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Thermal Comfort and Cold Air Distribution

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ABSTRACT

Cold air distribution systems supply air at temperatures between 38°F and 51°F. Cold air distribution systems are increasingly attractive when used in conjunction with ice storage systems to shave peak load by shifting the demand to off-peak hours. They also require less operating and capital costs because they use smaller fans, ducts, piping, and pumps. However, an important issue in design and application of cold air systems is the effect on occupant comfort.

There are several techniques and methodologies that practitioners use for evaluation of conventional air distribution systems. Among these is the Air Diffusion Performance Index (ADPI). It is widely used in the U.S. and is referenced in the 1993 ASHRAE Handbook—Fundamentals. However, this technique is based on empirical correlations obtained from tests conducted with conventional systems and it cannot be guaranteed that they will be equally applicable to cold air systems. This study was undertaken, therefore, to extend the existing techniques (especially the Air Diffusion Performance Index) to applications where cold air distribution systems are utilized.

This work presents a critical review of the evolution of the ADPI technique and offers several recommendations for developing a firm foundation for future room air distribution research. For this work, no new comfort tests were conducted. However, the experimental data of the research conducted by Nevins and Miller were employed for extending the existing ADPI methodology to cold air systems. The tests conducted by Nevins and Miller covered a wide range of discharge flow rates and temperature differences for different types of diffusers. A large number of those test conditions could be categorized as cold air conditions.

As a first step in extending the existing ADPI to cold air systems, the local velocities in the occupied zone are correlated to the total momentum of the inlet air jet. These correlations can be used for directly relating the ADPI to the supply air momentum.

An approach for determining the ADPI of cold air systems is introduced. In this approach, a one-step procedure is adopted where the ADPI is directly linked to the momentum number of the supply air. A set of curves correlating the ADPI to the momentum number of the room/diffuser combination under different loads are presented.

INTRODUCTION

Cold air distribution systems supply air at temperatures between 38°F and 51°F (3°C- 11°C), with emphasis on systems supplying air at 44°F (7°C). Cold air distribution systems are increasingly attractive when used in conjunction with ice storage systems to shave peak load by shifting the demand to off-peak hours. In this technology, ice makers are used to make and store ice during off-peak hours (at night) and use it during the day as the cooling source that can easily produce air at 44°F. Cold air distribution systems require less operating and capital costs because they use smaller fans, ducts, piping, and pumps.

As with any new technology, there remain some issues that are of concern to engineers and need to be addressed. One such issue in cold air distribution systems is the issue of comfort and adequate room air motion. The existing ASHRAE Handbook and ASHRAE standards offer a wealth of information about conventional air distribution systems and related human comfort issues and design parameters. However, the diffusion of air at lower temperatures requires a different emphasis on design parameters from that used in conventional systems. The object of heating, ventilating, and air-conditioning systems is to "create the proper combination of temperature, humidity, and air motion in the occupied zone

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of the conditioned room—from floor to 6 ft above floor level—to satisfy the comfort requirements of the occupants" (Nevins and Miller 1972). Thermal comfort is not directly measurable but, instead, is a perception that is evaluated by each individual on a subliminal level. Comfort is the subjective physiological and psychological reaction of each individual to environmental parameters. This perception is triggered by as many as 15 different measurable environmental parameters as identified by Nevins (1968), the most significant of those being air temperature, air speed, air humidity, and thermal radiation. Acceptability of the combined effect of these parameters is influenced by the personal factors of activity level and amount of clothing. Thermal comfort is defined in ASHRAE Standard 55 (ASHRAE 1992) as "the condition of mind that expresses satisfaction with the thermal environment." The standard specifies those conditions that 80% or more of the occupants will find thermally acceptable. Discussion of each parameter can be found in the *ASHRAE Handbook—Fundamentals*.

Over the past fifty years, researchers have introduced various thermal indices to evaluate the state of the thermal comfort of the occupants. Some of these indices evaluate the effect of environment variables such as activity level, clo-value, air temperature, air velocity, mean radiant temperature, and air humidity on human comfort, as stated in *ASHRAE Fundamentals* (ASHRAE 1993). Others, such as the Air Diffusion Performance Index (ADPI), evaluate the performance of air diffusion systems based only on local temperature and velocity.

This work is focused on the ADPI technique and its application to cold air distribution systems. However, during a background investigation of early studies related to room-air distribution, ADPI, and human comfort, several concerns over the validity of the historical basis of the research were identified. As a result, it was decided appropriate to include a critical review of the existing literature related to comfort, effective draft temperature, and ADPI. In this review, we point out concerns that we have over the existing ADPI criterion and its development to its current state. Several relevant recommendations are presented for future work. Following the review of the existing comfort design parameters, especially Air Diffusion Performance Index (ADPI), this work presents, based on the available limited information, an updated ADPI methodology for the design of air distribution systems for use with cold air systems.

BACKGROUND REVIEW AND CONCERNS OVER THE EXISTING COMFORT CRITERIA

This background review is focused on the historical and technical development of ADPI criterion and is not intended to present an overview of the existing comfort criteria other than ADPI. Readers are encouraged to refer to Fountain (1991) for a comprehensive review of other comfort criteria. Interest in and research activities on thermal comfort date back to the early 1920s. Houghten and Yaglou in 1923 and 1924 developed the early ASHVE "effective temperature" concept. Their work was conducted in test rooms with estimated velocities of about 15 to 25 fpm. Following these studies, Nelson and Stewart (1938) were the first to develop a comfort criterion with application to air diffusion performance. However, their work was dismissed by later researchers because it was based on limited observations and did not cover a wide and diverse group of subjects.

