START-UP OF AIR CONDITIONING SYSTEMS AFTER PERIODS OF SHUTDOWN

(HUMIDITY CONSIDERATIONS)

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ABSTRACT

In many cases the single most important energy conservation measure that can be taken is to turn equipment off when it is not needed. In the case of air conditioning, this generally means turning it off when occupants leave and turning it back on in time to have the space comfortable when they return.

In humid climates special problems are often encountered when a system is restarted after a period of shutdown. The temperature and humidity in the space rises during the period of shutdown. Unfortunately the latent load required to bring the space back to comfort conditions is usually much higher than the sensible load. Most methods of control are ill suited for this duty. This paper examines the response of various types of air conditioning systems during this recovery period and makes recommendations for system designers.

INTRODUCTION

During the summers of 1983 and 1984 Engineering Sciences, Inc. personnel performed field surveys of approximately one hundred buildings at U.S. Marine Corps Camp S. D. Butler, Okinawa, Japan. In the summer of 1985 similar field surveys were performed for sixty-six buildings at U. S. Naval Base, Subic Bay, Republic of the Philippines. The purpose of these studies was to identify equipment deficiencies in the air conditioning systems which affect its ability to provide the required comfort conditions in an energy efficient manner. In each case the field survey was then followed by a thorough analysis of the buildings and their air conditioning equipment.

First, the ESI staff performed rather extensive field tests on the buildings, their air conditioning equipment and its controls. Among the items measured (noted) were: air flows, dry and wet bulb air temperatures, air static pressures, water flows, flows, water/refrigerant temperatures, water pressure drops, all appropriate motor electrical data and nameplate information on all major equipment. All control equipment was exercised over its entire range of motion and readings retaken for all variables. In addition, dry and wet bulb air temperatures were taken throughout the conditioned spaces; and a thorough visual inspection was made of the equipment and the space that it served.

Using the data collected and manufacturer's literature (e.g., pump and fan curves, etc.) the operating point for each piece of equipment was determined. This, in turn, was used to analyze the operation of the air conditioning systems in whole. The operation of the systems under the various likely loadings they will experience during the course of a year could then be determined. (This was done through the use of a number of computer routines that ESI has developed for this purpose.)

Having analyzed the systems in this fashion, deficiencies (in design and/or condition) which cause them to fail to perform their desired function in an efficient manner could be identified. Recommendations were then made to correct these problems and supporting calculations performed to assess the economic attractiveness of each investment.

As might be expected, many of the problems observed in the course of the work were humidity related. Both Okinawa and Subic Bay have very humid climates. 97.5% design conditions are 88°/80° and 93°/79° for Okinawa and Subic Bay respectively. As a result, cooling coils frequently see loads which have very low sensible heat ratios (50% or less). Under these conditions everything must be close to perfect to avoid humidity related problems. Unfortunately, the opportunity existed to observe numerous such problems.

Although there are many different types of humidity problems that can occur, those most commonly found in the course of this work were:

1. Condensation on supply air diffusers and ceiling panels during normal system operation.

2. Condensation on diffusers and ceiling panels at the time of start-up after a period of equipment shutdown. It was frequently reported that unoccupied shutdown had been discontinued because upon restarting moisture would condense on the diffusers and cause maintenance problems and in some cases, property damage. This condition was verified by the field team. The magnitude of the resulting energy waste makes the solution of this problem of considerable importance.

The problem of condensation during normal operation (coupled with generally poor humidity control) was addressed by the author in a previous paper (1). This paper concentrates exclusively on the problem of start-up after a period of shutdown. The issue of selecting a control strategy for start-up is closely related to that of selecting a method of capacity control for normal operation because the same equipment is hopefully used for both purposes. The purpose of this paper is, therefore, to select a method of air conditioning control which will function well under normal operation (being careful of the humidity problems which tend to occur at part load conditions) and which will regain control of room conditions after shutdown without causing condensation problems. The previous paper by Todd (1) should be referenced for a discussion of the general issue of part load operation.

THE NATURE OF THE PROBLEM

During the course of the field investigation it was discovered that night shutdown of the air conditioning systems had been largely discontinued in Okinawa due to problems associated with restarting it in the mornings. Three basic problems were reported. First, due to improper control, compressors were being damaged during the shutdown. The lack of a pump down cycle and oil migration were the principal causes for these events. Second, the buildings were never able to "catch up" with the latent load during the period of occupancy. Improper capacity control during normal operation is at the source of this difficulty. Finally, there was considerable condensation on diffusers and the surrounding ceiling tile when the systems were restarted.

