Units Used

The units used in this paper are the same as those used in ASHRAE Standard 52-68[3]. To convert these units to International (S.I.) units the following conversions are provided.

Area

Square feet (ft^2) x 9.2903×10^{-2} = square meters (m^2)

Flow Rate

Feet per minute (ft/min.) x 5.0800×10^{-3} = meters per second (m/sec)

Cubic feet per minute (cfm) x 4.7195×10^{-4} = cubic meters per second (m^3/sec)

Pressure

Inches water gage (in. wg) 70 F x 2.486×10^2 = Newton per square meter (N/m^2)

Dust Load

Grams per square foot (g/ft^2) x 10.764 = grams per square meter (g/m^2)

ACKNOWLEDGMENTS

The author is much indebted to the late Lewis A. Tomes for his invaluable assistance, and particularly for collecting the data represented in Fig. 1.

This paper is an outgrowth of work initially supported by the Post Office Department to evaluate filter media for postal work spaces.

REFERENCES

1. Hans-Joachim Strauss, "Observation of the present situation in the field of air filter testing", Staub *28* (10), 391-395 (1968).
2. Hans-Joachim Strauss, "Suggestion for a procedure for the testing of roll filter media", Staub *29* (10), 404-406 (1969).
3. ASHRAE Standard 52-68, Method of Testing Air Cleaning Devices Used in General Ventilation for Removing Particulate Matter, effective August 1968, p. 36.

No. 2258 RP-55 and 88 (Research Report)

RALPH G. NEVINS

Member ASHRAE

PAUL L. MILLER

Member ASHRAE

ANALYSIS, EVALUATION AND COMPARISON OF ROOM AIR DISTRIBUTION PERFORMANCE - A SUMMARY

The distribution of conditioned air in an occupied space is a complex problem, upon which the success or failure of a given environmental control system will depend. As stated in the ASHRAE GUIDE AND DATA BOOK,[1] the object of air distribution in warm air-heating, ventilating, and air conditioning systems is to create the proper combination of temperature, humidity and air motion in the occupied zone of the conditioned room to satisfy the comfort requirements of the occupants. If it is assumed that sufficient heating or cooling capacity is available to maintain the desired average temperature and humidity within the space, the ability of the system to provide comfortable thermal conditions will then be almost completely dependent upon the distribution of the air. The standards for thermal comfort of the occupant have been established by ASHRAE, through research beginning with the original comfort criteria of Houghten and Yaglou in the mid 20's to the present work of Gagge *et al.*, at the John B. Pierce Foundation Laboratory, Nevins and co-workers at Kansas State University, and Fanger at the Technical University of Denmark.

From a thermal standpoint, it is possible to have an average temperature and humidity which satisfy the criteria for thermal comfort, and these conditions may exist at some point in the space. At the same time, there may be local areas of discomfort caused by excessive variations in air temperature, excessive air motion, failure to distribute the conditioned air according to local load requirements and fluctuations of these conditions with time.

Ralph G. Nevins is Dean, College of Engineering and Director, Institute for Environmental Research, Kansas State University, Manhattan, Kansas. Paul L. Miller is Associate Professor, Department of Mechanical Engineering, Kansas State University, Manhattan, Kansas. *This paper was prepared for presentation at the ASHRAE Fall Conference, Cincinnati, Ohio, October 19-22, 1972.*

ASHRAE has sponsored research in this important area for many years. Nottage,[2] Helander,[3] Koestel and Tuve,[4] Nelson and Stewart,[5] Rydberg and Norback[6] and Straub and Gilman[7] have made major contributions to the understanding of air distribution. The studies of Straub were, for that time, the most thorough and complete investigation of the effect of various air distributing devices on room air motion and temperature distribution. Related work dealing with air distributing ceilings has been published by Boyer.[8]

Beginning with the work of Helander in 1945, Kansas State University has been engaged in air distribution research. Early efforts were directed toward the study of the downward projection of heated air. There was also concurrent development of instrumentation and techniques for the measurement of low velocity air flow. In the early 1960's the test facility used for the recent air distribution studies was developed as part of a research program sponsored by the Whirlpool Corporation.[9] With this experience, interest and facility, a study was initiated at Kansas State University in 1965 to determine the factors which affect air distribution from perforated panels used as an air distributing ceiling and to formulate a means for predicting the performance of these panels. This project was developed and monitored by ASHRAE TC 4.1 (Technical Committee on Air Distribution and Duct Design; now TC 5.3, Room Air Distribution) as ASHRAE RP-55. The Acoustical Materials Association supplied test materials and assisted in the monitoring of the project. With the cooperation and assistance of the Air Diffusion Council, a follow-up project, as ASHRAE RP-88, was initiated through TC 4.1 to study other types of terminal devices.

Test Facilities

The 12 x 20 x 9 ft test room used for air distribution research simulates the interior room of an intermediate floor of a multi-story office building. Details of the test

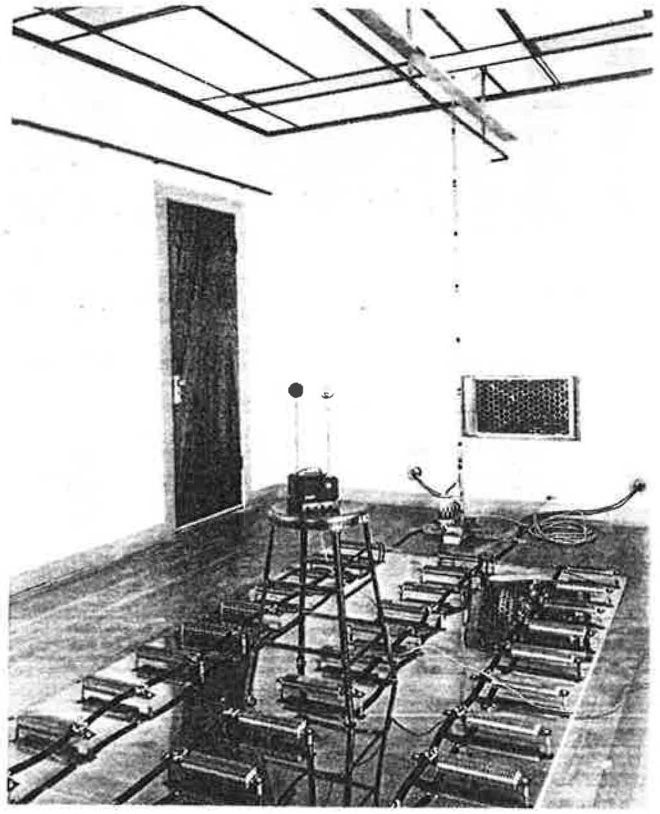

Fig. 1 Interior view of the air distribution test room showing two-sphere radiometer, floor to ceiling temp probe, uniform floor load (finned heaters) and return air grille.