Houghten et al. (1938) explored the reaction of ten male subjects, ranging from 18 to 30 years old, to different combinations of temperature and velocity. In this work, effect of the drafts on skin temperature and feeling of warmth was reported. Houghten et al. (1938) data were reported in terms of fixed ambient temperatures. That is, the testing environment stayed at 70° F while cooler drafts were blown at the subject either on the neck or ankles. Results are given for the effects of air velocity ranges of 10 to 90 fpm and air temperatures of 65°F through 70°F. The measured results were skin temperature differences due to the drafts and the percentage of subjects indicating a perception of comfort. Their results in the original form are shown in Figure 1. Houghten's data and comfort curves have been widely accepted and extensively used as the primary basis for assessment of human comfort in enclosed spaces as it related to temperature and velocity. Such a limited survey sample is a good beginning to the important study of human comfort in enclosed spaces, but it is hardly definitive. In the discussion following Houghten's paper, J.J. Aeberly observed that "the paper does not establish a definite relationship between drafts and man's response to them and is far from the results which we hope ultimately to obtain." Despite these warnings, subsequent researchers have adopted and generalized Houghten's comfort data without further extension of its results or verification of its accuracy.

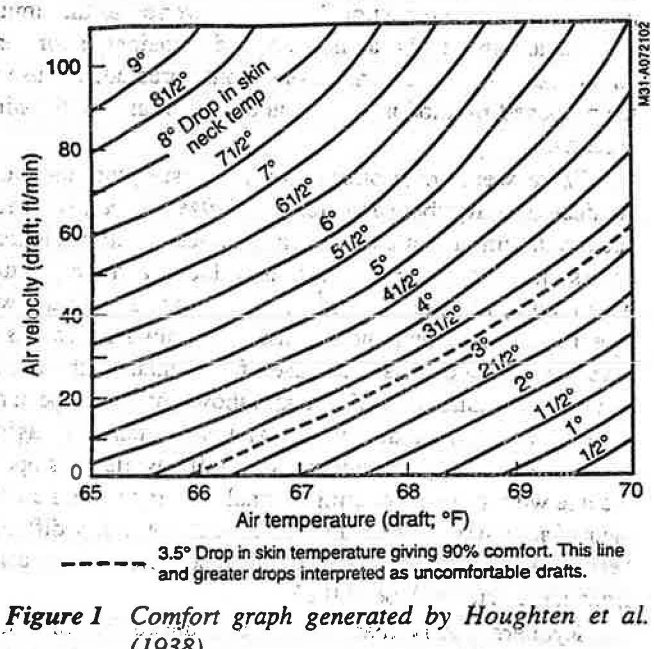

Figure 1 *Comfort graph generated by Houghten et al. (1938).*

Rydberg and Norback (1949) presented a relationship, given by Equation 1, between the local air velocity and temperature drop to account for the local comfort conditions. In this equation, the effective draft temperature, θ, indicates the difference between the "draft," at velocity v, air temperature, and the temperature of still air that would give the same cooling effect on the human body.

$$\theta = \Delta T - 0.07v \tag{1}$$

In this correlation, the effect of humidity was ignored and the coefficient 0.07 was obtained from tests conducted by Norback (1946) in which subjects were placed in a horizontal jet of hot air and the velocity and the excess temperature of the jet were regulated so that the jet, according to the subjects, caused the same temperature feeling as the zero velocity surroundings. In their work, leading to Equation 1, Rydberg and Norback limited the application of their equations to free jets (jets that are not disturbed by surrounding walls and in which no guide vanes are employed). They also assumed that the ratio between the excess temperature and velocity was constant in all parts of the free jet. This assumption was questioned by Nottage, who reviewed the paper and noted that "in non-isothermal jets, the temperature has been shown by other experiments to equalize more rapidly than the velocity." It was his recommendation that this assumption should be used advisedly. Nottage added in his review that "effective temperature as used in this country involves comfort sensations which are not simple heat transfer phenomena and which should not be oversimplified to a general linear equation without great caution."

Humphreys, in his review of Rydberg and Norback's work, stated that "the proposed method of predicting draft is open to question, however, and any attempt to make practical application of the equations given for this purpose should be made with extreme caution." He also referred to the limitations, stated above, that authors applied to their equations and emphasized that "such limitations would preclude the use of the proposed method in the actual design of air conditioning systems."

There were some other important issues that the work conducted by Rydberg and Norback (1949) failed to address, such as the impact of clothing on the effect of draft. Figure 2 shows the relationship between air velocity and temperature drop required for feelings of equal warmth. This graph was presented by Rydberg and Norback and shows the results of several researchers that were used for estimating the coefficient 0.07 of Equation 1. This graph shows that the slope of the three curves by Norback, Weiss, and Houghten and Yaglou, with subjects stripped to the waist, is 0.07, while the slope of the line with subjects wearing normal clothing gives a coefficient of about 0.03. Use of this coefficient would give different "effective air temperatures" than those obtained by using Equation 1 with a slope of 0.07.

