高甫生空调论文选集

1979－1993

高甫生　等著

学术专著

Scholarly Monograph

SELECTED WORKS OF GAO FUSHENG ON AIR CONDITIONING

哈尔滨工业大学出版社

高甫生空调论文选集

(1979–1993)

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内 容 提 要

本书收入了三十三篇论文，用中英两种文字发表，其中，在国际学报和国际会议发表过的论文采用英文书写. 论文集反映了作者多年来在暖通空调方面的研究成果。内容包括表面式空气冷却器换热规律的试验研究、空调机组热工特性的研究、空调机组的噪声与气流阻力关系、冷凝热在恒温空调器上的应用、潮湿车间的去雾除湿、低温贮粮库空调设计、空调试验中的温度取样法测量问题、高层建筑防排烟系统设计、高层建筑空气渗透理论和计算方法等。书中阐述了作者的许多新观点、新方法，以及作者发现的新规律和研制出的新设备. 本书可供从事暖通空调专业的技术人员、研究人员、大专院校教师、博士生、硕士生和本科生参考.

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序

论文集作者自 1962 年毕业于清华大学以后，长期在哈尔滨建筑工程学院致力于通风空气调节的教学与研究工作，特别在空调机组热工性能研究和高层建筑空气渗透理论与计算方面有独到见解、造诣颇高，深受已故前辈徐邦裕教授的赞赏。

论文集收集了作者近十几年所撰写的有代表性的论文，其中水平突出的有以下三方面内容:

1. 空调机组热工性能方面。在国际上首次提出空调机组热工参数变化的两条重要规律，即：进出口空气相对湿度变化的相关规律和空调机组空气湿球温降近似等于常数的规律，确立了空调机组的热工特性理论。在此基础上提出的一种确定空调机组全套热工性能曲线的简易方法，简化了空调机组产品的研制和测试工作，并为正确选择使用空调机组提供了依据。

2. 表面式空气换热器热质交换方面。作者根据大量试验结果，证实了表面式冷却器干、湿表面换热系数之间存在着一定的换算关系，从而可以将表面式空气加热器和表面式空气冷却器干工况的大量试验数据应用到湿工况，致使湿工况下表面式空气冷却器的试验得以简化。

3. 高层建筑空气渗透理论与计算方面。作者在国际上首次提出用理论分析求解具有不同内部隔墙的建筑物的热压系数和内部隔断系数，从而形成一套比较完整的高层建筑空气渗透的理论和计算方法。

祝贺作者在国内外同行们支持下取得丰硕成果，盼作者进一步发挥才智为暖通空调事业做出更大贡献。

清华大学教授

彦启森

1993 年 3 月于北京

前　　言

本论文集从作者近十几年来发表的四十篇空调及暖通方面的学术论文中选编三十三篇，其中包括在国际权威学术刊物及国际学术会议发表(或接受)的论文五篇，国际学术会议交流论文两篇；在全国性学术刊物及全国性学术会议上发表的论文二十六篇。为了不使论文集篇幅过长，另有在省级学术年会及其他学术会议上发表的七篇论文未收人论文集。

论文分国际和国内两部分，其中一部分论文是与其他作者共同完成的。论文集反映了作者近十几年来在空调领域中的一部分研究成果。有些是在国际上首次发现的新规律，或是作者提出的新观点，新理论和新的计算方法。这些内容对于学术研究和工程设计均具有较高的实用和参考价值。改革开放的浪潮为我国经济建设注人了新的生机和活力，也为空调事业的发展迎来了黄金时代。作者希望通过本论文集的出版，把自己多年来的研究成果奉献给我国的暖通空调事业，期待它能对空调技术发展有所贡献。

在此，作者诚挚地感谢我的老师、中国制冷学会空调制冷专业委员会副主任、清华大学空调教研室主任、国内外著名的空调制冷专家彦启森教授为本论文集做了评阅，并提出宝贵意见。

作者把本论文集奉献给已故的我国空调事业的前辈徐邦裕教授、缅怀他为我国暖通空调事业发展所作出的贡献。同时感谢徐教授在我的科研工作中给予的指导与帮助。

作者感谢国内外同行专家们所给予的一切支持和帮助；感谢我的合作伙伴陆亚俊教授马最良教授、王慕贤副教授、王群高工等在部分研究项目中的支持和帮助；感谢郭守信高工、吴隆滨、刘子文、谭滨立、焉晨利等工程师在论文 15、16 的测试工作中给予帮助。我的研究生丁力行、王凤波在渗风研究中，特别是在渗风计算机程序方面做了很多工作，在此一并表示感谢。

