Simulation on energy performance of air-conditioning system assisted with thermosyphon used in telecommunication base station

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ABSTRACT

Spaces with high heat generation such as telecommunication base stations (TBS) increase sharply recently. Huge energy is consumed for air-conditioning because of large indoor heat rejection and year round cooling in these spaces. Thermosyphon heat exchanger is an appropriate cooling technique for TBS, which can make full use of natural outdoor cooling resources without much electrisity consumption. To analyze the energy performance of air-conditioning system assisted with thermosyphon (ACT) used in TBS, a detailed model has been developed, which consists of cooling load model of TBS, ACT performance model and operation model. It is noteworthy that in the ACT performance model, a distributed parameter model for thermosyphon heat exchanger has been established and verified by experiments. Then, the energy-saving effects of the ACTs located in different climates are investigated using the model. The results show that the energy saving effect is significant, and the saving rate in cold regions is larger than that in warm regions.

1. INTRODUCTION

The information technology has developed rapidly in the last decade, especially for the telecommunication and data industries (Cho and Kim, 2011; Shehabi et al., 2011). As a result, spaces with high heat generation such as telecommunication (TBS), data center, increase greatly (Cho and Kim, 2011). IT server, air conditioning system, and power system mainly consume the energy of these spaces. Therein, up to 30–50% of the total energy is consumed by air-conditioning system because of the large indoor heat rejection and all-year-round cooling (Banerjee et al., 2011; Kant, 2009),which indicates the proper scheme of air-conditioning system means a great energy saving potential.

In response to this growing energy problem, many energy-saving techniques have been adopted to improve energy efficiency by making use of natural cooling resource, such as waterside economizers (Yury, 2010), airside economizers (Chen et al., 2009; Scofield and Weaver, 2008; Keith, 2011). For waterside economizer, it is dangerous for IT devices to bring water into the telecommunication base station room. Airside economizers have mainly been considered in TBS cooling, which include direct fresh air-cooling system and separated air heat exchanger system. Direct fresh air-cooling system is considered as the most efficient method to make use of cooling resource. Unfortunately, it will bring many problems such as accumulation of dust and too low or too high relative humidity. Separated air heat exchanger mainly includes aluminium plate heat exchanger and thermosyphon heat exchanger. The actual temperature efficiency of plate heat exchangers, which is limited by the structure and the heat transfer mode, is not high (Zhou et al., 2011). Thermosyphon heat exchanger can transfer much heat in quite high efficiency without the problem brought by outdoor air. Therefore, thermosyphon heat exchanger is considered as an appropriate energy-saving technology for TBS. Consequently, the air-conditioning system assisted with thermosyphon (ACT) was proposed for TBSs and IDCs by some researchers, such as Zhou et al. (2011). The system contains two independent devices: the thermosyphon heat exchanger and the conventional air-conditioner, in which the former works in cool season for energy saving, the latter works in hot season for safety. Some feasibility analysis and experimental investigations about the ACT system have been carried out. However, the simulation study about the ACT annual energy performance in different climates is still needed.

To analyze the energy performance of ACT used in TBS, a detailed model has been developed, which consists of cooling load model of TBS, ACT performance model and operation model. Therein, the cooling load model is based on annual hourly simulation software (DeST); the operation is optimized for maximum energy-saving effect. The ACT performance model consists of thermosyphon heat exchanger model and air-conditioner model. Compared to the common air conditioner model, much attention has been paid on the modelling of the thermosyphon heat exchanger. Conventionally, in the existing energy performance simulation, the thermosyphon model simply considers the heat transfer (or *COP*) proportional to the temperature difference between indoor and outdoor (Tu et al. 2011), which will reduce the accuracy of the simulation. In the paper, a steady state distributed parameter thermodynamic model for thermosyphon heat exchanger is established by using the conservation equations of mass, momentum and energy and the model will be verified by experiments, which is supposed to increase the simulation accuracy of thermosyphon a lot and realize a more reliable simulation on energy performance of the thermosyphon.

Then, the energy saving effect of the ACT located in different climate (four china typical cities: Harbin, Beijing, Shanghai, Guangzhou) is investigated. The results show that the energy saving effect is significant, and the saving rate in cold regions is larger than that in warm regions.