The problem of compressor damage can be solved by properly interrupting the internal controls of the units. The difficulties associated with failure to catch up with latent load can be addressed by retrofitting the systems with capacity control systems which are better suited to the particular requirements of the the high humidity environment.

The issue of condensation during start-up, however, requires information which connot be found in the literature. For this reason an analysis was undertaken to explore the events which occur during the first minutes after the system is restarted.

Most simply, condensation takes place when moist air flows past surfaces which have been cooled below its dewpoint. In air conditioning this occurs when the cold supply air being delivered into the space induces warm, moist room air to flow over surfaces which have been cooled by the supply air. These areas generally include the metal of the supply air diffuser and the ceiling tile surrounding the diffuser. The problem is often aggravated by the leakage of supply air above the ceiling at the point where the flexible connector joins the diffuser. Figure 1 shows a typical ceiling diffuser. Whenever the mixture of supply air and room air comes into contact with a surface which is below its dew point, condensation can occur.

The likelihood of condensation occuring is, of course, dependent upon the flow pattern of induced room air and the extent to which the surrounding surfaces are cooled. This is highly specific to the particular supply air diffuser design and location. Field observations have shown that some diffuser designs are much more likely than others to have problems. Unfortunately the actual heat transfer and fluid flow problem is very complex and no analytical or experimental solutions can be found in the literature. No quantitative guidance is available to the designer.

It has been found that, for a given diffuser design and location, the difference between the dew point of the room air and the dry bulb temperature of the supply air being discharged is a good indicator of condensation . The approach taken in this study was to develop a model of the room and the air conditioning equipment serving it which is capable of predicting the room dew point and supply air temperature versus time during the start-up period. This model was then to be used to test the behavior of various control strategies. Control strategies can then be compared on the basis of how well they perform in terms of bringing the room conditions down to those desired while keeping the supply air as close as possible to the room's air dewpoint. (The dew point depressions from the analyses can be compared with experimental observations of the conditions under which condensation was found to occur to gain some absolute indication as to whether condensation will occur for a particular design.)

THE COIL MODEL

A key element of the analysis is the model used to predict the behavior of the cooling coil. William Goodman (2) developed an approach for treating wet cooling coils as heat exchangers in which the driving potential is the difference between the enthalpy of the entering air and that of saturated air at the temperature of the entering water or refrigerant. O. J. Nussbaum (3) extended this using the notion of heat exchanger "effectiveness" and the "NTU" approach (4) showing that most coils can be modeled very accurately as counterflow heat exchangers. This author has used this basis to develop a coil model which uses one known operating point (i.e., entering and leaving dry bulb and wet bulb, air flow, water flow and entering water temperature) to determine constants which represent the physical characteristics of the coil in the effectiveness calculations. These constants are then used to calculate the effectiveness of the coil under other conditions of interest. With effectiveness known, the enthalpy of the leaving air can be determined. Since wet bulb lines are almost isenthalpic, the wet bulb temperature of the leaving air can be determined from its enthalpy through a simple curve fit. The dry bulb temperature of the leaving air can be calculated using the concept of the "wet bulb depression" ratio defined as:

$$R = (LDB-LWB)/(EDB-EWB) \tag{1}$$

Where:

LDB, LWB Leaving dry, wet bulb temperatures

EDB, EWB Entering dry, wet bulb temperatures

This can be calculated for any known operating point. It can also be shown that:

$$R = f_1(CFM, GPM, C_1) \quad (2)$$

Where:

CFM Air flow

GPM Water flow

C_1 A constant representing the physical characteristics of the coil Equation 2 can be solved for C_1 using the value of R calculated using Equation 1 at the known operating point. The value of C_1 can then be used in Equation 2 for any other point of interest.

In a similar fashion, the effectiveness can be calculated for any known operating point using:

$$E = (h_{out}-h_{in})_{min}/(h_{air}-h_{water})_{in} \quad (3)$$

Where:

$(h_{out}-h_{in})_{min}$ Actual enthalpy change for the stream having the smaller capacity

$h_{air,in}$ Enthalpy of entering air

$h_{water,in}$ Enthalpy of saturated air; the temperature of the entering water

As before, it can be shown that:

$$E = f_2(CFM, GPM, C_2) \quad (4)$$

Where:

CFM Air flow

GPM Water flow

C_2 A constant representing the physical characteristics of the coil

Equation 4 can be solved for C_2 using the value of effectiveness determined at any single known operating point from Equation 3. C_2 can then be used in Equation 4 to determine the effectiveness of the coil under any desired condition. Equation 2 can be used to fully specify the condition of the air leaving the coil.