由于作者水平所限，论文中不妥之处在所难免，恳请读者批评指正。

高甫生

1993.2

目　　录

国际学报及国际会议论文

全国性学术刊物及全国性学术年会论文

Establishment of performance charts of an air conditioner with one certified test point

Gao Fusheng

Key words:air conditioner,performance charts,model

This paper demonstrates,in terms of both experiments and theory, that the wet bulb temperature depression of a given air conditioner at a given air rate is approximately constant and that the relation between the relative humidity of the inlet and outlet air can be determined by drawing a family of curves through a particular point in the saturated curve on the psychrometric chart. Thus it is possible to establish the complete performance charts for a particular air conditioner from one test point.

In recent years the production of air conditioners has increased very repidly and performance testing has become a significant task. The measuring and testing procedures are quite troublsome, especially since it takes considerable time to chang the inlet air condition parameters. Furthermore, after testing there is still much more work to be done in analysing the data. Also the usual air conditioner test cannot cover all the variables and is usually restricted to changing the wet bulb temperature of the entering air. This paper is based on results of a large number of tests conducted by the author who obtained the relations between wet bulb temperature and relative humidity changes in the air conditioner. A simplified method of testing air conditioners is proposed which allows its full performance characteristics to be predicted.

The characteristics of wet bulb temperature changes

It has been found that the wet bulb temperature of outlet air (OWB) or the wet bulb temperature depression (WBTD) will vary simultaneously with the paramenters of the inlet air,air rate and condenser pressure, in an air conditioner.

Both the test results and theoretical analysis show that when the air rate and condensing pressure in an air conditioner remain constant, the OWB t_{wb2}in the air conditioner is related only to the wet bulb temperature t_{wb1} of the inlet air (IWB). It cannot be influenced by any other parameters of the entering air such as (t_{db}, Φ). No matter what changes occur with the dry bulb temperature, a definite IWB must correspond with a definite OWB. ie when t_{wb1} increases by 1℃, t_{wb2} also increase by 1℃. In other words, for each type of air conditioner in which the air rate remains unchanged and the condenser pressure varies only within a small range, the WBTD of the air handled by the conditioner, Δt_{wb}, would be a constant value. The performance chart of wet bulb temperature changes for inlet and outlet air obtained in 1966 by experiments on a hitachi air conditioner Model RP−1004, is shown in Fig.1. Fig.2 shows the performance chart of WBTD corresponding to the changes in IWB. During the experiments, the inlet air dry bulb temperature flutuated widely, eg from 19.6℃ to 36.6℃. The charts show that the test points follow a distinct pattern.

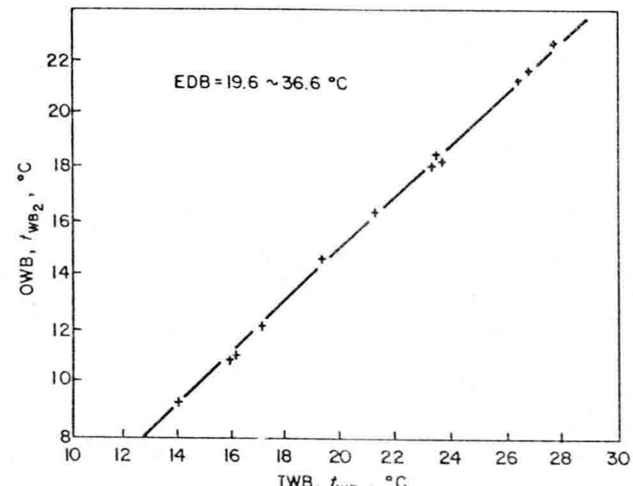

Fig.1 The relationship between the wet bulb temperature of both inlet (IWB) and outlet (OWB) air