Rydberg and Norback (1949) introduced the concept of room "control" or "reference" temperature (the room control

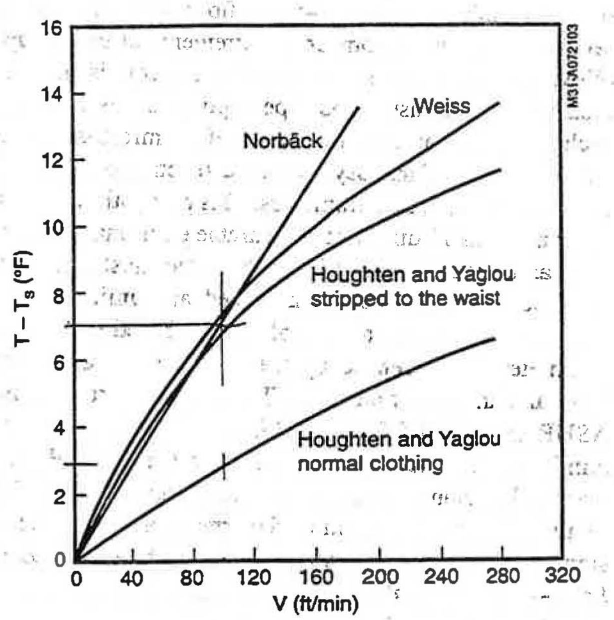

Figure 2 Relationship between local air velocity and temperature drop required for feeling of equal warmth.

temperature is the dry-bulb temperature in the center of the test room at 30 in. from the floor) and claimed that the draft applied to the body was entirely independent of this room control temperature. Their assumption that the ratio between the excess temperature and the velocity is constant implied that equivalent conditions of draft would be obtained with room and jet temperatures of either 68°F and 75°F or 75°F and 82°F, respectively. This issue was also raised by Humphreys in his review of their paper. Rydberg and Norback (1949) responded to this question by stating that their experiments were conducted over room ("control") temperatures ranging from 64°F to 72°F and no changes in the draft value were noticed. However, they stated, "if room air temperature is changed considerably from these values, the conditions will of course alter."

Following Rydberg and Norback, Koestel and Tuve (1955) attempted to determine optimum comfort conditions with minimum drafts in occupied spaces for high-volume air distribution systems with emphasis on heating with low supply temperature differences in small rooms. They kept their room control temperature at 73°F and assumed still air (zero control velocity) for control point during all their tests. They employed Houghten's comfort data (Figure 1) and Rydberg and Norback's local effective draft temperature concept for evaluating the percentage tolerability of their test data. Straub (1955) recommended a modification to Koestel and Tuve's approach by suggesting a control velocity of 30 fpm instead of still air assumption. Based on this recommendation, the expression for the effective draft temperature was reduced to:

$$\theta = (T - T_c) - 0.07 \cdot (v30) \tag{2}$$

The recommended control velocity of 30 fpm was based on the valuable opinion and observation of an experienced person rather than experimental observations. However, it is more reasonable to use an average velocity for the control point because keeping a control velocity of 30 fpm over the length of an experiment is a rather difficult task, and this quantity should be closely monitored and measured during a test rather than kept constant. It should be noted here that the original data of Houghten, on which the comfort conditions are based, did not address either the control temperature or the control velocity. However, Straub's suggestion was accepted, and future researchers such as Reinmann et al. (1959) employed his recommendation of using a control point velocity of 30 fpm. Reinmann et al. (1959) studied comfort issues related to three room air distribution systems for summer cooling. Their control point was assumed to be at 76°F and 30 fpm. They extrapolated Houghten's comfort data for use in their study and introduced Houghten's comfort graph, shown in Figure 3, which is based on temperature differences rather than fixed temperatures (this comfort graph will be referred to as "the modified comfort graph of Houghten").

The use of different definitions and values for control conditions by researchers has made later studies incompatible with each other and with Houghten's original data. Control conditions are used as a basis from which departures in temperature and velocity are considered and obtained from Houghten's modified comfort graphs. The basic idea behind this approach is that differences in temperature and increases in velocity that result from the room-air distribution are responsible for the discomfort felt in enclosed spaces. Yet it seems obvious that the absolute temperature of a room also plays an important role. Undoubtedly, the difference in temperature felt on a person's skin has an important impact on that person's comfort but not without reference to the room's

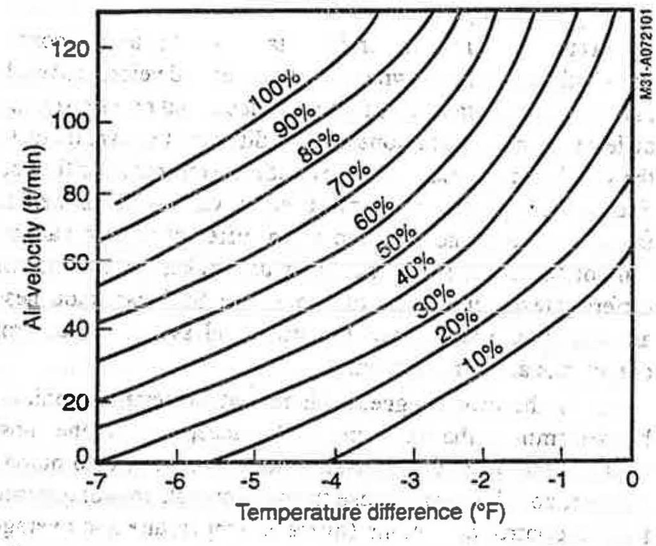

Figure 3 Modified comfort graph of Houghten (based on temperature difference).