THE ROOM LOAD MODEL

An energy balance on the air in the space yields:

$$(Mc)_{air}\, dT_{ra}/dt = hA\,(T_w-T_{ra}) + 1.08\ SA\,(T_{sa}-T_{ra}) + 1.08\ XFIL\,(T_a-T_{ra}) + Q_{ig} \quad (5)$$

Where:

$(Mc)_{air}$ Capacity of air in room (BTU/°F)

T_{ra} Room air dry bulb temp (°F)

hA Heat transfer coefficient from room furnishings to air (BTU/HR-°F)

T_w Wall/furnishings surface temp (°F)

SA Supply air quantity (CFM)

T_{sa} Supply air dry bulb temp (°F)

XFIL Infiltration quantity (CFM)

T_a Ambient air dry bulb temp (°F)

Q_{ig} Sensible internal heat generation (BTU/HR)

In a similar fashion a mass balance on the moisture in the room yields:

$$M_{air}\, dW_{ra}/dt = h_mA_m(W_w-W_{ra}) + 4.5\ SA\,(W_{sa}-W_{ra}) + 4.5\ XFIL\,(W_a-W_{ra}) + M_{ig} \quad (6)$$

Where:

M_{air} Mass of air in room (LBM)

W_{ra} Humidity ratio of room air (LBM/LBM)

h_mA_m Mass transfer coefficient from room furnishings to air (LBM/HR)

W_w Humidity ratio associated with furnishing surfaces (LBM/LBM)

W_{sa} Humidity ratio of supply air (LBM/LBM)

W_a Humidity ratio of ambient air (LBM/LBM)

M_{ig} Internal moisture generation (LBM/HR)

The response of the wall surface to an attempt to pull the room air temperature down can be thought of as a disturbance which is superimposed on the temperature distribution that exists when the equipment is restarted. In fact, at typical morning start-up times the load on the space from the exterior envelope elements is very small. This can be seen by examining the Cooling Load Temperature Differences (5) at this time of day. The walls/furnishings might, therefore, be thought of as being essentially at the same temperature as the room air at the time of startup. As the room air begins to drop the walls/furnishings will give up heat to the air and the wall/furnishing surface temperature will drop. The actual transient heat transfer situation is very complex. (i.e., There are numerous objects in the space. Each has its own geometry and thermal properties. The conduction of heat from within each object to its surface will differ widely from one to the next.) Due to the short time interval of interest, the treatment chosen in this analysis was to treat the walls/furnishings as a single equivalent lumped capacity (with negligible internal resistance to

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conduction) of surface area, A, and thermal capacitance, MC. The rate at which the mass gives up heat and drops in temperature is determined by the values for A and MC chosen. The analyses which follow are based upon values which are felt to be physically "reasonable" for the buildings being considered (i.e., Furnishings area of 7.5 SF per SF of floor space. Heat transfer coefficient of 1.5 BTU/HR-°F-SF. Thermal capacity of 1.2 BTU/°F per SF of floor space.) These values are estimates only and are certainly not valid for all buildings. Field tests are planned for July 1986 in Okinawa to gather information from which parameters can be calculated for the various types of buildings and construction of interest.

A similar approach can be taken to the moisture released from the room furnishings to the air in the space. Much less is found in the literature with respect to this exchange of moisture. Tsuchiya, as reported by T. Kusuda (7) experimentally determined parameters which fit data collected in studies of a typical Japanese residence. T. Kusuda and M. Miki (8) later determined that the values reported earlier tend to over predict the rate of moisture. For the time scale of interest in the start-up period (less than 60 minutes), it is suggested that the amount of moisture given up by the furnishings will be negligible with respect to that brought in by infiltration. For this reason, this term has been neglected in the analyses which follow. (The field tests planned for July 1986 in Okinawa will also address this issue and a determination made as to an appropriate treatment.)

Due to the time variation in wall surface temperature and the rather complex relationship between supply air condition and room air condition, Equations 5 and 6 are highly nonlinear. For this reason a difference approximation was made for the differential equations and the room temperature, T_{ra}, and room humidity ratio, W_{ra}, determined at discrete time intervals from their values at the previous time interval. Runs were made to explore the sensitivity of the results to the time increment chosen. It was found that a time increment of 15 seconds gave results which were consistent with those of shorter intervals.

RESULTS OF THE ANALYSES

Analyses were done for the types of systems most commonly found in Okinawa and at Subic Bay. In each case the room air was assumed to be 81 °F and 0.019 LBM_w/LBM_a. The furnishings were assumed to be in thermal equilibrium with the room air at 81 °F. A representative 20,000 SF space was chosen for the analysis. Infiltration when no ventilation air is being brought in at the unit was assumed to be approximately 1.1 air changes per hour. Ambient air was assumed to be 80/76 (db/wb). These values are typical of summer start-up conditions in Okinawa and serve to illuminate the problems that will be faced in the humid climate.