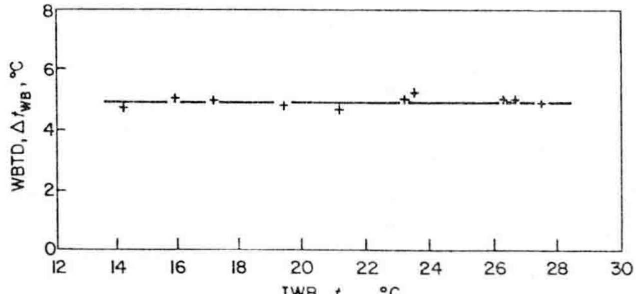

Fig.2 The test result of wet bulb temperature depression of air

The wet bulb temperature of air represents its enthalpy value. Therefore, the results mentioned above also represent the relation of enthalpy changes of inlet and outlet air in an air conditioner,ie when air flow rate and the enthalpy value of inlet air are known, the enthalpy value of the outlet air can be determined at the same time. It should be pointed out that although WBTD of the air conditioner is constant, its enthalpy depression is not constant. The ratio of enthalpy depression to wet bulb temperature depression $\alpha = \Delta h / \Delta t_{wb}$is not a constant. because the saturation line on the psychrometric chart is not a straight line. Within different temperature ranges, the ratio α is variale. For instance, when atomspheric pressure $B = 1.013 \times 10^2$Kpa and $\Delta t_{wb} = 1$℃ , α varies from 0.55 to 1.2 if t_{wb}differs within 10–30 ℃. Therefore, as IWB in the air conditioner varies, its enthalpy depression changes correspondingly and its cooling capacity would also be variable. It can be proved theoretically that WBTD of an air conditioner is constant at a certain air flow rate. According to the principles of a balance between the compressor and evaporator of a refrigerator. Yang has made theoretical assumptions and given an approximate formula for calculation of WBTD $\Delta t_{wb}$of an air conditioner[2].

$$\Delta t_{WB} = 0.03(\delta\varepsilon)^{0.465} \frac{(n_e Q_0)^{0.635}}{bG} \text{℃} \tag{1}$$

where $Q_0$is the standard cooling capacity of the refrigeration compressor in the air conditioning. W;G is the air flow rate of the air conditioner, kg s^{-1}; ε is the total efficiency of the evaporator under wet surface condition;$n_e$is the correctin factor for the condensing temperture; δ is the fouling coefficient of the evaporalor; and b is the correcting coefficient of atmospheric pressure for converting enthalpy depression and wet bulb temperature depression.

In (1) Yang[2]took into account the correcting coefficient of atomospheric pressure, b, for α, then converted enthalpy potential using wet bulb temperature depression, but neglected corrections of the atmospheric pressure of air specific volum v. These two correcting factors would be cancelled out by each other. If the two correcting factors are considered at the same time, the influence of atmospheric pressure onΔt_{wb}would be very small. Moreover, temperature also affects air specific volum V. If the two correction are introduced in (1), it will be complicated further. Obviously, the ideal approach is to calculate Δt_{wb}in the standard condition of the air conditioner. In this case it is unnecessary to introduce various correcting factors in the formula. These correcting factors are used only when cooling capacity under different conditions is calculated on the basis of Δt_{wb}. Therefore, in the opinion of the author, it is more appropriate to consider(1) as follow

$$\Delta t_{WB}=0.03(\delta\varepsilon)^{0.465}\left(\frac{n_e Q_0}{G}\right)^{0.635}\ ℃ \tag{2}$$

From (1) it can be seen that WBTD of an air conditioner Δt_{wb}varies with standard cooling capacityQ_0, air flow rate of the air conditioner, total efficiency of the evaporator (which varies with air flow rate), condensing temperature (n_e) and fouling coefficient of the evaporator(δ). So that, for a certain air conditioner ($Q_0$is known when it is new, δ can also be neglected)when air flow rate and condensing pressure remain constant, wet bulb temperature depression Δt_{wb}must be a constant value.

The WBTD of an air conditioner Δt_{wb}varies with the air flow rate. Fig 3 shows that Δt_{wb}decreased rapidly as the flow rate increases. However,since the heat exchange coefficient of the evaporator increases due to the increase in air flow rate, the cooling capacity of the air conditioner increases. Statistical data on air conditioners made both in China and elsewhere show that the relation between air flow reate and cooling capacity is that the cooling capacity increases by approximately 1% for each air flow rate increase 3–5%. This value varies slightly with the heat exchange efficiency of the evaportor. But for an ordinary air conditioner, the variation in air flow rates is usually not large in amplitude. Therefore, when the ait flow rate varies, correcting the cooling capacity with this coefficient could satisfy the requirements of general calculation.