room and the instrumentation are included in refs 10 and 11; however, for completeness, the following summary of the facility is provided. The test room was located within an outer room which provided a 2 ft wide dead air space around the four walls of the room. Sufficient temperatures were monitored around the test room to determine heat load through the walls, through the plenum ceiling and the test room floor. The external heat load was minimized to simulate an interior office and was approximately 15% or less of the total load. All four walls of the test room were insulated as were the two walls of the outer room which were exposed to non-air conditioned space. The supply air plenum, above the ceiling panels, was 16 in. deep. The return air grille (16.5 x 30 in.) was located in the center of the north wall of the test room 19 in. from the floor. The return air traveled through a measuring section, a blower, refrigeration coil, electric heater (for fine adjustment of inlet temp), a second flow measuring section and then was supplied to the plenum. Fig. 1 is a photograph of the test room showing the two-sphere radiometer used to measure mean radiant temperature, the floor to ceiling temperature probe, the uniform floor load and the return air grille. Fig. 2 shows the remotely controlled instrument rack consisting of 12 heated thermocouple anemometers, part of the uniform floor load and the concentrated load in its normal test position. Fig. 1 is taken looking north in the test room; Fig. 2 is looking south.

The interior load consisted of lighting and additional electrical load representing people and equipment. The electrical load consisted of a "uniform" load of 42 finstrip heaters (rated at 750 watts ea.) mounted on a 5 x 12 ft aluminum sheet. The uniform load was normally operated at fin surface temp of less than a 100 F (6-8 Btuh/sq ft) to prevent unusual thermal currents. The "concentrated" load consisted of four 2850 watt finstrip heaters mounted in a 36 x 38 x 12 in. cabinet. The heaters were operated considerably below rated power to diminish convective velocities from the unit.

The remotely controlled test rack shown in Fig. 2 determined temperatures and velocities at 12 points in each position. A total of 24 points on a 2 ft grid could be obtained with the rack in each horizontal position. Nine vertical planes on 2 ft centers were scanned throughout the test room to obtain a total of 216 points per test run. The location of the test points is illustrated in Fig. 3.

In addition to temperatures and velocities at the 216 points (240 points when the concentrated load was not used), floor to ceiling temp gradients were obtained with a vertical probe having thermocouples located 1, 3, 5, 7, 8, and 9 ft above the floor. Readings were taken on the N-S centerline of the room, 3 ft from the south end and 2 ft from the north end. Qualitative measurements of air flow using a smoke gun were recorded for each typical test arrangement to obtain the general patterns of air movement throughout the space.

Air Distribution Performance Index

To manage the tremendous amount of data obtained from the test program and to provide a single number rating system for the evaluation of any air distribution system, an Air Distribution Performance Index (ADPI) was proposed by Miller, Hanni and Nevins.[9] Since the ultimate determination of performance is the comfort of the occupant, ADPI is based on subjective responses to draft (temp difference and air velocity) obtained by Houghten *et al.*, in 1938[12] and the work of Rydberg and Norback.[6] The latter work

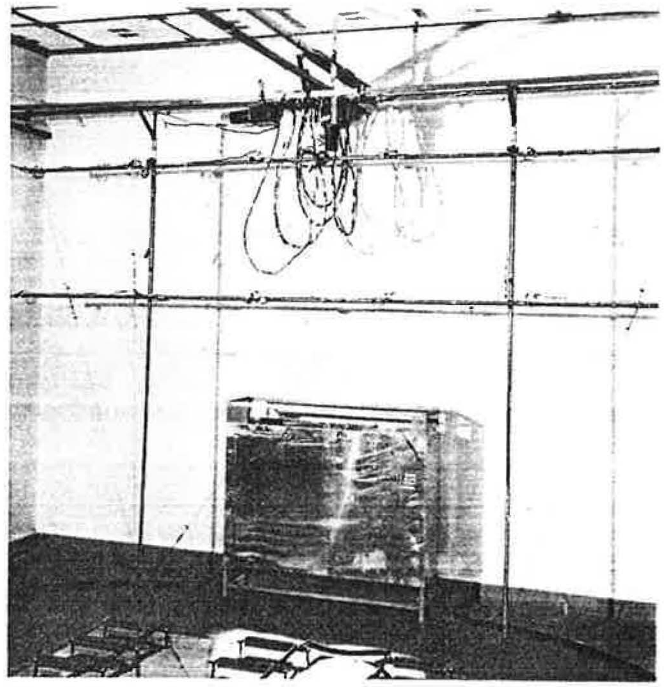

Fig. 2 Remotely controlled instrumentation rack (12 heated thermocouple anemometers) shown in the "up" position. Part of the uniform floor load is shown in the foreground. The concentrated load is shown in its normal position against the south wall.

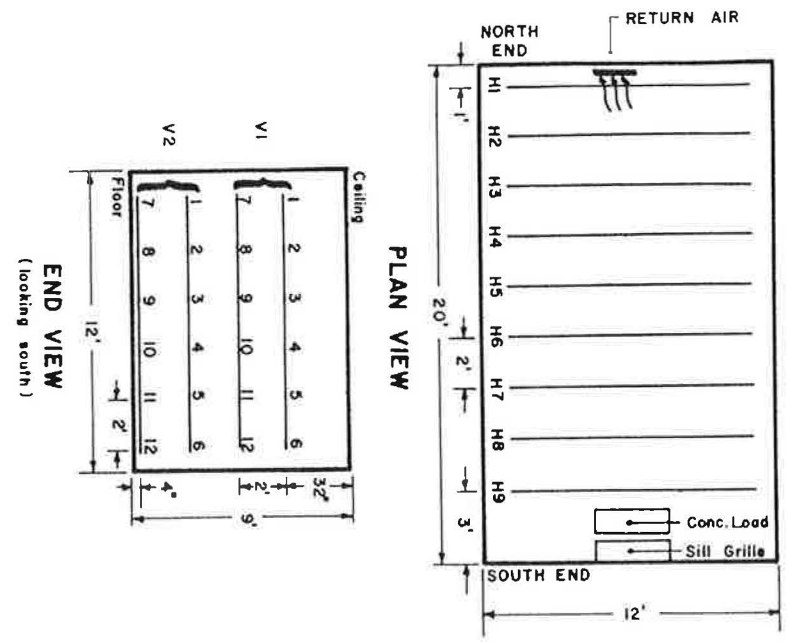

Fig. 3 Test Room and Instrumentation positions for measurement of temp and velocities at 216 points throughout the test space.

