the preheat period.

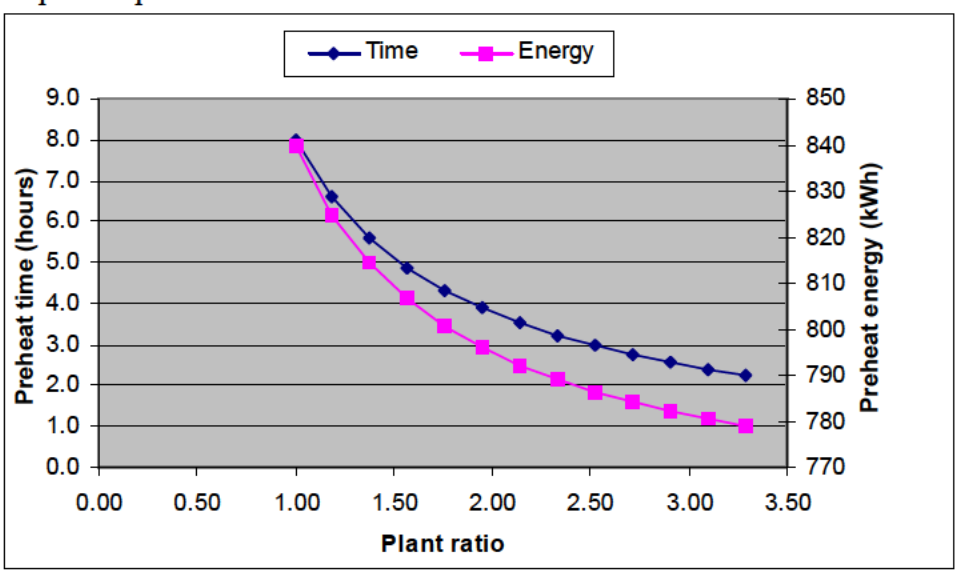


Chart 1b8

Saving	Annual	Life time discounted saving	Extra cost of plant	Total
kWh	kWh	£	£	
0	0	0	0	
15	3031.3	372.5	140	-232.5
26	5136.0	631.2	280	-351.2
33	6682.9	821.3	420	-401.3
39	***7868.1***	***966.9***	***560***	***-406.9***
44	8805.1	1082.1	700	-382.1
48	9564.6	1175.4	840	-335.4
51	10192.7	1252.6	980	-272.6
54	10720.8	1317.5	1120	-197.5
56	11170.9	1372.8	1260	-112.8
58	11559.2	1420.5	1400	-20.5
59	11897.7	1462.1	1540	77.9
61	12195.2	1498.7	1680	181.3

Table 1b9

It can be seen from table 1b9 and chart 1b10 that the optimum plant ratio size is 1.8 and is highlighted on table 1b7 as 187kw