2. MODELLING

An air-conditioning system assisted with thermosyphon consists two independent device: air-conditioner and thermosyphon. To analyze the energy-saving effect and applicability of ACT used in TBS , an energy performance model has been developed, which consists of cooling load model of TBS, AC performance model and operation model. The cooling load model provides the TBS annual hour-by-hour cooling load by software. The cooling capacity and input power of AC and ACT are provided by the AC performance model and thermosyphon model. Finally, in the operation model, the relationship between the load of TBS and the performance of AC and ACT is built, based on which the energy consumption of AC and ACT can be calculated.

2.1 Cooling load model

In the load model, DeST is adopted to simulate the annual cooling load hour-by-hour. DeST is a building thermal environment simulation software developed by Tsinghua University, which is widely used for building load prediction.

Tu et al. (2011) have collected the information on current TBS in china. A typical TBS is chosen for simulation. The TBS is an unattached building with no windows, with the size of 5 m (L)×5 m (W)×3.3 m (H). 24 brick wall (240mm claymortar+20mm lime mortar) is adopted as the envelop according to the recommendation by Tu et al. (2011). The air change rate is set at 0.5 taking the infiltration through the door and other gaps into account. The indoor IT heat dissipation rates is set as 5 kW, and the indoor temperature is controlled at 28±2°C.

2.2 ACT performance model

ACT performance model consists of thermosyphon heat exchanger model and air-conditioner model. They have been developed respectively to calculate each cooling capacity and energy consumption at various conditions.

2.2.1 Thermosyphon heat exchanger model

Thermosyphon heat exchanger is a passive circulation (Fig. 1 and Fig. 2), in which the working fluid absorbs heat and evaporates in the evaporator, vapor (or two-phase) rises in the riser (gas tube), releases heat and condenses in the condenser, then the liquid returns back to the evaporator through the downcomer (liquid tube) by the force of gravity. The internal state parameter is difficult to calculate because of the heavy coupling of momentum, energy and mass due to no mechanical cycling component, such as pump or compressor. So when the external thermal condition maintains constant, to calculate a certain loop steady state condition, three parameters of starting point, include pressure P_0, enthalpy h_0 and circulation mass flux *G*, must be assumed before the calculation.

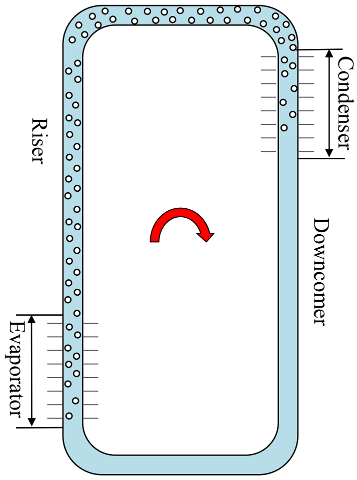


Fig. 1. Schematic diagram of the two-phase thermosyphon loop

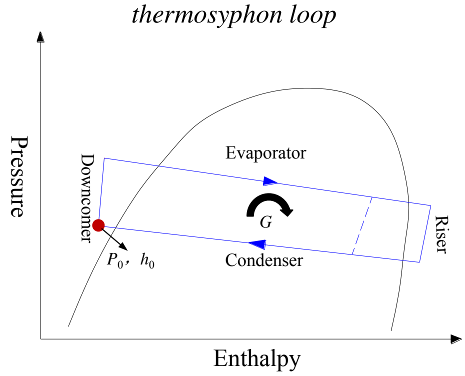


Fig. 2. State parameter of thermosyphon loop in P-h diagram

Thermosyphon heat exchanger is natural circulation, which means the pressure drop balances with the liquid head caused by the density difference of the liquid in the downcomer and the vapor/liquid mixture in the riser. Additionally, heat absorbed in the evaporator is equal to the heat released in the condenser when the heat loss through the connecting pipe is ignored. The sum of working fluid mass in the loop must be equal to the initial working fluid charge. Consequently, the three assumed parameters can be figured out by using the conservation equations of mass, momentum, and energy. Then, the heat exchanger state parameters and heat transfer performance can be determined.