TABLE I
Outline of Test Conditions

Ceiling configuration	100% active panels (holes) 50% active panels (holes) 100% active panels (slots) 50% active panels (slots)
Air flow (cfm/ft^2 of ceiling)	0.5, 1, 2, 5, 10
Total Temp difference (F)*	Min.**, 3.5, 7, 8.8, 15, 17.5, 25, 35
Load Type	Uniform plus concentrated

**Minimum of 4 values per configuration.*
***No load. Some temp difference was necessary to overcome uncontrolled heat gains, therefore, the Δt was not zero.*

was modified by Straub in a discussion of a paper by Koestel and Tuve.[4] The criteria define an effective draft temp (θ) in terms of the local velocity (V_x) in fpm, and the difference in temp, in degrees F, between the local point (T_x) and the control temp (T_c) as follows:

$$\theta = (T_x - T_c) - 0.07(V_x - 30) \quad (1)$$

For the test room, the control temperature is defined as the average of the 16 temperatures forming a 2 ft x 2 ft x 6 ft rectangular solid about a line 36 in. off the floor in the center of the test room. The 6 ft dimension is parallel to the 12 ft room dimension. Fig. 4, with values of effective draft temp shown as solid lines, shows that good agreement exists between $\theta = -3.0$ F and Houghten's data for 80% of the occupants reporting comfort. Koestel and Tuve indicated that an upper limit of $\theta = +2.0$ F is probably satisfactory. Considering velocities above 70 fpm objectionable, the area shown in Fig. 4 encloses the conditions which are considered comfortable. The number of positions in the test room (out of the 216 measuring points) which have an effective draft temp between −3.0 and +2.0 and a velocity less than 70 fpm, expressed as a percent of the total, is defined as the Air Distribution Performance Index (ADPI). It is the intent of this summary to show that ADPI can be generalized and for practical use may be simplified to require a smaller number of test positions. In general ADPI can be defined as "the number of measuring positions, uniformly distributed in a test plane or space at which the comfort criteria are satisfied, expressed as a percent of the total number of positions at which measurements of temperature and velocity are made."

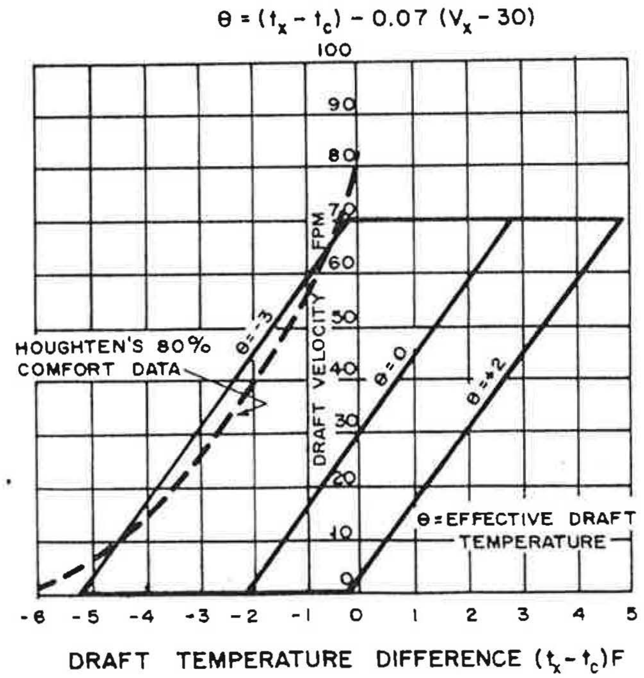

Fig. 4 Comfort criteria used to evaluate ADPI. Points within the area bounded by $-3.0 < \theta < 2.0$ and $V < 70$ fpm will be comfortable for 80% or more of the occupants in a given space.

