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# Experimental verification of a solar hot water heating system with a spiral-jacketed storage tank †

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**Abstract**

Simplifying solar thermal systems offers a number of potential benefits, including lower initial investment, lower maintenance costs, and reduced likelihood of operating faults. Reduced capital cost leads to increased competitiveness in the energy market. With appropriate care, a simplified solar thermal system design can maintain competitive energy performance with more complicated designs. We propose to simplify small-scale thermal storage systems by the use of a spiral-jacketed storage tank that combines the function of both the heat exchanger and storage tank. The new storage tank is designed and manufactured to maintain performance comparable to a conventional system, and its functional performance validated by retrofitting an existing system and operating it under real conditions over a multiple month period. The system retrofitted with a spiral-jacketed storage tank showed performance competitive with the previous system that utilized a typical storage tank and heat exchanger during a day with a good solar radiation but experienced somewhat diminished performance during a month that included cloudy days.

*Keywords*: Solar thermal energy; Heat exchanger; Storage tank; Spiral-jacket

## 1. Introduction

Most solar water heating systems in Korea are either a natural circulation type with a storage tank attached to the top of the collector or a forced circulation type with a heat exchanger for larger systems [1]. The natural circulation type is common for small systems, while a forced circulation type is generally used for large systems.

The primary disadvantage of the natural circulation type is that freezing temperatures are experienced in most regions of Korea. When the temperature falls below freezing, an electric heat trace is used to prevent water in pipes from being frozen. Apart from increasing the cost of operation, the electric heat trace is subject to failure due to frequent operational cycling. If the heat trace failure goes unnoticed, the system will freeze with the likelihood of creating mechanical failure of piping or other components.

A typical forced circulation solar system type has two piping loops: the primary circulation loop serving the collector units and a secondary circulation loop serving the storage tank [2] as shown in Fig. 2(a). While the forced circulation system avoids tank freezing with an anti-freeze solution in the primary circulation loop, the size of system becomes larger, more complex, and costly because a separate heat exchanger must be specified and installed. As a result, the possibility of system failure increases and initial costs are higher. An alternative to an external heat exchanger is the use of a coil-type heat exchanger installed internal to the storage tank. However, the increasing length of the coil results in additional system complexity as the system becomes larger. The

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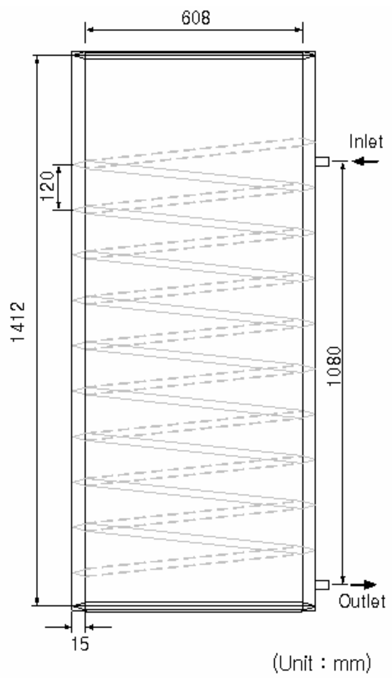
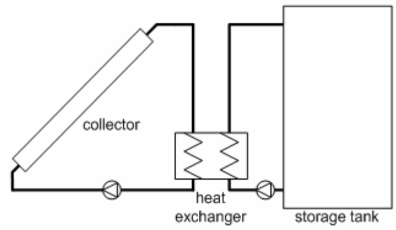



Fig. 1. Spiral-jacketed storage tank.

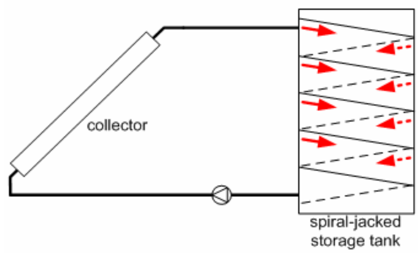
tank-in-tank type is an effective, simple, natural circulation type that is attractive due to its simplicity. But the heat transfer in a tank-in-tank system is dominated by natural convection, which is generally inadequate for larger systems [3].

If a solar system with the same performance as existing forced circulation systems could be developed while addressing the drawbacks associated with existing indirect systems, it would prove beneficial to the economic competitiveness and distribution of solar thermal systems for cold climate operation. To accomplish this goal, we propose a jacketed-type storage tank that features a forced-convection heat exchanger in the form of a spiral flow passage integrated directly to the outside surface of the storage tank as shown in Fig. 1.