Base on the above analysis, a steady distributed-parameter model of thermosyphon loop can be established. And the sequential module method is used in the solving the modelling (Fig. 3): Firstly, input the thermosyphon loop geometry and divide into n nodes in each section; Secondly, assume the initial parameters (at the outlet of condenser) and circulation mass flux; Thirdly, calculate each node along the loop according to the heat transfer and momentum transfer correlations (as shown below); Finally, judge whether each the conservation equation is satisfied: if all yes, the calculation is finished and output the results; otherwise, modify corresponding unknown parameter and continue.

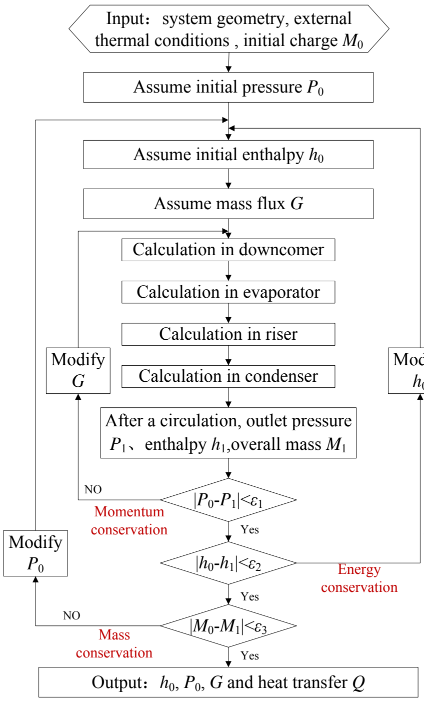


Fig. 3. Simulated flow diagram of two-phase thermosyphon loop

During the circulation, two-phase pressure gradient consists of three parts-gravitational, frictional, and accelerational, it may be expressed as:

$$-dP=-dP_g-dP_f-dP_a \tag{1}$$

The gravitational pressure gradient results from gravity acting on the fluid and is given by

$$dP_g=\rho_m gh \tag{2}$$

where ρ_m is the two-phase mixture density, which depends on the vapor void fraction,

$$\rho_m=\alpha\rho_g+(1-\alpha)\rho_l \tag{3}$$

Here, Tom model (Imura et al, 1988) is used to calculate the vapor void fraction, showed as

$$\alpha=(1+(\frac{1-x}{x})(\frac{\rho_g}{\rho_l})^{0.89}(\frac{\mu_l}{\mu_g})^{0.18})^{-1} \tag{4}$$

The frictional pressure gradient is modelled using the correlation from Lockhart and Martinelli (Lockhart and Martinelli, 1949).

The accelerational pressure drop results from the change in velocity due to the vapor quality and void fraction change, expressed as

$$dP_a=G^2d\left(\frac{(1-x)^2}{\rho_l(1-\alpha)}+\frac{x^2}{\rho_g\alpha}\right) \tag{5}$$

For the energy modelling, in the evaporator, the boiling heat transfer coefficient is calculated by a semi-empirical correlation developed by Rahmatollah (Rahmatollah, 2005). In the condenser, the Nusselt correlation is used to calculate the condensation heat transfer coefficient. For single-phase flow heat transfer, the conventional Dittus-Boelter correlation is adopted.

The thermosyphon heat exchanger model has been validated by experiments. The experiments were carried out in a standard air enthalpy difference lab. The thermosyphon heat exchanger is air-to-air type and the specifications are listed in Table 1. The airflow face velocity is 2 m/s. The working fluid is R22. The temperature difference between the evaporator inlet air and condenser inlet air varies from 4ºC to 26 ºC. As shown in Fig. 4, the simulation results are in good agreement with the experiment results, which verifies the accuracy of the thermosyphon model. Additionally, the cooling capacity increases approximately linearly with the temperature difference increase. The cooling capacity increases and then decrease with the increase of refrigerant charge. The reason for the comparatively worse agreement at lower charge may be the “dryout” phenomenon is underestimated in the simulation. The maximum cooling capacity is achieved when the refrigerant charge is 6 kg, which is considered as the optimal charge and will be adopted in simulation hereinafter.