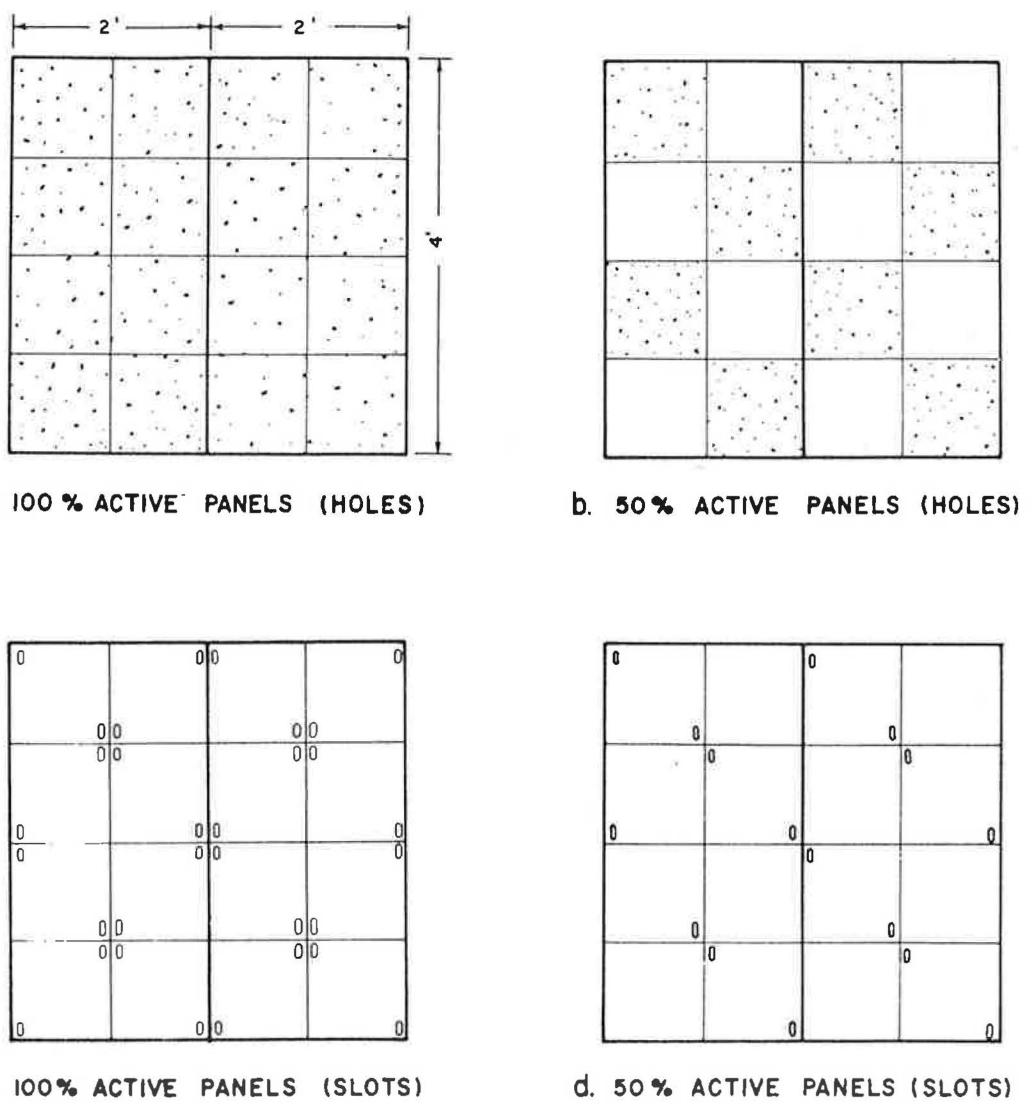

Fig. 5 Ceiling panel perforation geometry for air distributing ceiling tests.

Test Conditions and Experimental Program

The study of room air distribution performance with air distributing ceilings was initiated as Project RP-55. To have a basis for comparison with the results obtained with the ceiling tile, RP-55 included tests using circular ceiling diffusers. RP-88, the follow-up project, included other terminal devices of typical manufacture. No attempt was made during any of the tests to obtain the "best" ADPI. All devices tested were subjected to conditions which were known to be "unsatisfactory" as well as to conditions which were known to be "satisfactory." Table 1 indicates the test conditions used with the air distributing ceilings. For these cases, the total temp difference from plenum stub-duct to return grille was used to determine the total load on the system. Fig. 5 indicates the ceiling panel geometry used in the tests. Table 2 outlines the test conditions for the terminal devices including the diffusers used in RP-55 and the high side wall grille, the sill grille, the ceiling slots and the light troffer diffusers used in RP-88. Fig. 6 shows schematically the location of the various devices in the test room.

The air distributing ceiling data were reported graphically and in equation form by Nevins and Ward[10] and Miller and Nevins.[11] ADPI was correlated with total load. Total load was defined to include the energy picked up by the supply air in both the plenum and the room. In systems

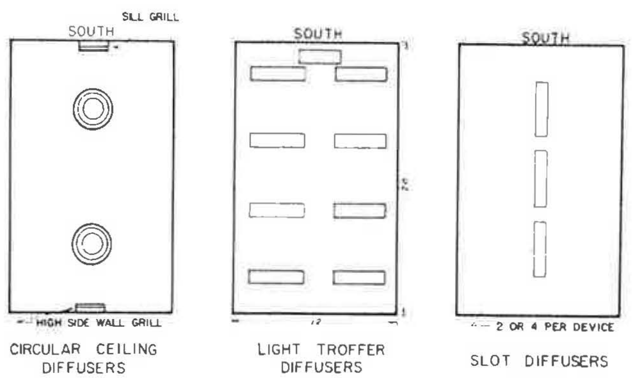

Fig. 6 Location of the terminal devices in the test room: (a) round circular diffusers, (b) high side-wall grille, (c) sill grille, (d) ceiling slot diffusers, (e) light troffer diffusers.

using terminal devices, such as circular ceiling diffusers, the supply air picks up energy (load) only in the room. To compare systems, therefore, the air distributing ceiling data were analyzed using both room load and total load as parameters. The room load was determined from the temp measured in the ceiling perforations and the return air temp measured at the return grille. For the terminal devices the supply air temp was measured at the diffuser and the return air temp to determine the room load. Initially, the uniform load was used to generate the entire load on the system; however, at loads of over 45 Btuh/sq ft, the room air exhibited considerable instability, probably due to the high fin surface temperatures. To correct this situation, a uniform plus concentrated (U + C) load configuration was designed. The uniform load was then limited so that the maximum surface temp of the heaters was 100 F. The concentrated heat load, located at the south end of the room, provided additional load as required. A detailed study of the instability showed that local temp swings of 4.0 to 13.0 F (50 min period) were reduced to 1.0 to 2.3 F when the load was changed from "U" to "U + C."

SUMMARY OF RESULTS

The ADPI for an air distributing ceiling was found to be a function of load and was not independently affected by the type of ceiling, total air temp differential, or air volume flow rate, except at high flow rates when plenum pressures were high. The functional relationship for ADPI determined using a least squares technique was:

$$ADPI = 99.8 - 0.278(Q_t) \qquad (2)$$

where Q_t = Total load, Btuh/sq ft.

The 95% confidence interval for Eq 2 is ± 5% and represents all the data from the tests outlined in Table 1 except 100% holes at 10 cfm and 100% slots at 5 cfm. Correlating equations for these two test conditions are

a. for 100% active panels (holes) at 10 cfm/sq ft.

$$ADPI = 92.8 - 0.317(Q_t) \qquad (3)$$

b. for 100% active panels (slots) at 5 cfm/sq ft.

$$ADPI = 87.5 - 0.378(Q_t) \qquad (4)$$

The ceiling panels tested were representative of existing commercial types. Two types of hole patterns were prepared and supplied by the Acoustical Materials Association:

1. Perforated panels using round holes 0.1 in dia (about 300 holes per sq ft) and
2. Perforated panels using slots, each 3/16 x 1-5/8 in. (two holes per sq ft of ceiling panel).

The base material for both types was plain, painted-finish, mineral-fiber panel. The necessity for the separate correlations represented by Eqs 3 and 4 is attributed to the high plenum pressures for these two flow conditions, about 0.14 in. of water for 10 cfm/sq ft and 0.19 in. of water for 5 cfm/sq ft. The plenum pressures for other tests did not exceed 0.11 in. of water.