Figures 2, 3 and 4 show the results for a coil which runs at full capacity until it reaches some setpoint temperature. This might be typical of a DX coil or a chilled water coil operated without modulating control. Figure 2 shows room air temperature and humidity ratio versus time. Figure 3 shows the same process plotted on a psychrometric chart. Figure 4 shows the depression of supply air below room air dew point versus time. As can be seen, the room air is quickly brought down to the desired 78 °F. Unfortunately the humidity in the room at this time is approximately 0.0145 (LBM_w/LBM_a) instead of the 0.0125 that one would desire. From this it is clear that the most desirable situation would be to have the process follow a line on Figure 3 which goes directly to the desired room condition. Figure 4 shows what is perhaps the worst aspect of this method of control. The supply air is from 10 °F to 14 °F below the room dewpoint throughout the pull down period. Field observations have shown that this is almost certain to cause condensation problems.

One way of solving the dew point problem is to by-pass a fixed portion of the supply air around the cooling coil. Figures 5, 6 and 7 display the same information for a system in which 50% of the air is by-passed. As can be seen the room is again brought down to 78 °F very quickly. Again, however, the room humidity ratio at this time (although improved) is still higher than desired. Figure 7 shows that the desired effect was achieved in that supply air ranges from 8 °F to 2 °F below the room dew point. Experience has shown that this system is not likely to experience condensation problems.

With both of these systems one could either stop at 78 °F and accept the higher humidity or continue cooling until the desired humidity was achieved. It should be pointed out that the 0.0140 with the 50% by-pass system is actually not much worse than what is frequently found during normal operation. In either case the difficulty arises when the unit goes off. The infiltration continues to bring a large amount of moisture into the space. The sensible load from the furnishings has been significantly reduced as the surface temperature was brought down. Figure 8 shows the results for the 50% by-pass system where the coil is turned off at 77.5 °F and back on at 78.5 °F. As can be seen, the sensible load is not sufficient to give enough run time to keep up with the humidity being brought in by infiltration. This makes an important point about start-up in humid climates. The sensible load available in the furnishings is important in bringing the room back under control after a shutdown period. Once it is gone it is very difficult, without reheat, to regain and maintain humidity control. For this reason care should be taken in timing the start-up so that the space comes to temperature just in time for occupancy. When the occupants arrive and lights are turned on it is much easier to maintain humidity control.

In general the sensible heat ratio of the coil can be decreased by increasing the fraction of air which is by-passed. The recovery time required increases with the by-pass factor. It is possible that the by-pass fraction required to keep supply air temperatures at acceptable values may result in unacceptably long recovery periods.

Figures 9 and 10 show the results for a chilled water coil with a valve which is controlled by room air dry bulb temperature. Values are shown for thermostat setpoints of 78 °F and 76 °F. In both cases the throttling range is 4 °F. As can be seen, the 78 °F setpoint brings the room down to temperature very quickly but never gets the humidity down to the desired level. The 76 °F setpoint is capable of bringing the humidity down to the desired level but does so at the expense of over cooling. Figure 10 shows that the supply temperature is well below the room dew point throughout the entire recovery period. This is almost certain to cause condensation problems.

Like the DX system, the chilled water valve unit is unable to hold the space humidity down after the walls have had sufficient time to cool. If the sensible load does not increase in the space, the humdity will begin to creep upward and may reach unacceptable levels before occupants arrive. In the case of the chilled water valve this tendency is made worse by the fact that the valve tends to warm the coil and increase the sensible heat ratio whenever the sensible load is small. This issue has been discussed previously by this author (1) and is one of the primary reasons that chilled water valves are not appropriate for humid climates.

Variable air volume is often recommended because of its energy efficiency. Unfortunately it does not work well in humid climates. In those systems in which the chilled water coil is uncontrolled the supply air temperature drops rapidly as the air volume is reduced. This frequently leads to condensation problems. In those systems in which the chilled water coil has a valve controlled to give a fixed discharge air temperature, the system suffers the usual problems associated with chilled water valves (i.e., loss of latent capacity at low loads).

Figures 11 and 12 summarize the performance of VAV systems (with wild coils) during the recovery period. As can be seen, the system recovers nicely but produces a supply air temperature which stays well below the room's dew point throughout the pull down period. This will almost certainly lead to condensation problems.