With a spiral-jacketed storage tank, the tank can be charged without the need for a separately installed heat exchanger and circulating pump, thus leading to a more simple system design as illustrated in Fig. 2 (b). While envisaging the aforementioned benefits of the spiral-jacketed storage tank design, the task remains to evaluate its performance relative to the existing forced circulation system. In the present paper, we experimentally compare the performance of a system with a spiral-jacketed storage tank with that of a typical forced circulation system design.




(a) Existing system with heat exchanger




(b) Remodeled system with spiral-jacketed storage tank

Fig. 2. Schematic diagrams of solar hot water systems.

## 2. Experiment and simulation

### *2.1 Spiral storage tank design*

The spiral-jacketed storage tank considered in this study has a capacity of 400 L with the exterior tank envelope surrounded by an annular jacket arranged in a spiral orientation. An anti-freeze solution circulates through the collector to this flow passage, thereby transferring heat from the outside wall of the tank to the water stored inside the tank. The width of the spiral flow passage is 15 mm and the pitch of the spiral jacket is 120 mm. The jacket spans from the tank base to a point 300 mm below the top of the storage tank. The space above the point where the spiral heat exchanger terminates is used as an expansion tank, not a flow passage.

The spiral heat exchanger's flow passage dimensions are sized to yield the same total heat transfer as the existing base case system [4]. To establish operating conditions for the passage design, one day was selected with good solar radiation. Solar radiation data as well as the average temperature of the storage tank for the existing system were established at 12:00 on Oct. 15, 2003 as standard status. The design conditions are given as follows:

Incident solar radiation on collectors: 9.211 kW
Acquired heat: 4.906 kW
Average water temperature in storage tank: 37.9°C
Outdoor air dry bulb temperature: 14.4°C

Inlet temperature of collector: 45.6°C
*UA* value of the existing system: 370 W/K

The following are assumptions made in order to simplify the spiral passage calculations:

Hydrodynamically, the spiral portion of the jacket heat exchanger is simplified as a straight flow passage (total length: 17.4 m) with a flat square cross-section ($15 \times 120$ mm)

The tank surface, except for heat exchange surfaces contacting the inside tank, is considered to be completely insulated.

The Nusselt number for the spiral jacket is calculated to be 30.8 using the Gnielinski Eq. [5] yielding a corresponding heat transfer coefficient of 531 W/m$^2$K.

The heat transfer at the inside wall of storage tank is governed by natural convection.

The temperature difference between the top and the bottom part of the storage tank is the same as that between the inlet and exit of the collector, the Nusselt number is 840 and the heat transfer coefficient 475 W/m$^2$K on the inside surface of the storage tank [6].

With a steel plate 3.2 mm in thickness used as the wall of the storage tank, the overall heat transfer coefficient *U* between the spiral-jacket and the inside of storage tank is 270 W/m$^2$K and *UA* become 492 W/K.

A flow chart and detailed procedure of the calculation are summarized in the Appendix.

The energy gain of the existing forced circulation system (4.906 kW) was established in the new system as a design goal. The product of heat transfer coefficient and area (*UA*) of the spiral-jacket part is determined to obtain the same energy gain with both the spiral jacket channel width and pitch variables to satisfy the target *UA*. A narrower flow passage width promotes heat transfer, but a 15 mm passage is considered the lower limit, to balance flow resistance and manufacturing constraints.

As a characteristic of a jacket-type storage tank, the working fluid from the collector does not directly enter into a storage tank, thereby avoiding a direct disturbance on the tank's thermal stratification. Since the temperature difference between the inlet and exit of the collectors in the present system is in the range of 2 to 3°C, the storage tank can be expected to exhibit a similar temperature range. Accordingly, although the spiral-jacketed storage tank has a good structure to maintain higher stratification, it was operated under the condition near fully mixed state to represent a worst-case storage tank scenario.

### *2.2 Composition and measurement of the system*

An active solar water heating system is retrofitted with an external heat exchanger whose characteristics of performance were sufficiently known through verification tests and simulations during a two-year period. The solar thermal systems before and after modification to include the spiral-jacketed storage tank are shown in Fig. 2.