To compare the ventilated ceiling data with the circular cone-type ceiling diffusers the correlating equation was modified so that ADPI was a function of room load rather than total load. The correlating equation was then

$$ADPI = 98.8 - 0.517(Q_r) \qquad (5)$$

where Q_r = room load, Btuh/sq ft.

The two sizes of circular ceiling diffusers, 12 in. round and 6 in. round, were tested with three different cone positions to provide a vertical pattern, an intermediate pattern and a horizontal pattern. In contrast to the air distributing ceilings, the circular ceiling diffusers were found to produce ADPI's which were functions of room load, air flow rate and the diffuser discharge pattern.

Good air distribution (high ADPI values) was obtained with circular ceiling diffusers when proper attention was given to diffuser throw characteristics in relation to the room size.[13] For lower room loads (20 Btuh/sq ft) and inlet air volumes (2 cfm/sq ft), the vertical temp variations in the room were generally less than 1.0 F for both systems tested; i.e., air distributing ceilings and circular ceiling diffusers. However, at higher room loads (80 Btuh/sq ft) and inlet air volumes (5 cfm/sq ft), the vertical temp variations

TABLE II
Outline of Test Conditions

DEVICES TESTED	High Sidewall Grille 38 tests, Control Temp:	74.0 ± 0.7 F	24 x 6 in. 16 x 6 in.	vanes straight vanes straight
	Sill Grille 39 tests, Control Temp:	73.9 ± 0.7 F	24 x 4 48 x 6 24 x 3 48 x 6 24 x 6	vanes straight vanes straight vanes straight vanes 45 and 22.5 deg vanes 45 and 22.5 deg
	Ceiling Slots 38 tests, Control Temp:	75.3 ± 1.4 F	2 or 4	active per diffuser
	Light Troffer Diffusers 30 tests, Control Temp:	74.3 ± 0.7 F	4 or 8	active per diffuser
	Circular Cone-Type Ceiling Diffusers 111 tests, Control Temp:	76.0 ± 1.2 F	12 in. pair 6 in. pair 6 in. pair	horizontal deflection horizontal deflection intermediate deflection

AIR FLOW (cfm/sq ft of floor) 0.5, 1, 2, 3, 4, 5
ROOM LOAD (Btuh/sq ft of floor) . . . . Min. 17.5, 35.0, 50, 70
LOAD TYPE . . . . Uniform plus Concentrated (U + C)

within the room were slightly larger with air distributing ceilings than with circular ceiling diffusers (2.0 vs. 1.5 F). The lower temp variation for the diffusers apparently was obtained at the expense of higher room velocities which resulted in a lower ADPI. For flow rates of 2 cfm/sq ft or less, no clear cut choice between these systems (ventilating ceilings and cone type circular ceiling diffusers) existed. At flow rates of 5 and 10 cfm/sq ft the ventilated ceilings produced higher ADPI values than did the 12 in. diffuser system. Under these conditions this diffuser system would be categorized as poorly designed since the isothermal throw values at these air flow rates far exceeded the room dimensions.

The performance of the other four terminal devices was found to be a function of room load, air flow rate and type of diffuser.[14] Good air distribution may be obtained with any of the diffuser systems provided proper attention is given to the diffuser throw characteristics and room size.[15] Vertical temp variations for the lower loads and air flow rates were generally less than 1.0 F as with the other systems and devices. With loads of 70 Btuh/sq ft and air flow rates of 5 cfm/sq ft, the vertical temp variations were larger (2.6 F). However, these temperature variations could be minimized by using a diffuser having an isothermal throw which more nearly approaches the characteristic dimension of the room. The spatial relationships between the concentrated load and the supply air outlet and between the return air grille and the concentrated load were important in some configurations. The highest ADPI was obtained with the outlet above the concentrated load and the load directly in front of the return grille.

Optimum Design Performance

As stated above, through careful study of the data produced by this project, ADPI has been shown to be a function of the type of terminal device, the room load, the flow rate of the supply air and the room geometry for all systems except air distributing ceilings. Miller and Nash[15] combined all of these parameters and presented ADPI as a function of the ratio of the isothermal (T_v) jet throw distance to a characteristic room dimension with room load as a parameter. This analysis simplifies considerably the specification of air distribution performance. The isothermal throw of the jet is defined as the distance from the outlet to a point in the isothermal air stream where the maximum velocity occurring in the stream cross section has been reduced to a selected "terminal" velocity.* (The subscript on T indicates the selected terminal velocity). The characteristic room length (L) is in some installations an easy dimension to determine and in some installations more difficult. For example, for a high side wall grille located in the one end of the test room, the characteristic room length would be the length of the room in the direction of the throw. A characteristic dimension for the slot diffusers would be half of the width of the room since the jet pattern was perpendicular to the long dimension of the test room. In some installations the air pattern will be directed toward an adjacent device such that the primary jets will interact. This configuration occurs in large spaces served by multiple outlets. In the research reported above, this situation existed for those tests involving the light troffer diffusers. In this case, L was taken as one-half the distance between units plus 2 ft, the distance from the ceiling to the top of the occupied zone. Curves of ADPI versus T_{50}/L for circular ceiling diffuser are shown in Fig. 7. For most devices tested a characteristic maximum point occurs at T_{50}/L of 1.0 to 2.0 and varies somewhat with load.

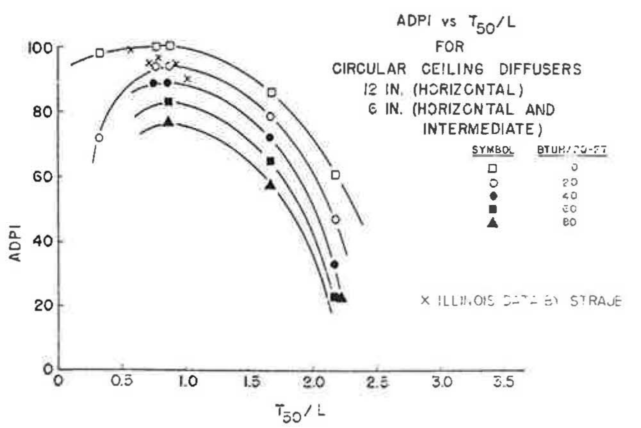

Fig. 7 ADPI vs. T_{50}/L for circular cone-type ceiling diffusers with University of Illinois and ADC data points added for comparison.