The "face and by-pass" system offers the improved sensible heat ratio of the VAV system without the problem of low supply air temperature. As discussed by the author previously (1), the face and by-pass system appears to be one of the more attractive alternatives for modulating control in humid climates. Figures 13 and 14 show the results for a system with face and by-pass dampers controlled by room air dry bulb temperature and an uncontrolled cooling coil. The system achieves good humidity control and pulls the system down to temperature within a brief period. While the supply air temperature is significantly below the room air's dew point at first, the depression quickly vanishes. It is unlikely that the system would experience condensation problems. A simple modification to the controls, however, would insure that no problems occur. During the first phase of the start-up, the dampers could be positioned for some fixed by-pass fraction (chosen to deliver a safely high supply air temperature and a reasonable recovery period). When some lower temperature is reached the dampers would be switched to normal modulation in response to the room thermostat. Figures 15 and 16 display the results for such a system. As is seen, the supply air temperature is now safe throughout the recovery period. The control modifications necessary to achieve this operation are minor and the increased complexity is minimal.

As has been suggested previously (1), the benefits of face and by-pass operation and the fan energy savings of VAV can be combined by controlling the fan speed on a face and by-pass unit in response to the room load and the face and by-pass dampers to give some set (safe) discharge air temperature. This system would automatically achieve the objectives of start-up without further modification. (Latent capacity would be preserved while supply air temperature would be kept safely high.)

Control strategies which depend upon humidity measurement have not been investigated. Current experience has shown that humidity sensors are not reliable under the field conditions and maintenance constraints generally found in building HVAC applications. The approach taken in this work has been to use dry bulb temperature and a knowledge of the most likely loads that the system will see to select control strategies which, by their basic nature, give good humidity control.

CONCLUSIONS

1. The prediction of whether a given supply air diffuser will condense moisture during start-up is highly site specific and cannot be made on the basis of field observations and the analyses reported in this paper. Further work is needed to solve the air flow problem for common air diffuser designs and orientations. Objective guidance can then be prepared to aid the design engineer in selecting air distribution equipment for humid climates.

2. For any given supply air diffuser, field observations have shown that the extent to which the supply air dry bulb temperature is below the room air dew point is an excellent indicator of when condensation can be expected to occur.

3. This paper presents an analysis technique which can be used to evaluate the suitability of various control strategies for the recovery period after a period of shutdown. The criteria useful in this evaluation are: the time required to bring the room down to the desired temperature, the humidity in the room when the desired tempera-

ture is reached and the relationship between supply air dry bulb and the room dew point temperature during the recovery period.

4. When modulating control is not required, a horizontally (face) split coil performs well during the start-up period. If, for instance, the coil has two refrigerant circuits one solenoid is kept closed throughout the start-up. This fixed by-pass keeps the sensible heat ratio of the coil down and the mixed temperature of the supply air up as desired. In a previous work (1) this system has been shown to perform well during normal part load operation with the low sensible heat ratios often found in humid climates.

5. When modulating control is required, the face and by-pass system with an override for fixed by-pass during the early minutes of start-up is shown to be the preferred control. This control strategy has also been previously shown (1) to perform well under likely loads found in humid climates and appears to be the most attractive system in all respects. (A modified version in which the speed of the fan is controlled by the room load and the face and by-pass dampers may be be superior, in that it achieves the benefit of fan savings due to VAV operation.)

6. In general, it has been observed that the problems associated with restarting a unit in a humid climate originate with the fact that there is not sufficient sensible load to allow the unit to pull the room humidity down as desired without overcooling the space. For this reason the heat stored in the walls and furnishings should be thought of as an asset which is to be used wisely. With this in mind, the start-up should be carefully timed to coincide with the arrival of the occupants of the space. If the building is cooled off too long before the occupants arrive the humidity will creep upward and may become unacceptably high.

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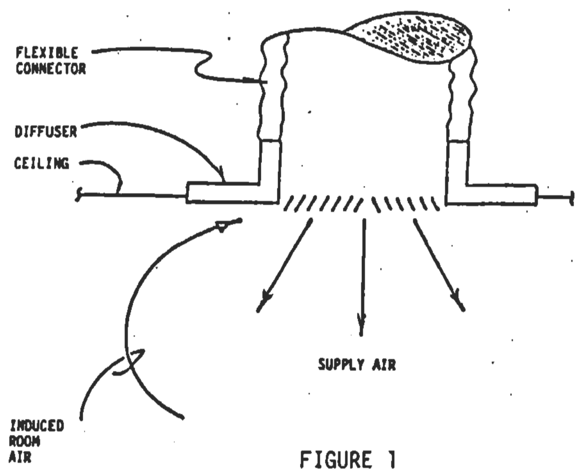