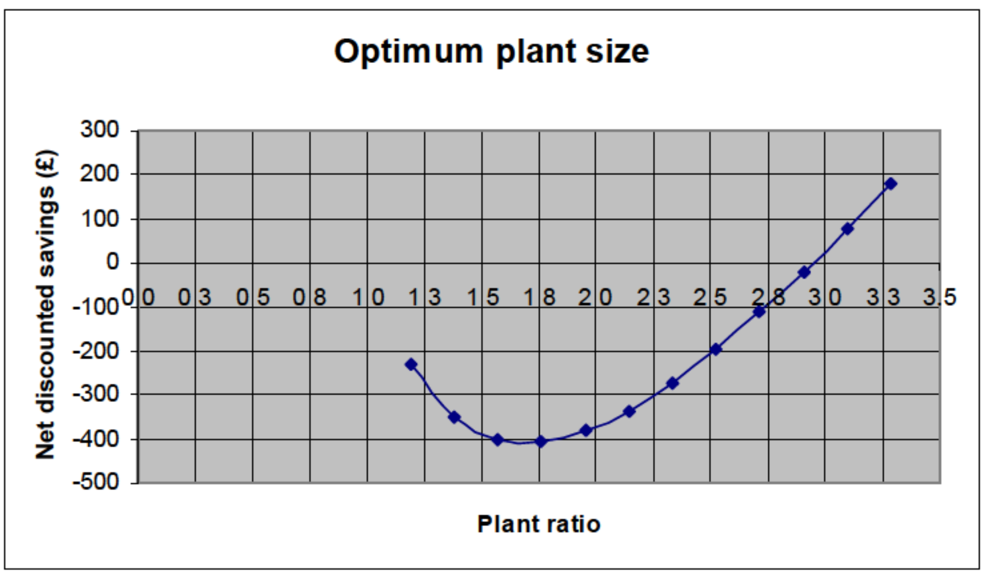


Chart 1b10

Conclusions

In conclusion of this analysis plant size is influenced by the building thermal capacity along with boiler capital costs, fuel costs and length of the unoccupied period.

The results show that plant ratios over 2 (100% larger) are not justified for this building or any other typical building with a similar thermal capacity.

It is possible that boiler efficiency will decrease for bigger plant ratios this depends on the modular nature of the plant installed, for example a small number of large boilers would compromise efficiency as opposed to several smaller boilers.

The heating requirement for this building at 105kw design heat loss is about a third of the overall boiler plant requirement as other services such as the swimming pool, domestic hot water system and tempered air supply make up the majority of the plant load.

The extra capacity therefore proved to be required for the unoccupied heat up period therefore already exists but for the purposes of analysis knowledge of the impact on the overall system demonstrated that the plant will meet expectations.

4K) Underfloor heating analysis

The building has quite a low level of heat loss due to the construction materials and U values employed and with this in mind several different types of heating system have been considered.
In order to choose the most effective system type the advantages and disadvantages have been considered a table H1[12] gives a summary of the main points taken from **BSRIA AG 12/2001**[12]**.**

System type	Advantages	Disadvantages
Electrical convector heating	Very responsive Low capital cost Flexible in installation Can be cost effective if infrequently used	Can be expensive to operate High production of Co_2 generation
All Air heating	Can be used to provide cooling Fairly flexible Can meet heating and ventilation requirements	May require large plant area High maintenance costs High capital costs
Warm air heating	Quick warm up times Less obtrusive than radiators	Can blow dust about Can have excessive losses through ductwork May loose part of loft space to accommodate ductwork Service engineers can be hard to find
Radiator heating	Quick response when required Can be replaced easily and cheaply Can be retrofitted with some ease	Local hot spots System can become scaled up in time without inhibitor. Convection currents aid the circulation of dust and mites Take up wall space and reduce flexibility.
Underfloor heating	Even temperature distribution Increased flexibility for furniture Reduced running costs Healthier Vandal resistant Reduced convection currents Less obtrusive than other methods	Poor response times Not very flexible Sensitive to certain floor coverings Floor penetrations should be avoided Any leaks disruptive and expensive to rectify

Table H1[12]

Although underfloor heating has some negative points it has been selected on the basis of low energy input to the building and the fact that the system will be out of sight and not impose its self on the spaces being used for sports activities.

Knowledge of when to use underfloor heating is summarised in the table H2[12] below, which is an extract from **BSRIA AG 12/2001**[12].

Suitable for	Not suitable for
Most housing applications	Buildings or areas that are used very Intermittently or infrequently
Buildings or areas with low heat loss	Buildings or areas with high heat losses or sudden heat losses
Buildings or areas that are continually or frequently used	Applications where the floor area has large amounts of equipment fixed into the floor
Buildings or areas with high ceilings	Buildings where future portioning or internal wall changes may occur

Table H2[12]

There are several design aspects, which have to be considered when selecting under floor heating as an option and they are listed as bullet points below.

a) If the height of the internal rooms is considerably high only the occupied zone height should be used to calculate heat losses.

b) Space temperatures due to radiant heating effect of underfloor heating can be 2K[12] lower than convective heating system for the same comfort conditions.