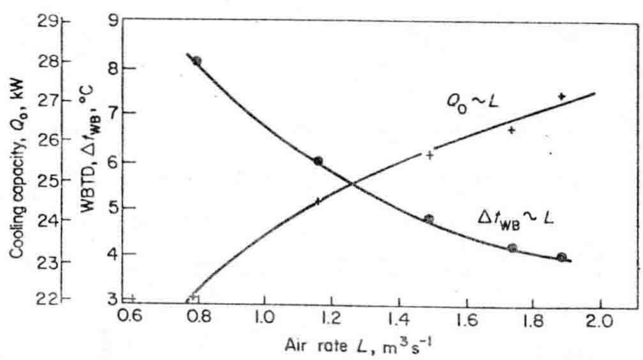

Fig.3 The relationship between wet bulbtemperature depression and cooling capacity for different air rates

Another factor which can influence Δt_{wb} is the condensing pressure. Fig.4 shows that OWB t_{wb2} and cooling capacity corresponding to the performance curve of condensing pressure changes when OWB t_{wb1} of the air conditioner is constant. The figures show that when the condensing pressure of R 12 changes from 6.5 to 10 $\times 10^2$Kpa (condensing temperature 30–40℃) t_{wb2} only increases by 0.95℃ . If the condensing pressure remaina at $8 \pm 0.5 \times 10^2$kPa (R 12). the value of t_{wb2} deviates less than ± 0.15℃, and cooling capacity deviates less than $\pm 2\%$.

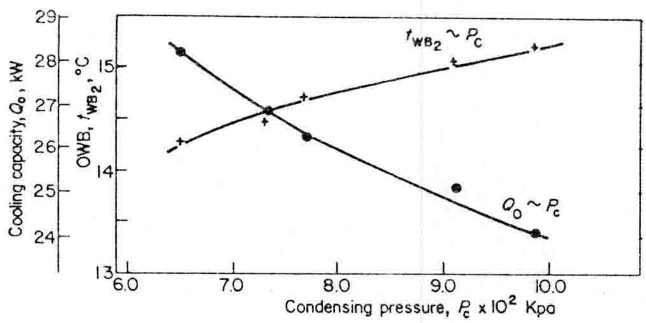

Fig.4 The relationship between outlet air wtc bulb temperature (OWB) and cooling capacity for different condensing pressures

Changes in relative hunidity of the air conditioner

Relative humidity of outlet air in an air conditioner is a function of the parameters of the inlet air, air flow rate and condensing pressure.

Tests have shown that at a certain air flow rate ,relative humidity outlet air from an air conditioner, φ_2, varies only with relative humidity of inlet air φ_1. Any other parameter changes of the inlet air condition do not affect the relative humidity of outlet air. When $\varphi_1$is a constant ,$\varphi_2$can also be determined. When $\varphi_1$is large, $\varphi_2$is large too, and vice–versa (see Fig.5). In other words, when the air flow rate is constant, if the inlet air condition varies following a certain φ–contour line on the psychrometric chart, the outlet air condition will vary following another corresponding φ–contour line.

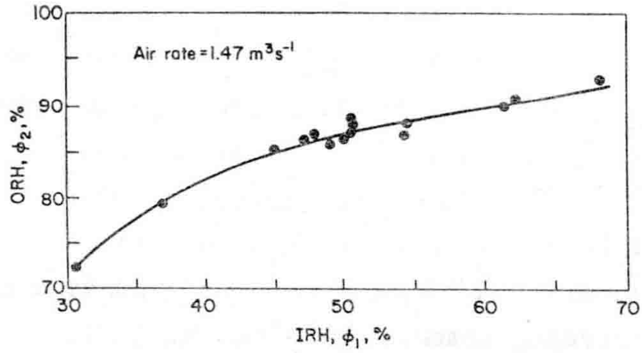

Fig.5 The relationship of relative humidity between inlet (IRH) and outlet (ORH) air