FIGURE 1
TYPICAL CEILING DIFFUSER

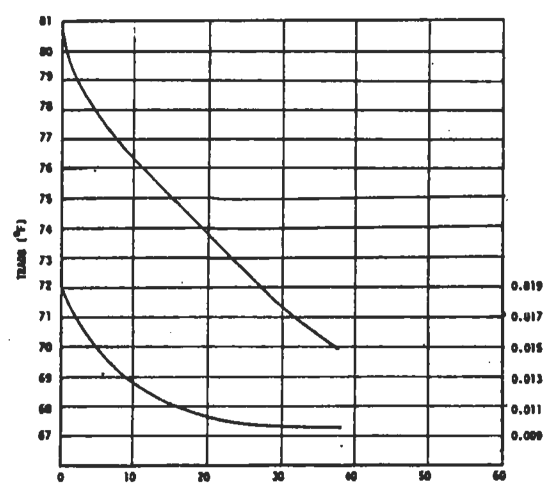


FIGURE 2
DX COIL WITHOUT CONTROLS

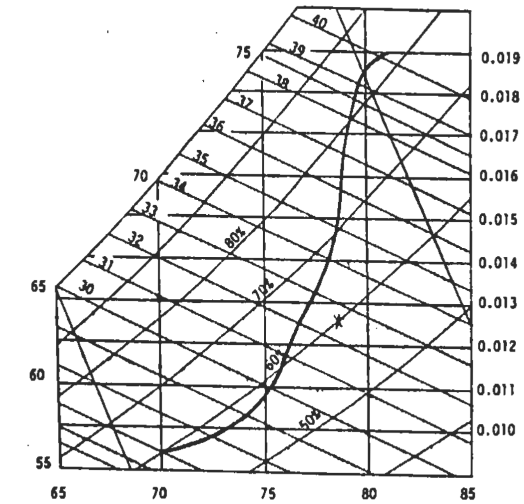


FIGURE 3

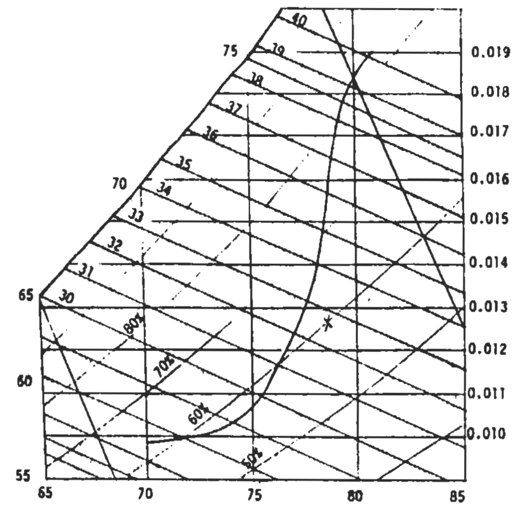


FIGURE 6

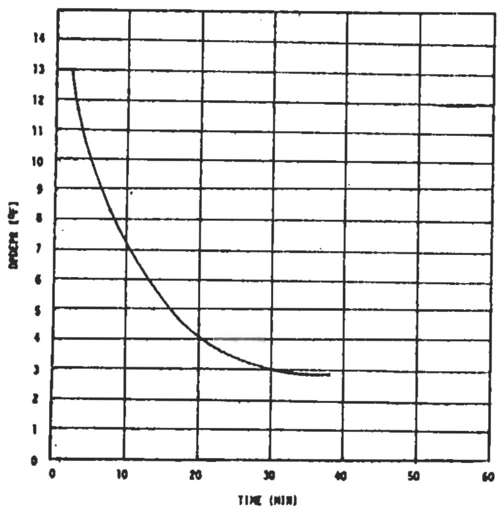


FIGURE 4

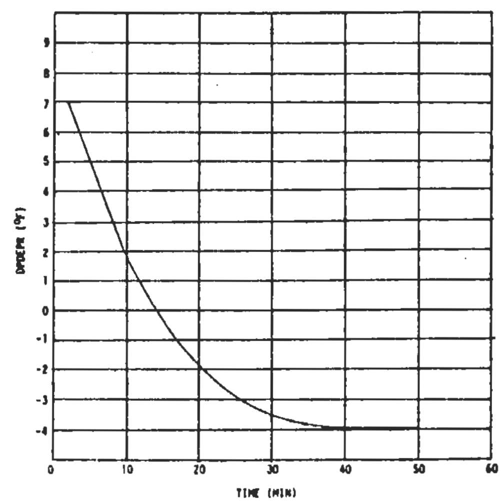