c) Only specified plastic pipe work should be used for underfloor heating as per BS 7291 part 1-3.

d) A maximum downward loss of 10%[12] should not be exceeded and insulation should be selected to achieve this.

e) In areas where the space temperature is higher than 20^{0}C floor temperatures should not exceed 29^{0}C[12].

f) The room volume should heat loss should only be considered in the occupied zone[12].

g) Depth of screed and surface finish factors should be considered in the calculation of underfloor heat outputs.

h) Pipe spacing should be calculated from the heat output from the floor and the temperature difference between mean water minus space temperature.

i) System water flow temperatures should not exceed 50^{0}C[12].

The design of an underfloor heating system should take into account the points above and further guidance on the subject can be found various British standard specifications and BSRIA.

The design parameters for the building are listed below for a fixed temperature system[12]

1) Water flow temperature 50^{0}C
2) Water return temperature 40^{0}C
3) Mean water temp 45^{0}C
4) System temperature differential 10K
5) Surface film coefficient $11W/m^2K$
6) Depth of screed 40mm
7) Depth correction factor 1.04
8) Surface correction factor for 10mm synthetic surface cover 1.3
9) Pipe spacing 300mm
10) Pipe material, cross-linked high-density polyethylene (PE-X)

Notes for the spreadsheets 1 and 2

Column

1) Net floor area m^2
2) Internal room apace temperature ^0C.
3) Design temperature between winter outside and internal space ^0C.
4) Volume of the internal space m^3.
5) Calculated ventilation loss of the building W/m^2K.
6) Calculated Fabric loss of the building W/m^2K.
7) Fabric loss with the floor element subtracted W/m^2K.
8) Downward heat loss W/m^2K.
9) The addition of the fabric loss and the ventilation loss minus the downward losses.
10) The output in W/m^2 after the floor area is divided in to the fabric and ventilation losses
11) The depth factor taken from tables relating to the depth of screed.
12) The floor finish factor from tables relating to the floor finish type and depth.
13) The output from column 10 multiplied by columns11 and 12 W/m^2.
14) Pipe spacing from selection chart of output against mean water temperature minus space temperature.
15) Floor surface temperature calculated from mean temperatures through the floor plus mean system temperature drop for flow and minus mean system temp drop for return.
16) The final output including the downward loses in W.
17) The calculated flow rate in kg/s.
18) Floor surface temperature minus internal space temperature^0C.
19) Same as column 10.
20) System temperature differential between flow and return ^0C.
21) The temperature drop through the screed element of the floor ^0C.
22) The temperature drop through the floor surface element of the floor ^0C.
23) The internal space temperatures ^0C
24) Flow temperature through the heating pipe work ^0C.
25) Return temperature through the heating pipe work ^0C.
26) Calculated floor surface temperature ^0C.

Spreadsheets 1&2 can be found in appendix 2 and have been used to calculate the temperatures and outputs required to select the pipe sizes, plant loads and pump duties.

The system will be controlled via outside air compensation with feed back from each individual space controlling a three port mixing valve arrangement as shown in diagram U1[6] below.