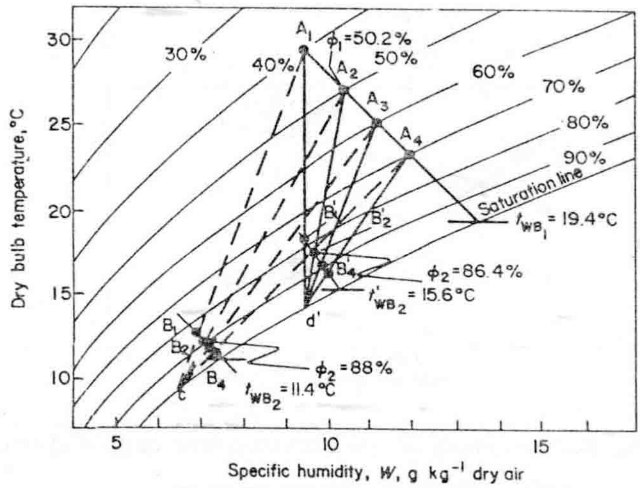

Fig.6 The performance of air conditioning process paths

The performance of φ in the air conditioner can also be explained by'focusing' the performance of the air condition process paths. So called 'focusing' of performance implies that where inlet air parameters of the air conditioner change along a wet bulb temperature contour in a pschrometric chart, outlet air parameters also change along another wet bulb temperature contour. The lines joining the points of inlet and outletair conditions intersect the saturation line at the same point C (focusing point)[3.4](see Fig.6). Point C is the average surface temperature of the evaporator. The performance of the air conditioner is determined by the fact that when IWB remains constant,the total amount of heat tranferred to the refrigerant by air will also be a constant value. Simultaneously, the evaporating temperature of the refrigerant is kept at set points, and the average surface temperature of the evaporator is also kept constant. When the dry bulb temperature of inlet air for the air conditioner changes, the total amount of heat released from the air will not change. Here only the ratio of sensible heat to latent heat of the air will change. Fig.6 shows that focusing performance has shown that there exists a certain relation between $\varphi_1$and $\varphi_2$of air ,namely a lower value of $\varphi_1$will result in a lower value of $\varphi_2$and vice–versa.

Tests show that air rate has little effect on changes of relative humidity. When the air rate increases and $\varphi_1$remains constant, $\varphi_2$decreases and vice–versa. Fig.7 shows the curve of $\varphi_2$corresponding with the changes of air rate when $\varphi_1 = 50\%$. For instance, when the air volum increases from $0.78m^3s^{-1}$to $1.85m^3s^{-1}$(by 136%) $\varphi_2$decreases by only 3.6%(88%–84.4%). Generally speaking, when an air conditioner is being used in practice, its air rate will not vary to such a large degree. When the ait flow rate is equal to 0.8–1.2 of the certified air flow rate, $\varphi_2$deviates by about ± 1%. Here the effectof air flow rate on $\varphi_2$can be neglected.

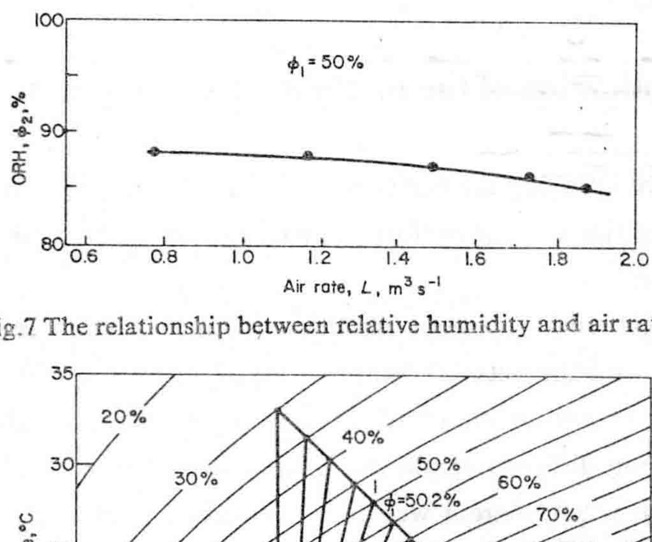

Fig.7 The relationship between relative humidity and air rates

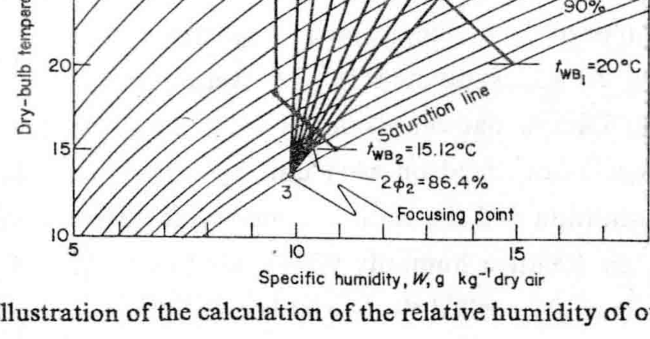

Fig.8 Illustration of the calculation of the relative humidity of outlet air

The effect of air flow rate on $\varphi_2$is clear when considering focusing performance. From Fig.6 it can be seen that when air volume increases from $0.78m^3s^{-1}$to $1.48\ m^3s^{-1}$(by 89%), t_{wb2}incresaes from 11.4℃ to 15.6℃, average surface temperature increases from poin c to point c'. At this time φ does not vary greatly between points of outlet condition $B_1$and B'_1.