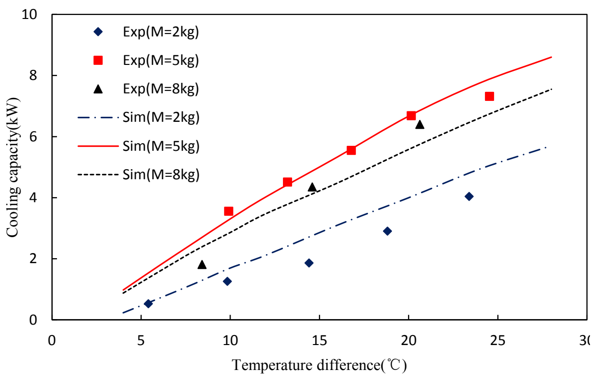


Fig. 4. Comparison between simulation and experiment results

Table 1 Specifications of thermosyphon heat exchanger

Items	**Condenser**	**Evaporator**
Tube bundle specification	3 rows, 28 tubes per row	3 rows, 28 tubes per row
	Tube length=700 mm	Tube length=700 mm
	Tube pitch=25.4 mm, row pitch=22.2 mm	
Tube size	Outer diameter=9.52 mm; Inner diameter=0.7 mm	
Fin specification	Wavy fin; Pitch=1.814 mm; Thickness=0.105 mm	
Height difference	2 m	

2.2.2 Air-conditioner model

Generally, only rated parameter is provided for split air-conditioner in the market. Therefore, some annual air-conditioner performance simulations have been carried out base on the constant rated *COP*. However, the air conditioner always operates under variable circumstance in practical application, which makes the real performance change accordingly. In this study, a relatively detailed AC performance model has been developed, which is capable to calculate the real cooling capacity and power consumption under variable conditions.

In the air conditioner performance model, a national standard is employed to calculate the real cooling capacity and power consumption, with assumptions showed below:

1) The cooling capacity and input power both depend on To and Ti;
2) When Ti keeps constant, with the drop of To, the cooling capacity increases and input power decreases linearly until To reaches 19°C, which is due to the decreasing condensing temperature. When To drops lower, the cooling capacity and input power keep fixed, because the condensing temperature has to stay to maintain the pressure difference between the expansion valve.
3) Under certain To, as Ti increases the input power keeps constant and cooling capacity rises by 3% due to the increase of evaporating temperature.
4) Latent cooling capacity is ignored, since sensible heat load accounts for most of the load in TBS.

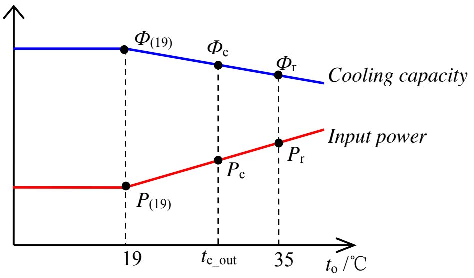


Fig. 5. Variation of cooling capacity and input power of AC with To

The functions of cooling capacity and input power with To under certain Ti are illustrated in Fig. 5. The rated cooling capacity (Φr) and rated input power (Pr) when To is equal to 35 °C are given by manufacture. Another cooling capacity (Φc) and input power (Pc) are needed to solve the function. The default Φc and Pc (Tc_out=29°C, Φc =1.077Φr, Pc=0.914Pr) are provided after investigation. As a result, the cooling capacity and input power of VC model can be figured out according to Fig. 5.

In the simulation, for conventional air-conditioner, the AC performance model is used to calculate the cooling energy consumption; for air-conditioning system assisted with thermosyphon, the part of air-conditioner is calculated by the AC performance model, the thermosyphon heat exchanger model above is used to calculate the part of thermosyphon. The rated parameters of the air-conditioner and thermosyphon heat exchanger are listed in Table 2.

2.3 The operation model

The operation model connects the air-conditioning system performance model and the TBS cooling load model, which tries to remove the TBS cooling load with minimum energy consumption. Since the cooling load is obtained hour-by-hour, the energy consumption will also be calculated hour-by-hour. In part load of both air-conditioner, start and stop control is adopted, and operation ratio R is used to calculate the energy consumption.