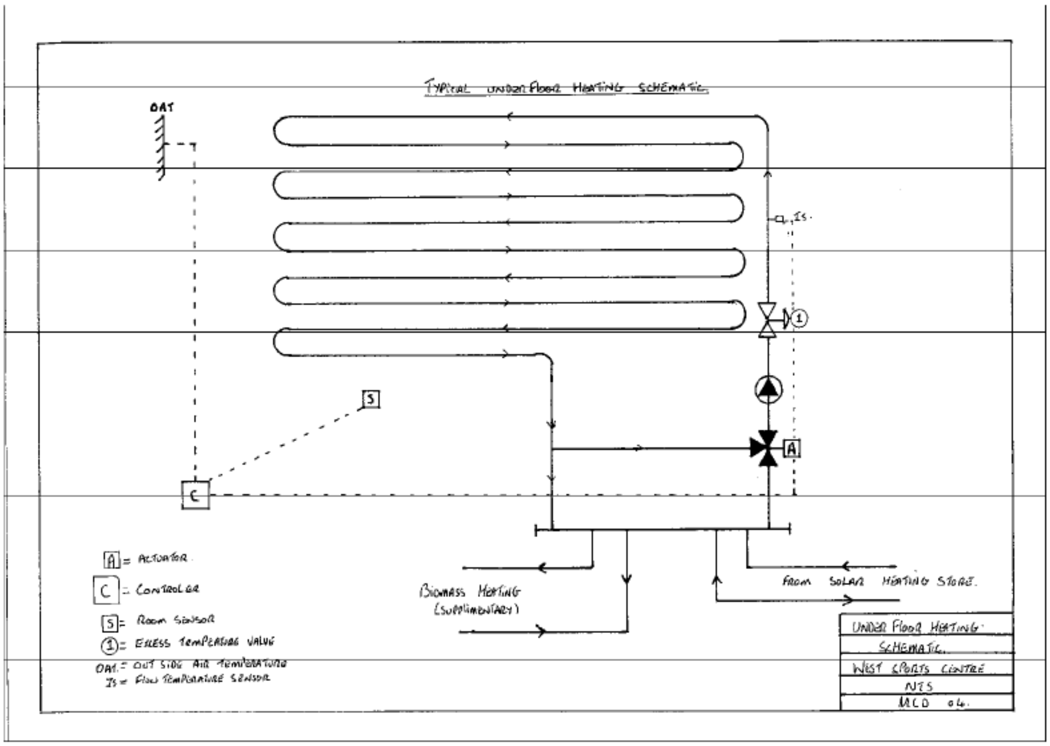


Diagram U1[6]

The outdoor compensation system operates to a scheduled ramp, which is shown in diagram U2[6]. It can be seen that when the outdoor temperature falls to zero degrees the flow temperature will be 45^0C similarly when the outdoor temperature falls to the design minimum of -5^0C the flow temperature will be 51^0C.

To protect the floor from becoming too hot in case of a mixing valve failure a two-port open/closed valve is fitted down stream of the mixing valve with a high limit sensor, which will close the valve at values higher than 55^0C.