absolute temperature. An effective temperature difference, θ, of 2 in a 70°F room is considered to have an identical comfort level as that in an 80°F room, though it is likely that comfort ratings would be different. Control conditions also serve as the 100% acceptable environment for a test, since no change from the control temperature and velocity results, by definition, in no discomfort. When Houghten's comfort data are applied to studies in which control conditions are different from those under which the data were taken, the concept of a specific 100% acceptable control environment fails. In fact, any condition can be 100% acceptable, according to current usage of comfort data, if the room temperature and velocities do not change from the control. Only when identical control conditions are used can comfort data be accurately applied and studies be reliably compared.

The validity of using Houghten's comfort data in terms of a temperature difference is questionable due to the lack of a "control" condition during his tests. When Houghten's data were converted, the initial temperature was used in place of a control temperature to determine the temperature differences. The initial temperature of a test environment is likely to be different from the control temperature of that environment because one is steady state and the other is not. Small changes in control temperature and /or control velocity might significantly change the ADPI values for various tests, and since no testing standards exist, all control conditions are equally valid (or invalid).

In later years, a comprehensive study of the performance of several air distribution systems was carried out by Nevins and Miller (1972). In their tests, Nevins and Miller did not use humans as comfort test subjects and measured local velocity and temperature at more than 200 locations inside the test room. Using the modified comfort graph of Houghten and effective draft temperature given by Equation 2, they established a comfort criterion to represent the conditions at which 80% of the occupants would be comfortable. In their criterion, Houghten's 80% comfort line was used to represent the outer bound to the sense of cooling. Houghten's 100% comfort line (represented by Equation 2 as $\theta = +2$) was extrapolated to represent the boundary between a sense of cooling and sense of warmth, and the maximum allowable draft velocity in the occupied zone was set at 70 fpm. Based on this criterion, they introduced a procedure for evaluating the performance of air diffusion systems. In this procedure, temperatures and velocities are measured at a given number of points uniformly distributed throughout the occupied space. Using these measurements, the effective draft temperature (using Equation 2) is calculated for each point. The number of points that satisfied the comfort criteria ($-3 < \theta < +2$, and $V < 70$ fpm) is expressed in terms of the percentage of the total number of points measured. This number is defined as the Air Diffusion Performance Index (ADPI). This index is a single-number rating of an air diffusion system, which can be used for the selection and design of diffusers and systems. The procedure and criterion developed by Nevins and Miller have been widely accepted by the HVAC industry, and ASHRAE recommends them as a basis for evaluating the performance of air distribution systems in relation to occupant comfort.

Several important concerns that were identified during this background investigation can be listed as follows:

1. The survey sample of original comfort tests conducted by Houghten et al. (1938) was a very good start; however, it was limited and hardly definitive.
2. The concept of "control" conditions was not historically well defined and seems to ignore the impact of absolute temperature on comfort. The conversion from absolute temperatures to temperature differences uprooted the conception of the 100% acceptable condition upon which Houghten's comfort data were based, enabled each researcher to choose his or her own temperature reference point, and reduced human comfort in enclosed spaces to a function of temperature differences rather than absolute temperatures.
3. The accuracy and applicability of the relationship between air velocity and temperature drop required for feeling of equal warmth is not well established. The coefficient 0.07 of Equation 2, introduced by Rydberg and Norback, was based on the experiments of Norback and did not take into account the effect of clothing on comfort. As stated earlier, and as shown in Figure 2, the relationship between air velocity and temperature drop can vary from 0.03 to 0.07 depending on the clothing worn by the subjects.

RECOMMENDATIONS FOR IMPROVING THE "FOUNDATION" OF ADPI TECHNIQUE

Based on the concerns presented above, several recommendations are made toward developing a firmer foundation for the ADPI and for future room-air distribution and human comfort studies:

Re-Evaluation or Incorporation of More Extensive Comfort Data

To reduce the uncertainty and increase the applicability of the ADPI criterion, the comfort database of ADPI should be expanded to incorporate the latest information on human comfort. During recent years, experiments under controlled conditions have been carried out by researchers such as Berglund and Fobelets (1987), Fanger et al. (1988), Rohles et al. (1983), Tanabe and Kimura (1989), and Fanger et al. (1968). (For a comprehensive list and description of studies, refer to Fountain 1991.) In these studies, many test subjects were exposed to different combinations of environmental variables that could influence their perception of thermal comfort. Kirkpatrick and Knappmiller (1996) carried out a computational study to determine the ADPI of cold air jets inside an enclosure. Unfortunately, the results obtained from these tests have not yet been incorporated into the criteria used for evaluation of air diffusion systems such as ADPI.

The expansion of the ADPI database should include research results with both positive and negative temperature differences (with respect to the control temperature) to verify and extend Houghten's findings. The extension should also include test results with a larger number of subjects including men, women, and people from diverse ethnic background. A larger and more diverse sample size than that utilized by Houghten would provide a more comprehensive human response to local velocity and temperature changes.