For conventional air-conditioner, in each hour, first calculate the cooling capacity Q_{AC} and energy consumption P_{AC} of the air-conditioner according to the indoor and outdoor temperature; then, figure out the operation ratio $R_{AC}=L_{TBS}/Q_{AC}$; So the energy consumption in this hour will be $E=P_{AC} \cdot R_{AC}$.

For air-conditioning system assisted with thermosyphon, *COP* is adopted as the criterion to determine which unit runs first. If the *COP* of thermosyphon heat exchanger is higher than air-conditioner, thermosyphon operation is prior to air-conditioner. In this case, if the cooling capacity of thermosyphon Q_{TS} is larger than the cooling load L_{TBS}, the energy consumption in this hour will be $E=P_{TS} \cdot R_{TS}$ ($R_{TS}=L_{TBS}/Q_{TS}$); otherwise, the air-conditioner has to run in parallel to remove the rest load (L_{TBS}-Q_{TS}), so the energy consumption in this hour will be $E=P_{TS} \cdot R_{TS} + P_{AC} \cdot R_{AC}$ (R_{TS}=1,R_{AC}=

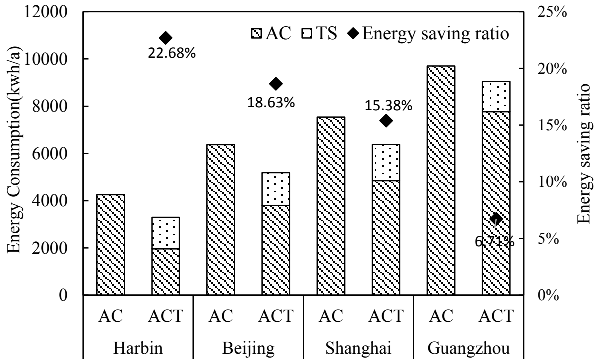
(L_{TBS}-Q_{TS}) /Q_{AC}). If the *COP* of thermosyphon heat exchanger is lower than air-conditioner, air-conditioner runs first, which is similar with the conventional air-conditioner above.

Table 2 Rated parameters air-conditioner and thermosyphon heat exchanger

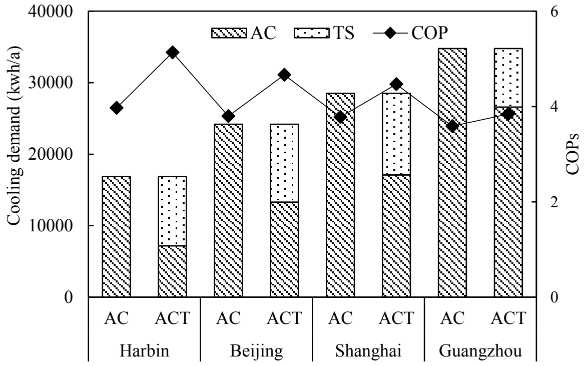
Mode	Rated parameters				
	Outdoor DB (°C)	Indoor DB (°C)	Cooling capacity (kW)	Input power (kW)	COP
Thermosyphon	18	28	6.65	0.65 (fans)	10.2
Air-conditioner	35	27	7.30	2.74 (fans and compressor)	2.70

3. SIMULATION RESULTS

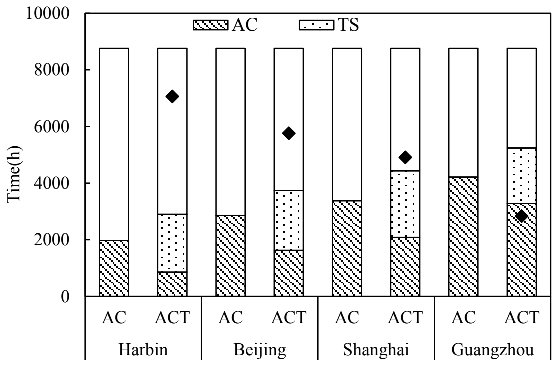
Four typical cities in the four climate zones of China are chosen to analyse the energy performance of ACT used in TBS: Harbin in severe cold zone, Beijing in cold zone, Shanghai in hot summer & cold winter zone, Guangzhou in hot summer and warm winter zone.