TABLE III

Comparison of University of Illinois Air Distribution Analysis with ADPI (Circular Ceiling Diffusers)

Diffuser Size (in.)	Flow Rate (cfm)	Temp Index (%)	Velocity Index (%)	ADPI (%)
10	222	98	87.0	99
8	222	90	79.0	95
6	220	91	83.0	95
10	304	95	74.3	97
10	406	96	53.0	90

Test Room 14 x 18 x 8.5 ft. Loading: 9.7 to 19.8 Btuh/sq ft.

To increase the usefulness and the practicality of ADPI, a study was made using the "center-line" ADPI, i.e., a reduction in the number of measuring points and thereby a reduction in cost and effort required to determine ADPI.[16] For all devices tested, center-line ADPI was determined to be a valid, practical single number rating for the performance of the device. In general, the center-line ADPI versus T_{50}/L curves have the same shape and the maximum points occur at essentially the same values of T_{50}/L as the curves for the whole-room ADPI for corresponding devices.

Attempts to correlate ADPI with average room velocity were unsuccessful, thereby further supporting the conclusion that ADPI and occupant comfort are not functions of velocity or average velocity alone. Room average velocity was determined to be a linear function of diffuser outlet velocity.[16] This correlation is not universally usable since the slope of the correlation curve is a function of room size and shape.

**Throw data certified under the Air Diffusion Council's Equipment Test Code 1062R3 must be taken under isothermal conditions. Throw data not certified by ADC may be isothermal or not, as the manufacturer chooses.*

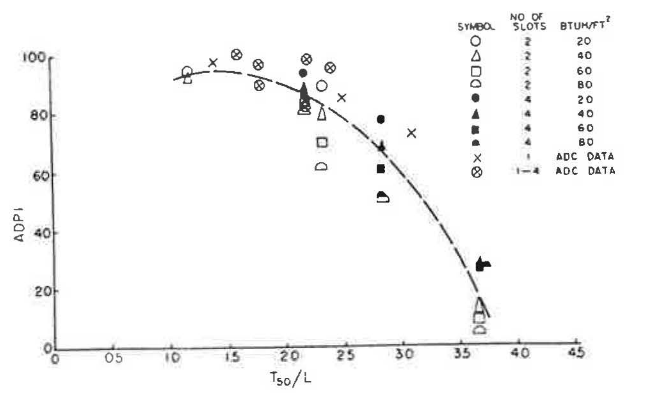

Fig. 8 ADPI vs. T_{50}/L for slot-type diffusers with ADC data points added for comparison.

System Evaluation-Application of ADPI

The ultimate usefulness of ADPI depends on the design engineer's understanding of its value and application. To illustrate the value of a "single number" rating, Straub,[7] in 1957, had to define six inches to rate room air distribution performance. 1. A temperature index – the percentage of points in the occupied zone within plus or minus 1.0 F of the control temp; 2. temp variation index – the difference in the average temp between the 4.0 and the 60.0 in. level; 3. temp variation between the 4.0 and the 90 in. level; 4. velocities less than 15 fpm – percentage of the traverse points at which the air velocity was less than 15 fpm; 5. velocities less than 35 fpm – the percentage of traverse points at which the air motion was less than 35 fpm; and 6. velocities between 15 and 35 fpm – the percentage of traverse points at which the air motion was between 15 and 35 fpm.

In Straub's discussion to ref 13, data from his Illinois studies were analyzed using ADPI. Table 3 compares the Temperature Index and Velocity Index with ADPI for 5 tests. The five ADPI values are shown in Fig. 7 for comparison. To calculate T_{50}/L, the characteristic room dimension was chosen as 8 ft. The test room dimensions were 14 x 18 x 8.5 ft. It would appear from Table 3 that the Velocity Index may be a more sensitive measure of system performance. However, this is not a general case, since this index was determined by using a narrow range of velocities, 15 to 35 fpm, whereas ADPI is based on human evaluation of draft conditions which would reject combinations of temp difference and velocity excluded by the Velocity Index, e.g., 35 fpm and 3.0 F ΔT.

With the permission of selected members of the Air Diffusion Council, ADC certified data relating "Throw", "Terminal Velocity" and "Room Velocity" were analyzed, using center-line ADPI. The ADC Test Code 1062R3, Section A.13, specifies the conditions and requirements for this test. For symmetrical devices, the test room is 12 x 12 x 9 ft, for asymmetrical devices test room is 12 x 24 x 9 ft.

Fig. 8 shows ADPI vs. T_{50}/L from ref 15 with the addition of data points for an ADC slot diffuser with one slot and an ADC slot diffuser with 1 to 4 slots of two different widths. The ADC data were taken in a single plane and are for loads varying from 25 to 85 Btuh/sq ft. The agreement is excellent from a practical standpoint, considering that Fig. 8 represents data from three different laboratories.

TABLE IV

ADPI values for Two Experimental Conditions From HVRA Report No. 65 (Sidewall Grille)

Air Flow Rate (cfm)	Avg. Room Temp (F)	HVRA V_R(fpm)	ADPI (%)	Load (Btuh/sq ft)
593	69.0	42.7	85.7	23.4
296	70.5	21.6	81.5	21.6

Outlet Size: 12 x 12 in., Room Size: 16 x 12 x 9 ft.

The research program at KSU did not include the testing of square perforated or square louvered diffusers. Again using ADC certified data, center-line ADPI values were determined for three devices from different manufacturers. These data are shown in Fig. 9. The characteristic curve of ADPI versus T_{50}/L is evident. The room load varied from 15 to 51 Btuh/sq ft, which accounts for some of the scatter. It is interesting to note that the 12 x 12 perforated pattern diffuser (1, 2, 3, and 4 way blow) shows identical results with the square louvered device.

The Heating and Ventilating Research Association (England) has developed a design procedure for sizing sidewall grilles.[17] Air movement within the occupied space is correlated with the inflow momentum (the momentum of the supply air). Using the two sets of data contained in ref 17, ADPI values were determined and are compared with the estimated room velocity in Table 4. Assuming V_R = 20 to 50 fpm as satisfactory, both situations are good designs. ADPI values are probably acceptable for most designs but indicate that some improvement could be made. Momentum can be related to throw using the equations:

$$\frac{X}{\sqrt{A_e}} = K\frac{V_o}{V_x} \tag{6}$$

where:

X = distance from outlet to a point on the center-line of the jet stream, ft.

$\sqrt{A_e}$ = effective area of the outlet, sq ft.

K = constant of proportionality.

V_o = outlet velocity, fpm.