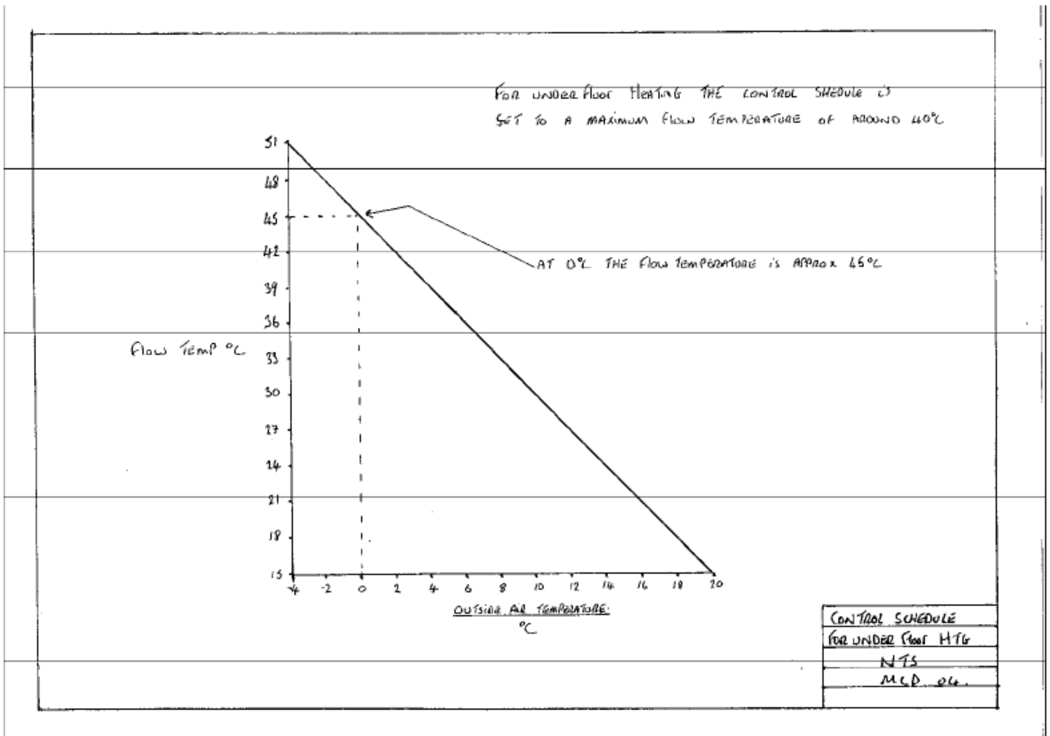


Diagram U2[6]

4L) Biomass Fuels

Introduction

The project being developed relies on the use of renewable energy for almost the entire energy requirement of the building. The varying technologies of energy sources such as Solar Thermal, Solar PV, Stack Ventilation, Underground air pipe supply and biomass are planned to be designed into this building.

The main boiler plant will be fired from biomass woodchip fuel, which is derived from short rotation crops such as Hazel and Willow saplings. The source of these supplies comes from commercially farmed biomass energy providers within the local area and wood management schemes.

A more detailed study into the development use and economics of biomass will now follow.

Short Rotation Coppicing

This method of utilising biomass requires the growing and harvesting of specific varieties of wood as the fuel source; particularly rapid growing plants, which have a good energy, yield.

The particular varieties commonly used in this country are Hazel and Willow, which are grown in plantations and managed just like any other commercial crop. There are fundamental issues, which have to be examined, and an appreciation of the soil stored carbon levels within root structures and how the conversion of carbon dioxide during photosynthesis needs to be considered.[11]

However, for this project study an in depth analysis of the carbon proliferation back into the soil is not required but an understanding that this takes place and directly affects the soil condition and ultimately the plant yields is enough.

It is estimated that a yield of 10 dry tonnes per year is possible from a 1 hectare[10] plantation which, is about 2.5acres. The site in question would therefore require approximately,

Fuel	Tonnes/ yr	Hectares	Acres
Hazel woodchip	1094.8	109	270

Costing and Economics

The capital cost of a biomass heating installation can be significantly more than the cost of conventional fossil fuel plant once all storage and feed mechanisms are accounted for. The table below gives a broad indication of the installed cost of woodfuel heating compared with that of a fossil fuel system.

Capital Cost per kW Installed[8]		
Energy Source	Conversion Technology	Cost per kW Installed
Gas or Oil Heating System (including train, controls, flue, plant room, pipe work, valves and insulation, pumps and pressurisation unit)	Combustion	£60-£115
Woodfuel Heating System (including controls, flue, plant room fitting, pipe work, valves, insulation and storage)	Combustion	£110-£265

Table BF1

Therefore whilst the concept of using wood fuel is often attractive, in the absence of grant funding, the initial increase in expenditure is often a disincentive to wood fuel installations, despite the fact that wood fuel can provide long-term fuel savings.

- There is grant funding now available to offset the initial cost increase associated with wood fuel systems.

Equally some wood fuel industry specialists have established energy Services Companies (ESCo's) which will put together a complete renewable heat or CHP package including the design, specification, installation, fuel supply and operation and maintenance of the system. Thereafter, the amount of heat used is metered and the ESCo charges the user for the kWh used, in much the same way as gas or electricity. This concept therefore avoids the user having to make a large financial purchase on a system.

Chart BF2 below highlights the different cost of heat given the cost of the fuel and its embedded energy. It highlights a cost range given the minimum and maximum price that could be expected for each of the different heating fuels in £/MWh. [13]

The chart clearly demonstrates the range of heating cost per megawatt-hour given the variability of fuel prices. The comparison is based on current price trends, but given the fluctuations in oil prices, it is hard to make long term cost comparisons.