(a) Energy consumption and energy saving ratio



(b) Cooling demand and total *COP*



(c) Different operation time

Fig. 6. Annual operation comparison between ACT and AC in 4 TBSs

The annual operation comparison between conventional air-conditioner (AC) and air-conditioning system assisted with thermosyphon (ACT) in four TBSs is shown in Fig. 6. Therein, AC refers to conventional air conditioner and TS refers to thermosyphon heat exchanger.

Compared with conventional air-conditioner, the air conditioning system assisted with thermosyphon has significant energy-saving effect (shown in Fig. 6(a)). The energy consumption drops sharply with the assistance of thermosyphon, which can be explained in Fig. 6(b) and Fig.6(c). In Fig.6(c), the air-conditioner operation time decreases, replaced by thermosyphon. According to the operation model: who has higher *COP* who runs first, the air-conditioner operation with lower *COP* is replaced by thermosyphon operation with higher *COP*, so ACT consumes less energy than AC. While, total air-conditioning operation time becomes longer (shown in Fig. 6(c)) with assistance of thermosyphon, due to thermosyphon lower cooling capacity than air-conditioner. As shown in Fig. 6(b), for the same cooling load, thermosyphon with higher *COP* provides part of cooling capacity, results in a higher total *COP* than AC.

From Harbin to Guangzhou, with the outdoor temperature increase, the annual total cooling load (or cooling capacity needed) increases largely (shown in Fig. 6(b)), the air conditioning system operation time increase accordingly (shown in Fig.6(c)), so the annual energy consumption has a significant increase (shown in Fig. 6(a)).The energy consumption of both AC and ACT in Guangzhou is more than twice those of Harbin respectively.

Additionally, from Harbin to Guangzhou, with the outdoor temperature increase, the energy-saving ratio decrease gradually, ranging from 22.7% in Harbin to 6.7% in Guangzhou (shown in Fig. 6(a)).This is mainly because the thermosyphon operation ratio (thermosyphon operation time/total air conditioning system operation time) decreases. As shown in Fig. 6(c), the thermosyphon prior operation time (TS with higher *COP*) reduces with the outdoor temperature increase. Even if the *COP* of thermosyphon is higher than air-conditioner, thermosyphon can not provide enough cooling to remove the cooling load on its own at relatively high temperature, at that time air-conditioner has to start for supplement. As shown in Fig. 6(b), the total *COP* of both AC and ACT decreases slightly as the outdoor temperature increases, due to higher condensing temperature for AC and lower temperature difference for thermosyphon. In a conclusion, the energy saving effect of ACT is more significant in cold regions than in hot regions. The thermosyphon heat exchanger is appropriate for TBS cooling in cold regions.

CONCLUSIONS

Thermosyphon heat exchanger is an appropriate cooling technique for TBS. To analyse the energy performance of ACT used in TBS, an energy consumption model of ACT used in TBS has been developed, in which a distributed parameter model of thermosyphon has been established in detail and verified by experiments. The energy saving effect of the ACT located in different climate has been investigated by simulation.

Results show that the energy saving effect of ACT is significant, and the saving rate in cold regions is larger than that in warm regions, ranging from 22.7% in Harbin to 6.7% in Guangzhou.

NOMENCLATURE

d	=	diameter, m
E	=	Energy consumption, W
G	=	mass flux, kg m^{-2} s^{-1}
g	=	gravitational acceleration, m s^{-2}
M	=	refrigerant charge, kg
P	=	pressure, Pa
P_f	=	frictional pressure drop, Pa
P_a	=	accelerational pressure drop, Pa
P_g	=	gravitational pressure drop, Pa
q	=	heat flux, W m^{-2}
Q	=	cooling capacity, W
R	=	operation ratio
T	=	Temperature, ℃
X	=	Martinelli parameter
x	=	vapor quality
α	=	void fraction
ρ	=	density, kg m^{-3}
μ	=	dynamic viscosity, kg m^{-1} s^{-1}
Subscripts		
g	=	gas
l	=	liquid
m	=	two-phase
i	=	indoor
o	=	outdoor
Abbreviation		
AC	=	air conditioner
ACT	=	air conditioning system assisted with thermosyphon
COP	=	coefficient of performance
TS	=	Thermosyphon
TBS	=	Telecommunication base station

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