Based on more recent comfort data gathered from various sources and based on a larger sample of people, a "new effective draft temperature" equation can be determined that would more accurately account for the effects of room-air velocity and temperature on skin temperature and comfort levels. Furthermore, a set of standard test conditions can be developed to equilibrate different studies and to take the place of random "control conditions" currently used in room-air distribution studies.

Use of Standard Testing Conditions to Achieve Comfort

A standard set of room-air distribution control conditions, including operating temperatures, velocities, thermal loads, and test procedures should be established to eliminate inaccuracies, individual interpretations, and arbitrariness in comfort tests. In 1955, H.E. Straub commented that "in the final analysis, the purpose of any study on air distribution is to predict possible comfort reactions." Many studies since the 1930s have investigated the distribution of temperature and velocity in occupied spaces, yet the contribution of these studies to the understanding of human comfort in enclosed spaces has been limited by "universality" problems: differing ideas about drafts, extrapolation of comfort data, and assumption of control conditions. Current tests are not conducted in a way that makes their results easily applicable or comparable to other nonidentical tests, making individual interpretation a necessity. This, in turn, has led to a number of inaccuracies, as described in the previous section.

At the root of the search for a standard set of testing conditions is the question of what temperatures and velocities result in a 100% acceptable condition. As described previously, the different control conditions used in different tests are de facto the 100% acceptable reference condition for that specific test. Each of the control conditions used by various researchers is rather arbitrary and they are often different among various runs of the same study, but their use makes comparison of different tests difficult if not impossible. Such capriciousness and lack of universality will continue unless standard control conditions are agreed upon.

It is, therefore, suggested here that an average condition be determined that is found to be acceptable to the most people. Any tests of room-air distribution related to human comfort could accurately use the comfort data for comparison if their control conditions (average temperature and average velocity) are the same as those defined to be the standard. For example, if standard conditions were defined to be an effective temperature of 70°F, suitable test control conditions could be 70°F and 0 fpm, 72°F and 30 fpm, etc. (using the currently accepted, but questionable, relationship between the velocity and temperature drop of Figure 2, 1°F is equivalent to 15 fpm).

Room-Air Distribution Characteristics

Analysis of the temperature and velocity measurements used in this study revealed that room-air distribution, as it relates to human comfort, may be best characterized by two factors: First, average room conditions (temperature and velocity), and second, distribution of effective temperatures (calculated from temperatures and velocities at different points within the enclosed space). The average temperature and average velocity would give an indication of the room's average energy level. The distribution of effective temperatures would show the comfort rating of each measured point within the room, as well as the number of points considered "comfortable" (similar to ADPI). Therefore, the recommendations for improving the ADPI technique can be listed as follows:

1. Comfort should be comprehensively described by absolute temperatures (in other words, comfort data are applicable only to tests with similar control conditions).
2. Standard test conditions (effective temperature control conditions) should be identified that will allow accurate application of comfort data to future studies.
3. Retesting and incorporation of existing comfort data to ADPI's database are necessary.
4. "Standard" effective temperature control conditions should be defined during retesting of comfort data.

ADPI AND COLD AIR-DISTRIBUTION SYSTEMS

In this study, we did not conduct any experimental investigation of the comfort level of occupants under cold air distribution systems. However, to extend the applicability of the existing ADPI methodology to cold air distribution systems, this study takes advantage of the experimental temperature and velocity data of the research conducted by Nevins and Miller (1972). Their comprehensive research covered the operation of a wide range of air distribution systems under various discharge conditions. The systems tested by Nevins and Miller included high side-wall grilles, circular cone-type ceiling diffusers, sill grilles, ceiling slot diffusers, and light troffer diffusers. Their test results and interpretation of the data as it relates to human comfort were published in *ASHRAE Journal* and *ASHRAE Transactions*. In the most recent publication, Miller (1991) presented a methodology for selecting diffusers for use in cold air systems. His methodology was based mainly on the existing ADPI procedure except that it differed from that used for conventional systems in the recommended procedure to ensure that mixing took place outside the occupied zone of the room. He also provided a new presentation of the existing ADPI data in terms of cfm/ft^2 and load.

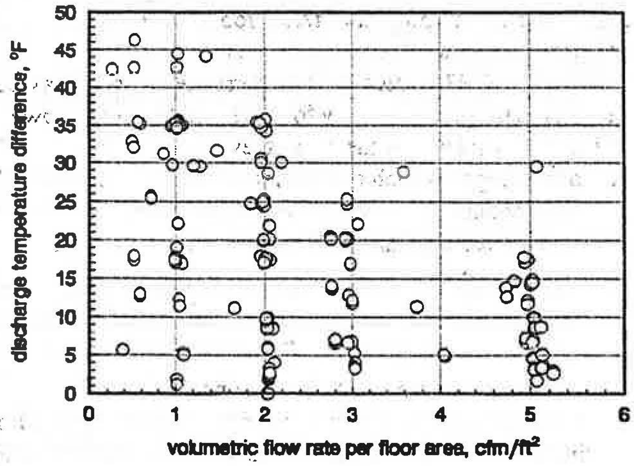

Figure 4. *Range of discharge temperature difference and cfm/ft^2 for experiments conducted for ADPI tests.*

Miller concluded that "the existing diffusers will (with proper attention to their limitations) perform very well with cold air."