Dimensions of the remodeled system are depicted in Table 1. There is only one piping loop circulating a 40 wt% water solution of propylene glycol between the collector and storage tank. The glycol solution gains heat as it flows through the collector before entering the spiral-jacket passage to transfer the collected heat to water stored in the tank. A detailed diagram of the remodeled system located in Suwon city (37.3°N, 127°E) is shown in Fig. 3. Included in the figure are the measurement points integrated into the solar system for use during the experiments.

The solar radiation projected on the collector surface is measured by a pyranometer inclined to match

Table 1. Specification of system.

| | | |
|---|---|---|
| Collector | Size<br>Area<br>Unit<br>Slope | 1,179×2,228 mm<br>2.5 m$^2$<br>4<br>40° |
| Storage tank | Type<br>Capacity<br>*UA*<br>Material | Spiral-jacket<br>400 L<br>492 W/K<br>Steel |
| Pump | Capacity<br>Output | 60 lpm(H: 4 m)<br>80 W |
| Control module | Type | Digital difference temperature control |
| Pipe | Material<br>Diameter | Copper & steel<br>20 mm |
| Auxiliary heater | Type<br>Capacity | Electric geyser<br>13.2 kW |

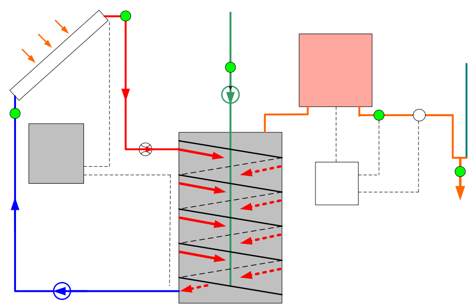
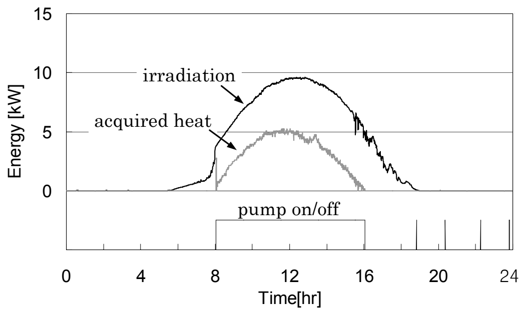


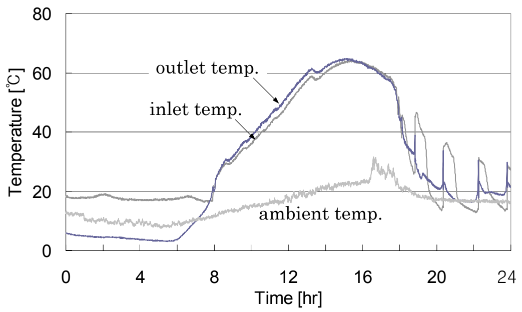
Fig. 3. Detailed diagram of remodeled system with spiral-jacked storage tank.

the collector's slope (40°). Temperatures throughout the system are measured by K-type thermocouples and recorded with a data logger. Other data associated with system operation such as load-side flow rate, electric power, and on/off status of the circulation pump are also recorded. Data collection commenced on March 2004 with data being recorded in 36-second intervals.

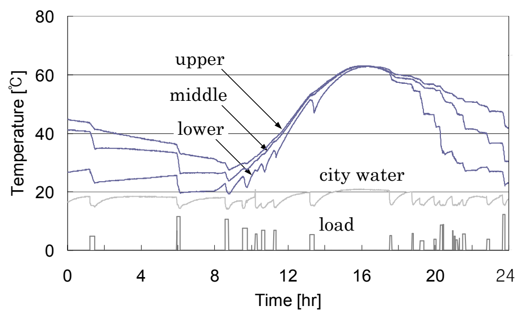
Operation of the circulation pump is controlled by monitoring the temperature difference between the pipe near the collector exit and the jacket exit of the storage tank. Whenever the temperature difference at no flow conditions exceeds 10°C, the pump is energized. When the pump is operating and the measured temperature difference falls below 1°C, the controller turns the pump off. The pump control strategy is designed to maintain a stable operating condition.