V_x = center-line velocity of the jet at the distance X from the outlet, ft.

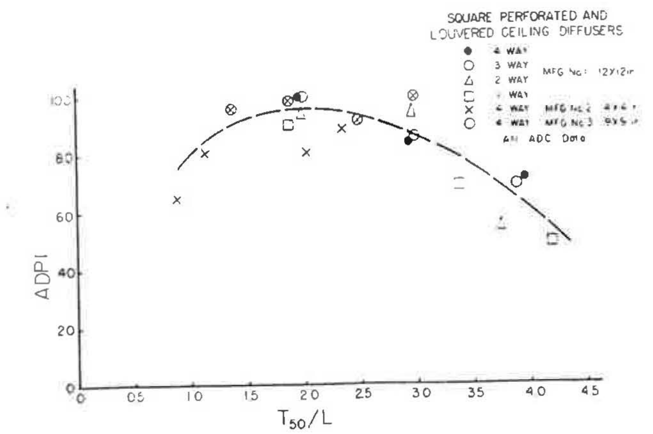

Fig. 9 ADPI vs. T_{50}/L for square face, perforated and louvered ceiling diffusers. All data from ADC laboratories. Room loads from 11 to 51 Btuh/sq ft.

Momentum is defined as:

$$M = \rho Q V_o \tag{7}$$

where:

M = Momentum, ft-lbs/min.
ρ = density of the supply air, lbs/cu ft.
Q = Supply air flow rate, cfm.

Combining these equations gives

$$M = \rho (V_x X/K)^2 \tag{8}$$

ADPI has been shown to be a function of load. Load is introduced into M (for constant Q) by the change in density. This effect is small, even for fairly large changes in supply air temp, therefore M is relatively insensitive to load.

The contributions of HVRA and Jackman[17] are significant and hopefully more data will be made available to the profession. Their design procedure recommends, as a result of their research, a change in the commonly used throw criterion of 50 fpm (0.25 m/s) at 0.75L to 100 fpm (0.5 m/s) at 0.75L. This is equivalent to the results obtained at KSU wherein the maximum ADPI occurs at T_{50}/L values greater than the common recommendation for outlet selection of $T_{50}/L = 1.0$.

CONCLUSIONS

1. The extensive studies sponsored by ASHRAE, AMA, and the Air Diffusion Council have resulted in considerable and significant contributions to our knowledge of room air distribution. A sensitive, valid single-number rating for air distribution performance has been defined and tested and has been compared with data from other laboratories. The procedure for determining the rating index has been simplified, with appropriate accuracy, so that it can be used as a manufacturer's and field test procedure.

2. As a result of these studies, the Air Diffusion Council has included in the latest revision of their test code a procedure for determining the center-line ADPI for terminal devices manufactured by members of the Council.

3. It has been shown that ADPI could be used as a "quality" rating similar to NC numbers or as a specification. Both are based on subjective evaluation of the environment (thermal and noise). NC numbers are used widely to specify or rate the sound level in a given space. ADPI can be used to specify or rate the air distribution in a given space. Environmental quality, rating or specifications may soon be written in terms of (a) the ASHRAE Comfort Standard 55-66, for the overall thermal environment, (b) NC numbers for the noise environment and (c) the ADPI for the "draft" or local environment.

4. For future work, two studies are recommended to be undertaken: one is an updating of the draft criteria of Houghten, i.e., the subjective reactions to air movement and temperature difference and the other a special study of ADPI for non-steady systems (systems which have on-off fan operation), and for variable volume systems.

5. As a next step in this continuing project to provide better and more useful data for application, design and installation of environmental control systems an evaluation of field installations using ADPI criteria is recommended.

ACKNOWLEDGEMENTS

The authors wish to acknowledge with thanks the support and encouragement of their colleagues, particularly Dr. E. Dawson Ward and Mr. Rodney T. Nash who made significant contributions to various phases of the work, and the more than twelve graduate and undergraduate students who participated in the project; the assistance of Mr. Sung-Nan Hsu is also appreciated for his preparation of drawings and data analysis. The financial assistance of ASHRAE, the Acoustical Materials Association, the Air Diffusion Council, and the encouragement and assistance of the members of TC 4.1 are also acknowledged with thanks.

REFERENCES

1. ASHRAE GUIDE AND DATA BOOK, Equipment, Chapter 1, Space Air Distribution, ASHRAE, N.Y. 1969, p. 1.
2. H. B. Nottage, Ventilation Jets in Room Air Distribution, Ph.D. Thesis, Case Institute of Technology, (3 vols. Loan copies available from ASHRAE), 1949.
3. Linn Helander and C. V. Jakowatz, Downward Projection of Heated Air, ASHVE Trans., Vol. 54, 1948, p. 71.
4. Alfred Koestel and G. L. Tuve, Performance and Evaluation of Room Air Distribution Systems, ASHVE Trans. Vol. 61, 1955, p. 533.
5. D. W. Nelson and D. J. Stewart, Air Distribution from Side Wall Outlets, ASHVE Trans., Vol. 44, 1938, p. 77.
6. John Rydberg and Per Norback, Air Distribution and Draft, ASHVE Trans., Vol. 55, 1949, p. 225.
7. H. E. Straub, S. F. Gilman and S. Konzo, Distribution of Air within a Room for Year Round Air Conditioning – Part I, University of Illinois Engineering Experiment Station Bulletin No. 442, March, 1957.
8. L. L. Boyer, Jr., Thermal Environment in Perimeter Office Spaces Cooled with Ventilating Ceilings, ASHRAE Trans., Vol. 73, Part 1, 1967, p. III.3.1.
9. P. L. Miller, Peter Hanni and R. G. Nevins, Final Report – Room Air Distribution Performance of Window Type Air Conditioners, KSU Engineering Experiment Station Project 296, Feb. 1964.
10. R. G. Nevins and E. D. Ward, Room Air Distribution with an Air Distributing Ceiling, ASHRAE TRANSACTIONS, Vol. 74, Part 1, 1968, p. VI.2.1.
11. P. L. Miller and R. G. Nevins, Room Air Distribution with an Air Distributing Ceiling – Part II, ASHRAE TRANSACTIONS, Vol. 75, Part I, 1969, p. 118.
12. F. C. Houghten, C. Gutberlet and E. Witkowski, Draft Temperatures and Velocities in Relation to Skin Temperatures and Feeling of Warmth, ASHVE TRANSACTIONS, Vol. 44, 1938, p. 289.
13. P. L. Miller and R. G. Nevins, Room Air Distribution Performance of Ventilating Ceilings and Core-Type Circular Ceiling Diffusers, ASHRAE TRANSACTIONS, Vol. 76, Part I, 1970, p. 186.
14. P. L. Miller, Room Air Distribution Performance of Four Selected Outlets, ASHRAE TRANSACTIONS, Vol. 77, Part II, 1971.
15. P. L. Miller and R. T. Nash, A Further Analysis of Room Air Distribution Performance, ASHRAE TRANSACTIONS, Vol. 77, Part II, 1971.
16. P. L. Miller and R. G. Nevins, An Analysis of the Performance of Room Air Distribution Systems, ASHRAE Annual Meeting, Nassau, Bahamas, June, 1972.
17. P. J. Jackman, Air Movement in Rooms with Side-Wall Mounted Grilles – A Design Procedure, Laboratory Report No. 65, The Heating and Ventilating Research Association, Bracknell, England, 1970.