The tests conducted by Nevins and Miller covered a wide range of cfm/ft^2 and discharge temperature differences for all the diffusers tested, as shown in Figure 4. This figure shows that many tests were conducted at low flow rates and at high temperature differences (greater than 25°F) in which case those tests can be categorized as cold air distribution cases. Since this set of data contains many points representative of cold air distribution, the comfort criterion developed here will be equally applicable to cold air distribution systems. However, it should be emphasized that during the tests conducted by Nevins and Miller there could have been situations where the diffusers were operated outside their design conditions, which would intentionally create unsatisfactory ADPI results.

In our analysis of room air motion, we realized that the momentum number of the room (J = supply air momentum/volume of the room) played an important role in overall room air motion, and it was also recognized that the existing ADPI and comfort data were never correlated to the momentum number. Hence, in this work, we focused our attention on correlating the existing ADPI and comfort data to the momentum number of the room. First, the relationship between the room mean velocity and the total momentum of the air jet entering the room was investigated because ADPI is based on the room mean velocity and local temperature difference. Following this investigation, an approach for determining the ADPI of cold air systems was explored, the results of which are presented here. In this approach, a one-step procedure was adopted where the ADPI was directly correlated to the momentum number of the room.

Relationship Between Room Mean Velocity and Total Momentum

One of the important factors that influences the perception of thermal comfort is the velocity in the occupied space.

It is, therefore, important to relate room mean velocity to the velocity of the incoming air or to its total momentum. Miller and Nevins (1972) attempted to correlate the room average velocity to the outlet velocity for different diffusers. However, due to large scatter in the data, they were not successful in demonstrating that such correlation existed. Jackman (1973), on the other hand, was able to show a relationship between the room mean velocity and the total momentum of air entering the room. He provided a correlation of the type

$$\bar{v} = aM^b \tag{3}$$

where the constants a and b depend on the diffuser type.

In the present work, the experimental data of Miller and Nevins were used to extend the work of Jackman to a wider range of diffusers and operating conditions. Figure 5 shows the mean room velocity plotted against the inlet momentum for different diffusers tested by Miller and Nevins. This figure also shows the best fit to all the data. In almost all cases studied here, some scatter in the data can be observed. In general, the spread in the data can be attributed to the wide range of discharge temperatures under which the tests were conducted. However, considering the variety of the diffusers tested, the spread shown in Figure 5 may be acceptable for correlating the mean velocity to the total momentum for all diffusers with a single expression. This correlation is shown in Figure 5 and is

$$M = 23.7 + 62.9 \cdot \bar{v} - 6.34\bar{v}^2 \tag{4}$$

where the correlation coefficient is 0.85 for the above fit. Similar correlations for individual diffusers have been obtained but are not presented here. Through correlations similar to that given by Equation 4, the effect of the inlet air on the overall room air motion can be estimated. In addition, ADPI calculations that are based on the average room air velocity and local temperature difference can be related to the momentum of the inlet air.

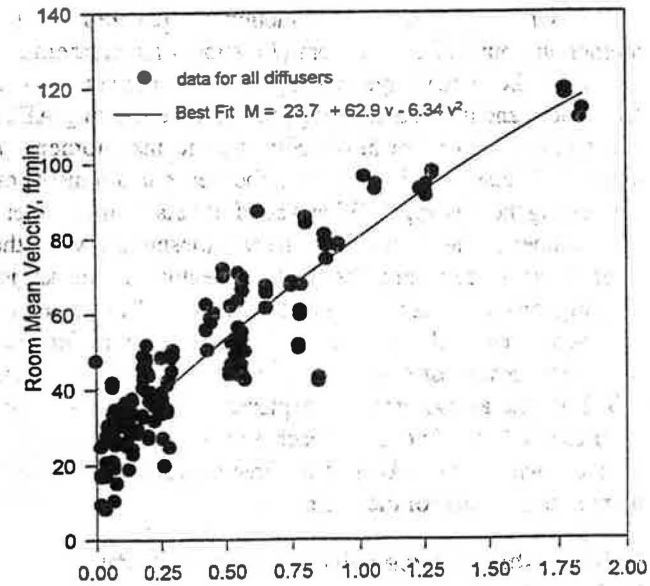

Figure 5 Room mean velocity vs. total momentum for all diffusers.

ADPI vs. Momentum Number

In this approach, for a given diffuser and room load in an enclosed space, the test room's ADPI value was correlated to the momentum number. Each type of diffuser demonstrated a different behavior, as shown in Figures 6 through 11, though the general trend was a bell-shaped curve with maximums between 0.03×10^{-3} and 0.12×10^{-3} lbf/ft^3. Figures 6 through 11 show that ADPI is a strong function of the momentum number, whereas the cooling load has a secondary effect. In general, the ADPI is reduced as the momentum number or the cooling load is increased.

The behavior of ADPI vs. momentum number for linear slot diffusers is presented in Figure 6, which shows that at a momentum number equal to 0.03×10^{-3} lbf/ft^3 all the curves (for different cooling loads) reach a maximum ADPI = 95. This behavior demonstrates that, regardless of the cooling load capacity, an ADPI of greater than 90 will be achievable as long as the momentum number is kept around 0.03×10^{-3} lbf/ft^3. However, this low value of momentum number indicates that the inlet momentum of the air jet is equally low. In the case of cold air systems, special attention should be paid to the behavior of cold air jets with low inlet momentum because it might cause early separation of the cold air jet. To avoid early separation, the throw of the jet and its separation point should be calculated and compared to the procedure outlined by Kirkpatrick and Hassani.