(a) Energy and pump on/off status




(b) Inlet and outlet temperature of collector

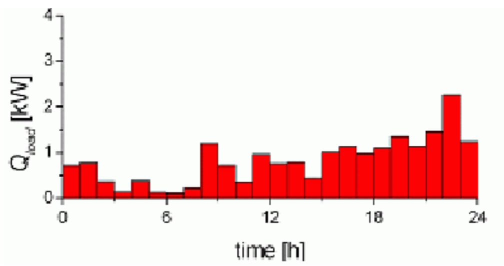



(c) Temperatures in the storage tank and load state

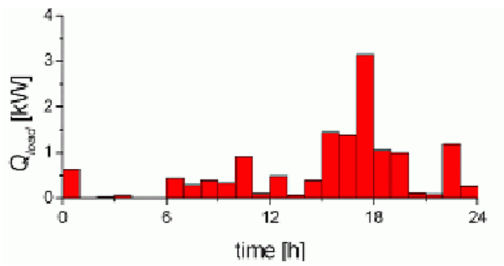
Fig. 4. Result graph on April 29, 2004.

### 2.3 Simulation

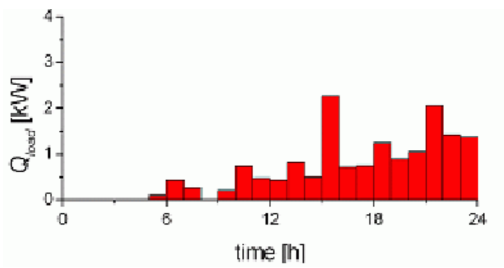
Because the forced circulation system was retrofitted to a spiral jacket configuration, it was not possible to directly compare the performance between the forced circulation and the spiral-jacketed systems experimentally under identical conditions. The comparative performance is established by utilizing long-term experiments for the tank operating under the forced circulation system arrangement. Once the forced system's detailed operational characteristics are known, it can be modeled with a load pattern as shown in Fig. 5 and simulated by TRNSYS [7]. Comparisons of the simulated and measured useful energy gain from the collector, and solar and auxiliary contribution for the existing system are summarized




(a) Weekday




(b) Saturday




(c) Sunday

Fig. 5. Simplified load patterns.

Table 2. Performance for the existing system by experiment and simulation.

(a) Daily result on October 15, 2003

| Performance | measured | simulated |
|---|---|---|
| Irradiation, [MJ] | 198.3 | 207.0 |
| Heat collected, [MJ] | 90.3 | 92.7 |
| Heat load, [MJ] | 137.2 | 137.2 |
| Efficiency, [%] | 45.5 | 44.8 |
| Solar fraction, [-] | 0.72 | 0.73 |

(b) Monthly results for December, 2003

| Performance | measured | simulated |
|---|---|---|
| Irradiation, [MJ] | 3,226 | 3,289 |
| Heat collected, [MJ] | 1,075 | 1,119 |
| Heat load, [MJ] | 2,007 | 2,002 |
| Efficiency, [%] | 33.3 | 34.0 |
| Solar fraction, [-] | 0.48 | 0.51 |

in Table 2. Agreement between simulated and measured data was good with comparative performance being within 5% [8]. This lends confidence in the validity of the simulation model for the system.

The performance of the new system measured experimentally was compared to the simulated performance of the existing system, which is discussed in next section. The TRNSYS simulation for the forced circulation system was performed by using the same climate and operation conditions as the experimental conditions for the new system.

## 3. Results and analysis

Fig. 4 shows the acquired energy rate, key temperatures, and load status for the spiral tank system over a typical spring day in Korea. Heat collection operation began at 8:00 am and terminated at 4:00 pm.

The temperature difference between the inlet and outlet of collector was approximately 2°C to 3°C, which is simultaneously distributed along the height of the storage tank as expected. This confirms the assumption introduced in the design process that the temperature distribution in the tank, which is used in calculating the log mean temperature difference for heat transfer, is equal to the temperature difference at the collector.

Temperature variations occurred frequently in the lower side of the tank because the city water comes into the bottom of the tank whenever the load is met. After the circulating pump is turned off, there are many intermittent loads, but the stratification is relatively well maintained as evidenced by the data shown in Fig. 4(c).