FIGURE 7

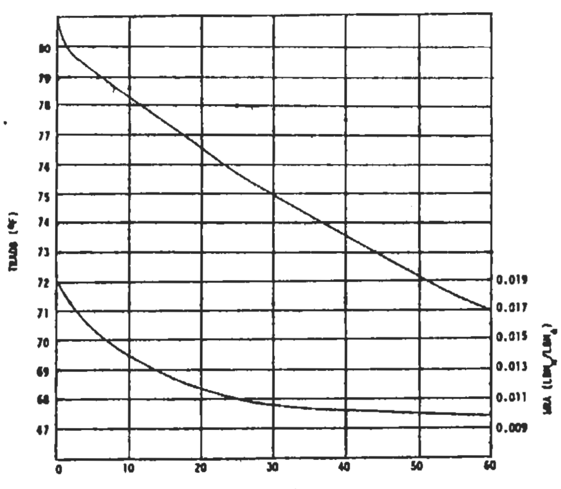


FIGURE 5

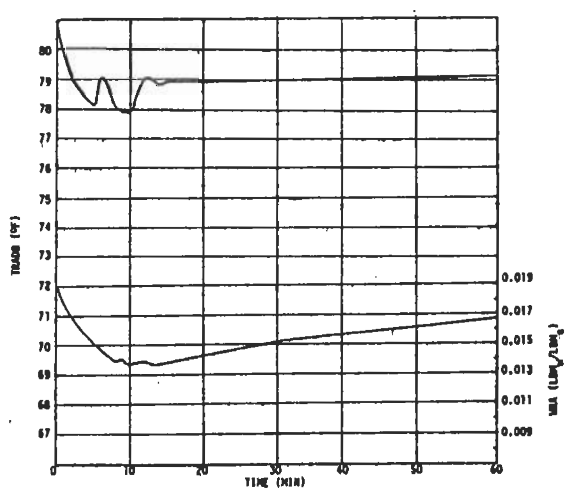


FIGURE 8

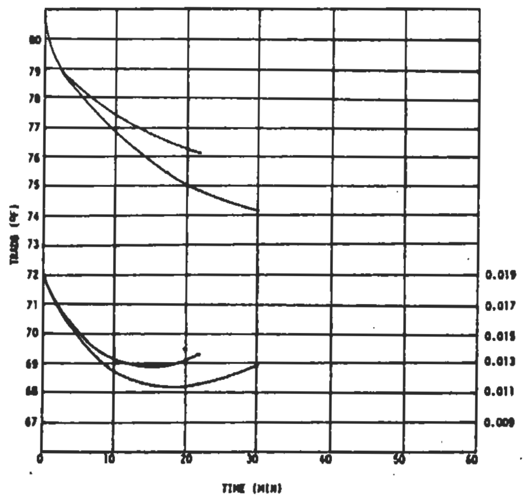
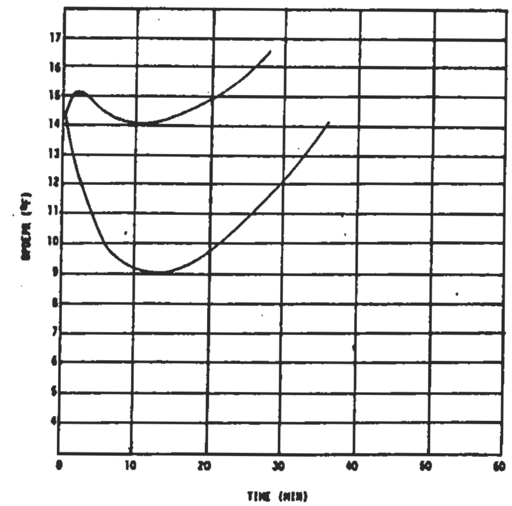


FIGURE 9



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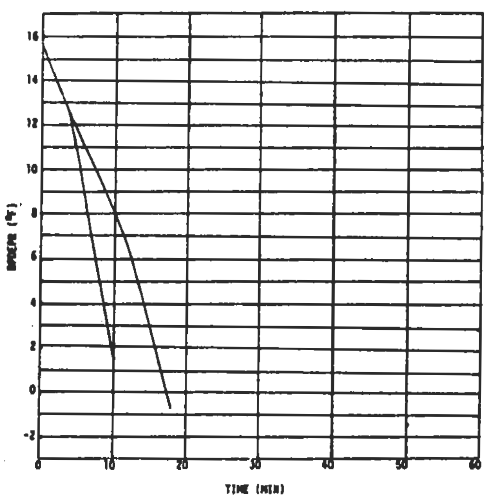