Tests show that variations of condenser pressure have to little influence on φ that it is difficult to measure this with an ordinary instrument. Because the variation of condensing temperature have little influence on the evaporation temperature, the surface temperature of the outlet evaporator and wet bulb temperature of outlet air will also vary very little. For instance, when the condensing teterature increase from 30℃ to 45℃, t_{wb2}increases only by 0.95℃ and the corresponding surface temperature of the evaporator also increases very little. It can be seen from Fig.6 that when point C and t_{wb2}only vary slightly, the variations in the value of φ are difficult to observe.

Application of the performance of t_{wb} and φ

The purpose of studying the performance of an air conditioner is for its applications. The applications of the performance of $t_{wb}$and φ of an air conditioner are evaluated as follow:

To simplify test work on an air conditioner. Since cooling capacity of an air conditioner varies with the wet bulb temperature of inlet air, cooling capacity of the air conditioner given only in standard test conditions can not satisfy the requirements of customer in different conditioners. In order to provide cooling capacities of the air conditioner at different wet bulb temperatures of inlet air,the manufacturer of air conditioners needs to experiment with different IWB, t_{wb}so as to obtain the performance curve of the cooling capacity Q_0correspondingwith the changes of t_{wb1}. For instance, it was stipulated in the testing methods of packaged air conditioners in the Chinese national code JB 1370–73(1974) that there should be full performance curves obtained on each unit by experiments, ie the relation between outlet air condition and the cooling capacity at different wet bulb temperatures of the inlet air (relative humidity 60%), air flow rate ,and the condensing temperature. This is usually called the 'variable condition test'. In the variable condition test it takes a long time to change inlet air parametera and needs a very good air–prehandling system to ensure the tests are carried outadequately. Varying the air flow rate and condensing terperature is much easier and takes less time. If we consider the wet bulb temperature depression as aconstant value, the test on air conditioners will be greatly simplified. Experiments on changing wet bulb temperature of inlet air may be omitted. It is necessary only to measure wet bulb temperature depression, $\triangle t_{wb}$of the air conditioner at any a certified air rating. Thus, testing work could be simplified from measuring a curve stipulated in the national code to the measuring of only one point. Moreover, the requirements for air pre–handling equipment would also be minimized. It is necessary only to keep the IWB temperature stable during the testing process for measuring this point.

Cooling capacities of the air condictioner at different inlet air wet bulb temperature may be obtained by simple calculation according to the measured wet bulb depression $\triangle t_{wb}$and relative humidity of inlet and outletair $\varphi_1$and φ_2.

For example, take a point from data measured with the air conditioner of type Rp–1004 as original data to calculate the performance curve Q_0VS $t_{wb}$and then compare the calculation values with test values.

It is given originally that $t_{wb1}=27.55℃$,$t_{wb2}=22.67℃$ $\triangle t_{wb}=4.88℃$, $\Phi_1=50.2\%$, $\Phi_2=86.4\%$, atmospheric pressure B=0.99kPa, air rate $L=1.47m^3s^{-1}$. Projections based on these figures are given in Table 1. The calculation in table 1 shows that the deviations of cooling capacities obtained by calculation and from the curve actually measured are not more than 3.5%. This deviationdoes not go beyond the range of test deviation.

To obtain a complet the thermotechnical performance curve of the air conditioner. Another problem which occurs when testing an air conditioner is the inability to obtain a complete performance curve,even though many condition tests are carried out, because the changes which occur in the air conditioning parameters are not fully understood. By using the performance of t_{wb}–Φ it is unnecessary to test for changing parameters of the inlet air,and is also possible to obtain the performance curve of the two air condition parameters ($t_{wb}$and φ) and cooling capacities corresponding with inlet air parameters, by calculation and having only the test data of one air condition point. In this way it is possible to express the thermotechinal performance of the air conditioner more completely.

As is mentioned above we can obtain two performance curves for the air con-