It demonstrates that for all heating fuels, woodchip can deliver the cheapest heating cost per unit of heat output given a number of price scenarios. However, it is generally unable to compete with mains gas heating with fluctuating tariffs.

Equally, wood pellets are also able to compete with the delivered cost of heat from other heating sources such as LPG and electricity heating. It is even able to compete with oil, and especially so if consumers are able to secure an oil tracker deal such as those offered by some UK wood pellet suppliers. See chart1 below[14]

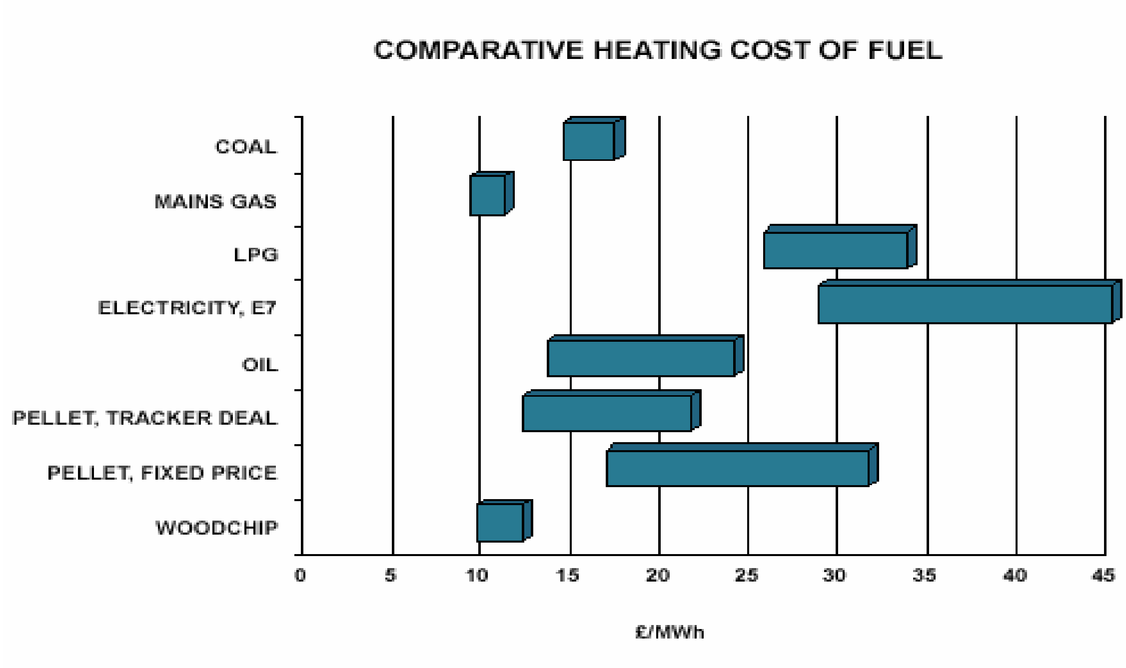


Chart BF2

The analysis is based on the following assumptions, which are concurrent with market trends. [8]

- **Coal - based on a price of £100-120 per tonne**
- **Mains Gas - based on a heating cost of 0.9-2.0p/kWh**
- **LPG - 20p-30p per litre delivered**
- **Electricity - Economy 7 off-peak (for storage heaters) - 2.9p to 4.5p per kWh; domestic supply is about 2.5 times the cost (£67 per MWh) and is not shown for reasons of scale.**
- **Oil - £0.14 to £0.30 per litre delivered, 35 sec.**
- **Pellets, fixed price - £80 to £150 per ton delivered, at an 8% moisture content**
- **Pellets, tracker deal - oil price as above less 10%, at an 8% moisture content**
- **Woodchip- £30 per gt at 60% moisture content to £40 per gt at 35% moisture content.**

To further analyse the possibilities of using woodchip as a fuel source direct comparisons with other fuels have been made in particular Natural gas and Gas oil, it can be seen from the table 1 below that by using woodchip as a fuel significant savings can be made as previously stated.