Circular cone type ceiling diffusers show a peak at momentum number 0.06×10^{-3} lbf/ft^3. The peak for higher cooling loads occurs at ADPI of about 85. As shown in Figure 7, the data for this diffuser behave rather nicely with respect to the load and momentum number.

Sill grilles with vanes at 22.5 and 45 degrees showed minimal dependance on cooling load. Figure 8 shows that the

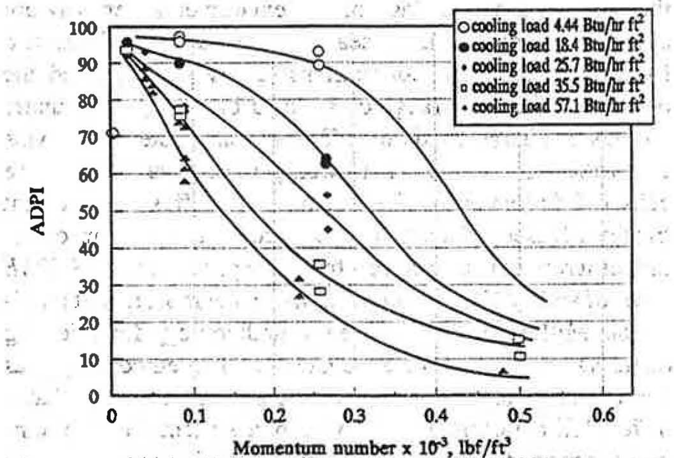

Figure 6 ADPI vs. momentum number for linear slot diffuser.

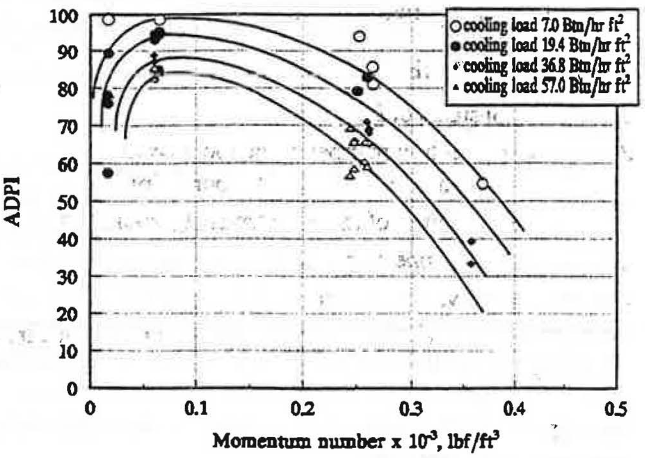

***Figure 7** ADPI vs. momentum number for circular diffusers.*

peak for all cooling loads occurs at 0.06×10^{-3} lbf/ft^3 and all data are very well represented by a single curve. However, the data for sill grilles with straight vanes behave totally differently, as shown in Figure 9. It is interesting to note that, contrary to previous cases, the peak for each cooling load for this type of diffuser shows a strong dependence on the momentum number of the test room. For each cooling load, the maximum ADPI is obtained over the range 0.03×10^{-3} lbf/ft$^3 < J < 0.1 \times 10^{-3}$ lbf/ft^3.

For high side-wall grilles, load dependence is strong for $J < 0.25 \times 10^{-3}$ lbf/ft^3 (see Figure 10). For momentum numbers higher than 0.25×10^{-3} lbf/ft^3, the curves converge into a single curve, indicating no cooling load effect. The maximum ADPI for each load is obtained at momentum numbers less than 0.12×10^{-3} lbf/ft^3 and higher than 0.1×10^{-3} lbf/ft^3.

For light troffers, the load dependance is more obvious. As shown in Figure 11, the maximum ADPI for all cases is obtained at a momentum number of about 0.1×10^{-3} lbf/ft^3. As the momentum number increases, curves representing differ-