Officers, Directors, Committeemen, Staff

OFFICERS

President
Walter F. Spiegel

President-Elect
Roderick R. Kirkwood

Vice Presidents
William J. Collins, Jr.
David Rickelton

Treasurer
William P. Chapman

Executive Secretary
Andrew T. Boggs III

Executive Secretaries Emeritus
M. Curry Turpin
Robert C. Cross

BOARD OF DIRECTORS

June 1970-June 1973
Perc M. Butler
Joseph D. Loveley
Bruno P. Morabito *(I)*
Jacob Klassen *(II)*
Robert J. Evans *(III)*

August 1971-June 1974
Jack K. James
Scott A. Spencer
William P. Wells *(IV)*
Charles F. Sepsy *(V)*
Joseph B. Olivieri *(VI)*

June 1972-June 1975
Paul R. Achenbach
Morris Backer
Robert L. Kilker *(VII)*
Hugh D. McMillan, Jr. *(VIII)*
Robert D. Pagliasotti *(IX)*
C. Stuart Perkins *(X)*

STAFF

Executive Director and Director of Publications
Andrew T. Boggs III
Assistant to Executive Director
Doris Flandorffer
Director of Research
Royal S. Buchanan
Manager – Technical Services
Nicholas A. La Courte
Controller
Walter A. Glasser
Meetings Secretary
Julia I. Szabo
Membership Secretary
Stephen C. Singleton
Editor – Handbook
Carl W. MacPhee, P. E.
Editor – Journal and Director of Public Relations
James H. Cansdale
Editor – Special Publications
David Houston
Circulation Manager
George Gray

COMMITTEES

(Term of appointment or election 1972-73, unless otherwise indicated. ExO, BOD means ex officio, Board of Directors; CO means Coordinating Officer.)

ADMISSIONS & ADVANCEMENT
Loran V. Appleby, *Chairman*
Felix J. Veliath, *Vice Chairman*
John E. Copeland
Robert W. Cumming
Charles R. Hiers
Bruce P. Hyde
Louis Lenz
William T. Smith
Scott A. Spencer *(ExO, BOD)*
Roderick R. Kirkwood *(CO)*

CHAPTERS REGIONAL
Regional Chairmen
Member and Alternate
from each Chapter

CHARTER & BYLAWS
Gayle B. Priester, *Chairman*
Jules A. Endweiss, Jr.
George A. Freeman
Scott A. Spencer *(ExO, BOD)*
David Rickelton *(CO)*

EDUCATION
Gordon B. Weld, *Chairman*
Staron E. Ammons, *Vice Chairman*
James H. Carpenter *(I)*
Lorne S. Wilson *(II)*
George W. Campbell *(III)*
William A. Smith *(IV)*
Kenneth N. Hawk *(V)*
Floyd H. Schneeberg *(VI)*
William J. Coad *(VII)*
Frank J. Biggi *(VIII)*
Donald G. Bergoust *(IX)*
Gerald A. Howell *(X)*
Charles J. McKeone *(ARI)*
Paul R. Achenbach *(ExO, BOD)*
Roderick R. Kirkwood *(CO)*

ENVIRONMENTAL STANDARDS FOR BUILDINGS
Frank H. Bridgers, *Chairman*
John Everetts, Jr.
A. Grant Wilson
Perc M. Butler *(ExO, BOD)*
William J. Collins, Jr. *(CO)*

EXECUTIVE COMMITTEE
Walter F. Spiegel, *Chairman*
Roderick R. Kirkwood
William J. Collins, Jr.
David Rickleton
William P. Chapman

EXPOSITION POLICY
Harry G. Gragg, *Chairman (1975 – 3 yrs)*
William P. Chapman *(1974 – 2 yrs)*
Alan W. Oakes *(MA – Chairman)*
Joseph B. Olivieri *(1973 – 2 yrs)*
Thomas J. White *(MA – Vice Chairman)*
Andrew T. Boggs III *(ExO)*

FINANCE
William P. Chapman, *Chairman & CO*
David Rickelton
Paul R. Achenbach *(1975)*
Jack W. Thompson *(1973)*
A. Grant Wilson *(1974)*

HANDBOOK
William L. Bryan, *Chairman*
Jack K. James *(ExO, BOD)*
William A. Schmidt *(ExO, Journal)*
William J. Collins, Jr. *(CO)*

Subcommittee, Systems Vol., 1973
Ralph Elsea, *Chairman*
Loran V. Appleby
Robert A. Barr
John S. Blossom
Leander Economides

Subcommittee, Applications Vol., 1974
Roger W. Haines, *Chairman*
A. Herbert Bennett
Gerald L. Biehn
Louis C. Nettleship
John F. Salsburg

Subcommittee, Equipment Vol., 1975
Morris L. Markel, *Chairman*
Robert L. Boyd
Fulton M. Cooke
John M. Schultz
Eric H. Schwenker

Subcommittee, Fundamentals Vol.
Donald G. Rich, *Chairman*

HONORS AND AWARDS
John Everetts, Jr., *Chairman*
Frederic J. Weiss, *Vice Chairman*
Jack B. Chaddock
Ray C. Chewning
Paul B. Christensen
William G. Hole
Don J. Massa
George H. Meffert
Ralph B. Tilney
Morris Backer *(ExO, BOD)*
John R. Schreiner *(ExO, R&T)*
Roderick R. Kirkwood *(CO)*