The unit costs per fuel type have been checked as being the current market values and it is worthy of note that the cheapest wood fuel is woodchip as opposed to pellets. As the moisture content varies between 60%to around 8% so too does the cost of the fuel.

The cost of woodchip currently is around 60%the cost of Natural gas which, with current trends is likely to increase along with price of oil. There will be an increase in the price of woodchip at around 8% per year.[13]

Fuel cost comparison table

Fuel	CV Mj/kg	Plant Size Kw	Mass flow Kg/s	Op hrs hrs	tonnes/yr	GJ per yr	Cost per unit of fuel £	Cost per GJ	Annual cost £
wood chips MJ/kg	15	528	0.0352	8640	1094.86	16422.91	35.00	2.33	38320
Fuel	**CV Mj/m3**	**Plant Size Kw**	**Mass flow m3/s**	**Op hrs hrs**	**m3/yr**	**GJ per yr**	**cost per unit of fuel £**	**cost per GJ**	**annual cost £**
Natural gas MJ/m3	39	528	0.0135	8640	421100.31	16422.91	0.30	7.69	126330
Fuel	**CV Mj/Ltr**	**Plant Size Kw**	**Mass flow Ltr/s**	**Op hrs hrs**	**Ltrs/yr**	**GJ per yr**	**cost per unit of fuel £**	**cost per GJ**	**annual cost £**
Oil 35 sec MJ/Ltr	45	528	0.0117	8640	364953.60	16422.85	0.33	7.33	120435

Table BF2

Kg of Carbon dioxide per GJ of fuel Table BF3

With the inclusion of solar thermal energy						
Fuel type	**Kw output**	**Fuel quantity**	**GJ per/t**	**GJ per yr**	**Co2 per yr tonnes**	**kg Co2 per GJ**
Nat Gas m3	528	421100.3	55	16422	**821.1**	50.00
35 sec oil ltrs	528	384611.52	42	16422	**1228.9**	74.83
woodchip tonnes	528	1094.8	15	16422	**1605.7**	97.78

Kg of Carbon dioxide per GJ of fuel Table BF4

Without the inclusion of solar thermal energy						
Fuel type	**Kw output**	**Fuel quantity**	**GJ per/t**	**GJ per yr**	**Co2 per yr tonnes**	**kg Co2 per GJ**
Nat Gas m3	850	677907.7	55	26438	**1321.9**	50.00
35 sec oil ltrs	850	619166.3	42	26438	**1978.4**	74.83
woodchip tonnes	850	1763	15	26438	**2585.0**	97.78

Savings in Carbon Dioxide per year

Fuel type	tonnes CO2 850 Kw	tonnes CO2 528Kw	CO2 Saving tonnes/yr
Gas	1321.9	821.1	500.8
Oil	1978.4	1228.9	749.5
Woodchips	2585.0	1605.7	979.3

Table BF5

The impact on the environment when fossil fuels are burned is in terms of the Carbon dioxide produced, which is a greenhouse gas and is contributing to global warming as many studies have indicated.

Tables BF3 & BF4 above indicate the level of Carbon Dioxide in tonnes per year and kg of CO_2 per GJ of consumed energy. Table BF3 shows the impact when solar thermal energy is integrated in all the heating making systems.

Table BF4 indicates the extra boiler power, which would be required to compensate for exclusion of solar thermal energy with the corresponding increase in Carbon Dioxide levels.

In table BF5 the savings in Carbon Dioxide in tonnes per year for the different fuels can be examined which is quite substantial at 38%.

It can be seen from table BF3 and BF4 that woodchip fuel has 97.78kg of CO_2 per GJ however this quantity of CO_2 produced by the burning of Woodchips is classed as **Zero** when the wood source is grown from sustainable crops.