Table 3. Daily performance comparison between the previous and the new systems on April 29, 2004.

| Unit : MJ | Prediction by simulation for existing System | Experimental results for new System | Difference (%) |
|---|---|---|---|
| Irradiation | 252.8 | 251.5 | (0.5) |
| Acquired heat | 102.4 | 95.9 | (6.3) |
| Heat load | 99.9 | 99.9 | 0.0 |
| Heat supplied by solar system | 85.8 | 85.1 | (0.8) |
| Efficiency (%) | 40.5 | 38.1 | (5.9) |
| Solar fraction | 0.86 | 0.85 | (1.2) |

To compare the two systems, daily-integrated results are summarized in Table 3. The results of the existing indirect system are estimated by simulation using the previously validated TRNSYS model and the comparative results of the new system equipped with the spiral-jacketed storage system are reported as measured during the experiment for the times shown.

A typical measure of performance for a solar thermal system is the collector efficiency, which represents the ratio of energy acquired by the system to integrated solar radiation incident on the collector surface. The collector efficiency and the acquired energy of the new system were approximately 6% less than those of the existing system. This means that the fluid temperature entering the collector was higher than for the existing system, resulting in a lower solar gain and suggesting that the effectiveness of the spiral-jacketed heat exchanger is somewhat lower than a traditional external heat exchanger.

Another important index for system evaluation is the solar fraction, which represents the ratio of solar energy delivered to the total load. If there is no difference between two systems, the auxiliary energy quantity, which accounts for a majority of operating costs, is supplied at the same level. In economic aspects, it is advantageous that the operating cost is quite similar to the existing system, while the initial cost is reduced due to the simplicity of the system.

It was verified that the new system with a spiral-jacketed storage tank designed under the condition of

Table 4. Monthly performance comparison between the previous and the new systems on October, 2005.

| Unit: MJ | Prediction by simulation for Existing System | Experimental Results for New system | Difference (%) |
|---|---|---|---|
| Irradiation | 4,213 | 4,183 | (0.7) |
| Acquired heat | 1,684 | 1,499 | (11.0) |
| Heat load | 2,004 | 2,004 | 0.0 |
| Heat supplied by solar system | 1,476 | 1,283 | (13.0) |
| Efficiency(%) | 40.0 | 35.8 | (10.5) |
| Solar fraction | 0.74 | 0.64 | (13.5) |

good solar radiation has comparable performance to the existing system during a day with a clear sky in autumn. Unfortunately, the performance such as collection efficiency and solar fraction is inferior to that of the existing system. Table 4 shows the operating results during the month of October, 2005. Predictions for the existing system were obtained by simulation with the same weather data and load patterns in the experiment for the new system.

For a clear day similar to the design conditions, the performance difference between two systems is negligible. On the other hand, the performance of the new system with the spiral-jacketed storage tank during a monthly operation including cloudy days becomes worse than that of the existing system. There are several potential explanations for the lower performance. First, the simulation model for the existing system slightly overpredicted system performance on intermittent sunny days as documented in Table 2(b). This overprediction behavior may be responsible, in part, for the higher predicted performance compared to the modified system with a spiral-jacketed storage tank. Another potential reason for poorer performance may involve greater collector losses during cooler ambient operation. Because the spiral-jacketed heat exchanger had lower performance compared to the external heat exchanger, the return fluid temperature to the collector is higher which raises the average collector operating temperature. Higher collector operating temperatures increase thermal losses.

It is clear that the new system with a spiral-jacketed storage tank has a lower performance that the existing external heat exchanger, on the whole. Consequently, it is not possible to design the storage tank under clear day conditions and guarantee the same performance to the existing system. As a result, we recommend that systems with spiral-jacketed storage tanks be designed for about 10% higher performance in a clear day is necessary to deliver comparable thermal performance to external heat exchanger systems.

Also, design should be accomplished to maximize the ability of the stratification of the spiral-jacketed storage tank. In the present work, the flow rate was set up relatively highly for direct comparison of two systems, which is very disadvantageous to the new system. So the pitch of the spiral passage can be adjusted in order to maintain a turbulent flow for high heat transfer coefficient in spite of low flow rate for greater temperature difference between collector inlet and exit, which in turn leads to strong stratification in a storage tank.

If the passages of the spiral jacket were optimized by adjusting the pitch and the flow rate, the heat transfer coefficient would be increased with a greater temperature distribution in a storage tank and the total performance of the system might be improved. More precise calculation and analysis will be carried out by a CFD technique.

In conclusion, it is not verified that the performance is at the same level, but the possibility is confirmed with a simple system.