FIGURE 10

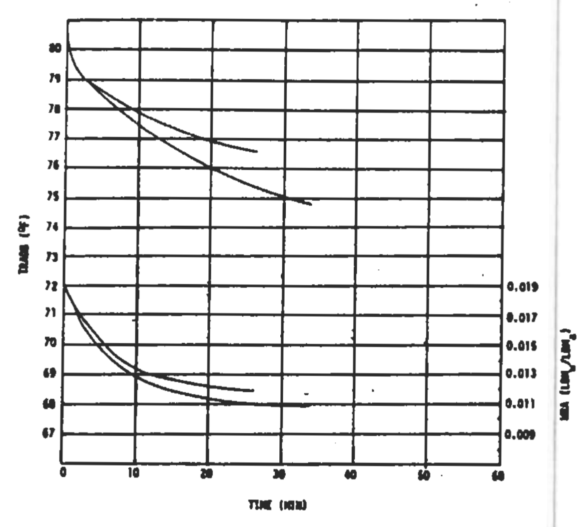

FIGURE 13

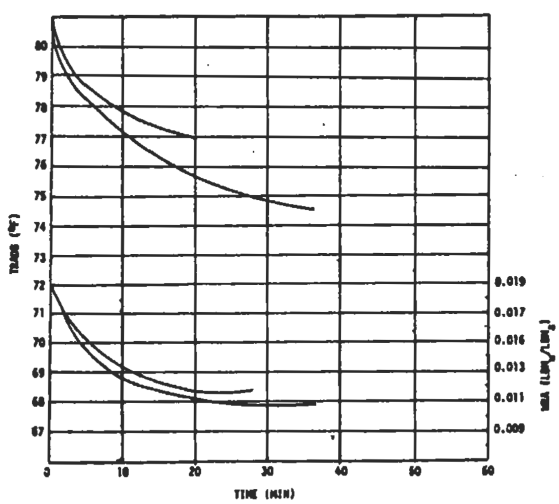

FIGURE 11

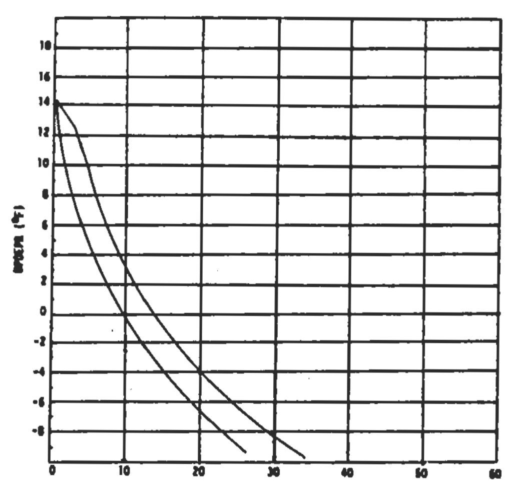

FIGURE 14

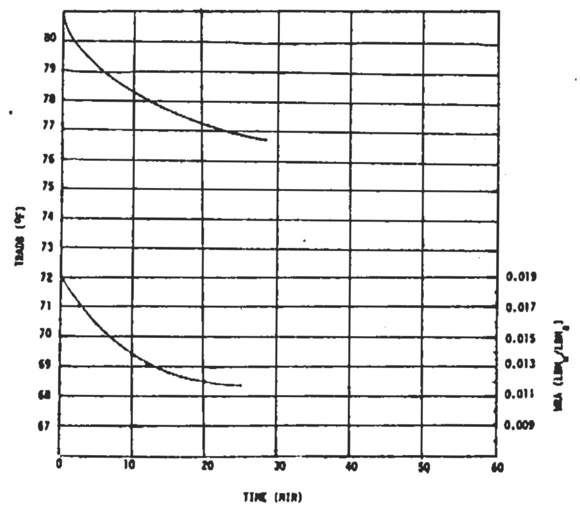

FIGURE 15

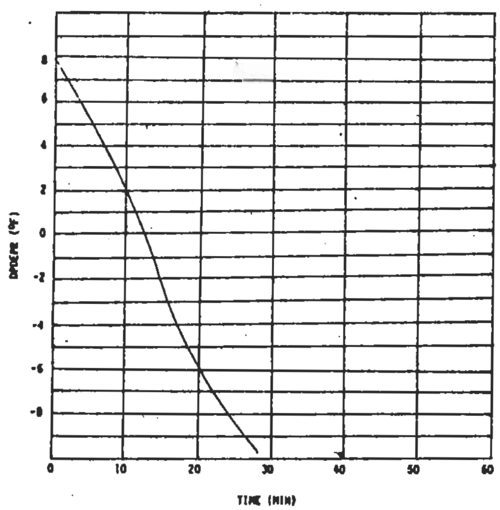

FIGURE 16