This theory is supported by the fact that the same amount of CO_2 produced by the burning of wood fuel is absorbed into growing wood crops via photosynthesis and oxygen is produced the net effect being zero CO_2 to the environment.

This is not the case for Natural gas, Coal and petroleum products with the exception of bio diesel.

Conclusions on biomass woodchip

In conclusion the use of biomass woodchip as a fuel source is a viable option providing the associated installation costs are fully considered with regard to fuel storage and feed mechanisms.

The main overriding advantage of this technology is the CO_2 mitigation when the crops are grown in a sustainable way. The only drawback being high moisture contents which has the effect of reducing the heat into the boiler plant

Other wood products such as wood pellets on the other hand have maximum moisture content of 8%, which is reflected in the cost per kg.

The potential of this fuel source has great potential and if managed in a sustainable way should outlast fossil fuels.
Consideration is required for transportation distance if in excess of 25 miles as distances over this limit are less cost effective and sustainable.

5) Overall Conclusions

Synergy and integration at conceptual stages are absolutely paramount for the future success of good quality operationally successful buildings.
It is hoped that the systems, which have been analysed in this report, demonstrate in stages the principles of the engineering involved and how integration into a recreational facility could be engineered.

The depth of analysis on for instance the stack ventilation system should provide the evidence that the system is operationally viable as well as being original.
The analysis on the boiler plant sizing and optimisation however provide the in-depth knowledge of the mechanisms for correctly sizing plant by taking into account the building thermal response.

It has not been the intention of this report to give a complete design, but to focus on certain innovative renewable aspects, which could be employed in the construction of a recreational building and to demonstrate the technical engineering knowledge and analysis where appropriate to achieve renewable synergy, it is my hope that this has been the case.

6) Evaluation and Reflection

The lessons, which have been learned, are:

1. Synergy and integration of the building as a whole with regard to building services and renewable energy technologies is required at conceptual stages.

2. Biomass is a very viable method of producing heat energy whilst contributing to Carbon Dioxide reduction by being a Zero Carbon technology when grown in sustainable way.

3. The consideration of transportation costs if the site is in excess of 25 miles from the fuel site, as this is becoming unsustainable as transport fuel costs rise.

4. Solar thermal can provide a considerable proportion of heat energy depending upon collector selection, orientation and percentage contribution. A swimming pool site being the optimum as a summer heat sink with interseasonal storage.

5. Ventilation can be achieved without the use electrical fan power to ventilate all areas including the swimming pool.

6. Pre-cooling and pre-heating with the use of underground air pipes provides a valuable contribution to the reduction of energy consumption.

7. The thermal mass of the building structure plays a vital role in the thermal inertia of the building with regard to time lag and the requirement or not of intermittent heating.

With regard to evaluation the most prominent criteria is the comparative measure of the reduction in Carbon Dioxide.

The reduction in Carbon Dioxide is currently at the top of the agenda with regard to the building regulations part L2 and the Energy Performance in Buildings Directive.

In the context of this report the inclusion of 38% solar thermal plus the use of Biomass makes for a considerable Carbon Dioxide reduction when compared to the same building operating on Natural Gas or Oil around 500 tonnes of CO_2 per year.

Reflection

On reflection the ambitious use of Solar thermal and Biomass as renewable sources of energy for such a building would not be without drawbacks relating to the initial capital costs which has not been part of this study.

However a more sensible approach is now being adopted regarding renewable and sustainable technologies in the form of life cycle costing rather than simple payback over a traditional 2 to 3 year period.
As engineers this is a welcome break and will give opportunities for innovation to unfold.

In addition is the utilisation of biomass future proof and could it be sustained if extensively used throughout the UK. Questions such as this need to be carefully examined to prove viability.

7) <u>Appendices</u>

i) **Heat loss calculation spread sheets**

ii) **Under floor heating spread sheets**

iii) **Stack ventilation calculation**

iv) **Drawings and schematics**

v) **References and bibliography**

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