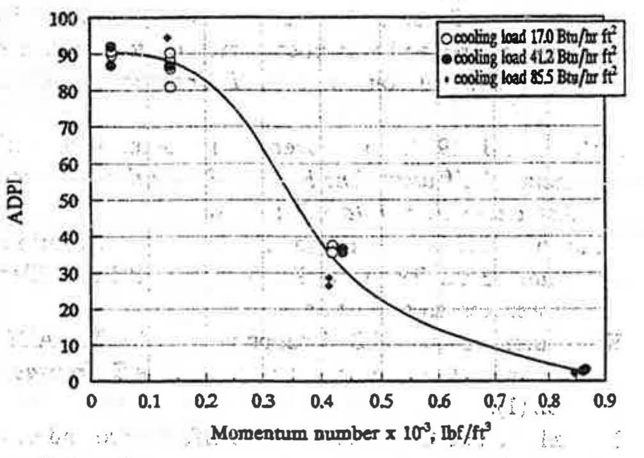

***Figure 8** ADPI vs. momentum number for sill grille, vanes at 22.5 and 45 degrees.*

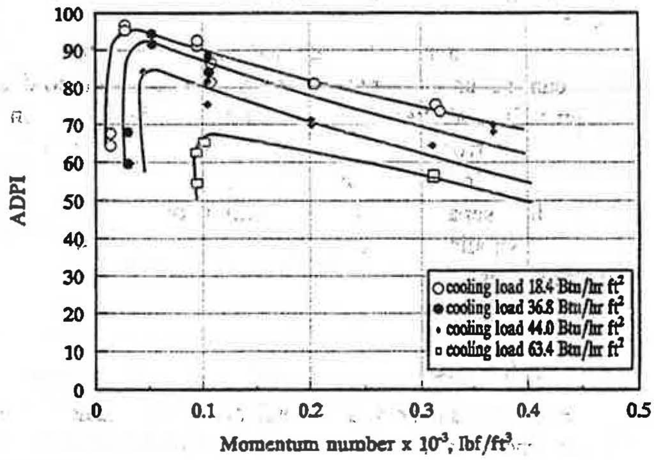

***Figure 9** ADPI vs. momentum number for sill grilles with straight vanes.*

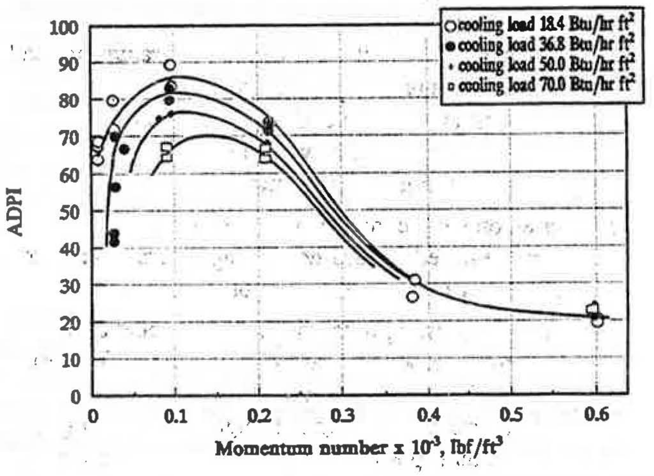

***Figure 10** ADPI vs. momentum number for high side-wall grilles.*

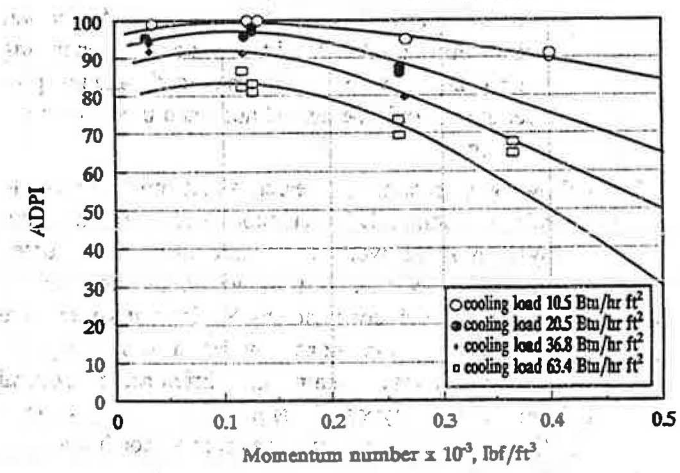

***Figure 11** ADPI vs. momentum number for light troffers.*

ent cooling loads diverge, indicating a strong dependance on cooling load.

Using the approach presented here, the existing ADPI data can be used to predict the performance of cold air systems. Special attention should be paid to the separation of cold air jets from the ceiling. The approach presented here does not employ any separation criterion to identify if a cold air jet has separated from the ceiling or not. The above approach should be used in conjunction with the separation criterion presented elsewhere (see Kirkpatrick and Hassani 1994).

CONCLUSIONS

1. Following a background search of the early studies related to room-air distribution for human comfort, several concerns were identified over the validity of the technical basis of the research. Those concerns can be categorized as:
   a. limited database and survey sample of original comfort tests,
   b. lack of well-defined concept of "control conditions" and the impact of absolute temperatures instead of temperature differences on comfort, and
   c. lack of accurate and well-documented relationship between air velocity and temperature drop required for feelings of equal warmth.
2. Based on these concerns, the following recommendations were presented toward developing a firm foundation for future studies of room-air distribution for human comfort.
   a. The existing database of ADPI should be expanded by incorporating all the reliable comfort test data from various resources. This expansion will help to broaden and verify Houghten's comfort graphs to include positive and negative temperature differences.
   b. Although a set of standard test conditions such as ISO 7730 exists for human comfort, no such standard set exists for relating room air distribution to human thermal comfort. A standard set of room-air distribution control conditions, including operating temperatures, velocities, thermal loads, and test procedures, should be agreed upon and used for future testing.
3. Following the background search, an attempt was made to extend the existing ADPI methodology to cold air distribution systems. In this work, no comfort tests were conducted because of limited resources. However, the experimental data of the research conducted by Nevins and Miller were used. Their test conditions covered a wide range of discharge flow rates and temperature differences for several different diffuser types. A large number of those tests conditions could be categorized as cold air conditions.
4. As a first step to extend the existing ADPI to cold air systems, the local velocities in the occupied space were correlated to the total momentum of the inlet air. These correlations can be used for directly relating the ADPI to the inlet air momentum number.

NOMENCLATURE

a,b	= constants of Equation 3
J	= momentum number of the room (= total inlet air jet momentum/volume of the room), lbf/ft^3
M	= momentum of the jet entering the room, lbf
T	= local temperature, ^{o}F
v	= local velocity, fpm
$\bar{v}$	= room mean velocity based on occupied zone, fpm
θ	= local draft temperature

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