## 4. Conclusion

The performance of an alternative indirect solar thermal storage system design by using a spiral-jacketed heat exchanger is presented. The design integrates the heat exchanger of an indirect system into a geometric spiral heat exchanger that jackets the storage tank. The intent of the new tank design is to reduce system complexity and cost while maintaining an acceptable level of overall thermal performance.

The performance of an existing external heat exchanger system is compared with a retrofitted spiral-jacketed storage tank. In comparing the operating of a traditional external heat exchanger system with the modified system using a spiral-jacketed storage tank, the spiral-jacketed storage tank system resulted in diminished thermal performance that ranged between 6-11%. The lower performance is likely due to a less effective heat exchange in the spiral jacket compared with an external heat exchanger. At the heat exchange effectiveness decreases, the supply fluid temperature to the collector increases, thereby increasing thermal losses. Techniques for improving the spiral-jacketed heat exchanger performance are presently being sought.

## Acknowledgment

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## Appendix: Design method for spiral-jacketed storage tank

Heat loss of the pipe connecting the collectors and the storage tank is neglected. The average temperature of storage tank is given, but the temperature difference between top and bottom part is necessary for calculating the log mean temperature difference of heat exchanger. So, it is assumed that the temperature difference is the same as that between inlet and outlet of the collector, which is easily obtained from collector characteristics.

Assumption:

$$T_{s,t} = T_{s,m} + (T_{c,o} - T_{c,i})/2$$
$$T_{s,b} = T_{s,m} - (T_{c,o} - T_{c,i})/2 \tag{A1}$$

Acquired heat at the collector:

$$\dot{Q}_u = \dot{m}C_{p,w}(T_{c,o} - T_{c,i}) \tag{A2}$$

Transferred heat to the storage tank:

$$\dot{Q}_{HEX} = UA\Delta T_{lm} \tag{A3}$$

Collector efficiency

$$\eta = \frac{\dot{Q}_u}{I_t A_c} = F_R(\tau\alpha) - F_R U_L \frac{(T_{c,i} - T_a)}{I_t} \tag{A4}$$

Log mean temperature difference at the storage tank

$$\Delta T_{lm} = \frac{(T_{c,o} - T_{s,b}) - (T_{c,i} - T_{s,t})}{\ln\dfrac{(T_{c,o} - T_{s,b})}{(T_{c,i} - T_{s,t})}} \tag{A5}$$

The characteristic values of collector used in experiment:

$F_R(\tau\alpha)$=0.7321 and $F_R\ U_L$ =6.1021 W/m$^2$K

By obtaining the hydraulic diameter of the flow passage with a rectangular cross-section, the Reynolds number can be calculated and is greater than 2300, so Gnielinski's equation for turbulent flow is used.

$$\mathrm{Nu} = \frac{(f/8)(\mathrm{Re} - 1000)\,\mathrm{Pr}}{1 + 12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \tag{A6}$$

Churchill and Chu's equation of natural convection for the vertical inside wall of storage tank:

$$\mathrm{Nu} = 0.68 + \frac{0.67\mathrm{Ra}^{1/4}}{[1 + (0.492/\mathrm{Pr}^{9/16}]^{4/9}} \tag{A7}$$

From the above equations, each heat transfer coefficient can be obtained inside and outside the storage tank wall, and the overall heat transfer coefficient is determined by

$$\frac{1}{U} = \frac{1}{h_i} + \frac{\delta}{k} + \frac{1}{h_o} \tag{A8}$$

A flow chart for calculation procedure is shown in Fig, A1, but actual calculation was done by using EES.

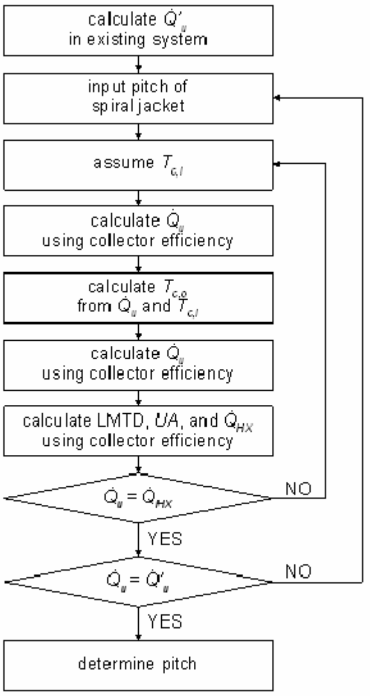


Fig. A1. Flow chart for design procedure.

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