Overview of Integrating Dedicated Outdoor Air Systems with Parallel Terminal Systems

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ABSTRACT

Integration of dedicated outdoor air (OA) systems with parallel terminal systems represents a huge shift in the standard engineering design paradigm. This paper will present some of the issues that guided the migration path toward separate dedicated OA ventilation systems and away from delivering the ventilation via single all-air variable air volume systems. The paper will then attempt to show that once a shift in the design paradigm in favor of a separate dedicated OA ventilation system has occurred, questions concerning the thermodynamic state of the delivered ventilation arise. Some of the boundary conditions that impact the best state of the delivered ventilation air will be addressed. Specifically, the first variable of interest is supply air temperature and its impact on the size and, hence, the cost of the terminal equipment and the terminal reheat energy. The second specific variable is the dew-point temperature (DPT) and the resultant latent cooling capability of the delivered ventilation air and its impact on the types of terminal equipment that may be practical. If in the paradigm shift the supply air DPT is dropped low enough to remove all of the space latent loads, then sensible-only terminal equipment, such as radiant ceiling panels, may become viable choices in some situations and eliminate the production of microbial problems in the spaces. Finally, this line of reasoning is extended further to note that when the space dew points are controlled with the dedicated OA, the terminal sensible equipment may use uninsulated sprinkler piping as the energy transport conduit. Companion papers (Mumma and Shank 2001; Shank and Mumma 2001; Janus 2001; Conroy and Mumma 2001) will address specific issues related to the overall integration of these concepts into a sound working design.

INTRODUCTION

The author believes that the engineering design community is on the verge of a major shift in paradigm concerning the delivery of ventilation air. This shift will be accompanied with some very exciting design opportunities that have only very recently begun to be investigated. The author also believes that this paradigm shift is inevitable because of the significant economic, comfort, and health benefits brought about by new ways of integrating the equipment with the building. As a result, the occupants of buildings will experience improved health, comfort, and productivity, and the floor-to-floor dimensions may be reduced.

THE MIGRATION PATH TO SEPARATE DEDICATED OA SYSTEMS

The idea of utilizing dedicated outdoor air (OA) systems is not new (Meckler 1986). Gershon Meckler designed systems and published articles on this subject for over 20 years. Meckler introduced this concept to the author over 15 years ago. Unfortunately, except for very unique situations with extremely restricted ceiling plenum spaces, the industry has largely overlooked this concept in favor of the all-air variable air volume (VAV) systems. Many designers have dismissed out of hand the concept of a separate dedicated OA system on the basis of cost and space limitations without a careful analysis.

More recently, some of the leaders in the industry have been advocating the dedicated OA approach. John Brady (Brady 1997) has been successfully employing dedicated OA systems in schools. Another is ASHRAE president-elect (2000-2001) Bill Coad. The author and Mr. Coad discovered their mutual interest in this subject about three years ago and

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have shared information regularly ever since. Mr. Coad's commitment to the concept of a dedicated OA system is exemplified by his role as instructor for an ASHRAE short course entitled "Variable Air Volume Design for Acceptable Indoor Air Quality." Session 3 of that short course is entitled "Enhanced Simplicity for Dependable Performance and Assured Indoor Air Quality." It is in session 3 that a strong case is made for a separate dedicated OA unit. Mr. Coad's session 3 thoughts have also been captured in a recent article entitled "Conditioning Ventilation Air for Improved Performance and Air Quality" (Coad 1999).

The author's own research on the subject has led to many publications (Mumma and Wong 1990; Mumma and Hwang 1992; Mumma and Bolin 1994; Mumma 1995; Ke and Mumma 1996; Sevigny et al. 1997; Ke and Mumma 1997a, 1997b, 1997c; Mumma and Ke 1998; Mumma and Lee 1998). Each new publication expanded the knowledge on the subject and improved (summarized in Table 1) the ability of engineers to make the all-air VAV system come closer to meeting *ANSI/ASHRAE Standard 62-1999* (ASHRAE 1999) with less energy penalty. However, another paper (Mumma and Lee 1998), entitled "Extension of the Multiple Spaces Concept of ASHRAE Standard 62 to Include Infiltration, Exhaust/Exfiltration, Interzonal Transfer, and Additional Short-Circuit Paths," which received the Willis H. Carrier Best Paper Award in 1999, illustrates the complexity of the problem and places the verifiability of an all-air VAV system to actually meet the ventilation rate procedure of ANSI/ASHRAE Standard 62 in serious question.

To illustrate this point, consider the review comments received from Dr. Max Eisenberg, executive director of CIAR (Center for Indoor Air Research), in response to the author's proposal to develop a computer-based tool to adequately accommodate the required extension of the multiple spaces equation concept as set forth in the paper (italicized words in the quote added by this author for clarity).

> Your proposal indicates that for each separate zone in a building, there are 51 variables that characterize the resulting indoor air quality, *see Table 2*. Forty-nine of them are time dependent and involve space short circuit airflow rates and paths *(variables 14 through 46, each starting with %)*, interzonal transfer air flow rates and paths *(variables 6 through 13, assuming a minimum of just 4 transfer paths)*, and air infiltration/exfiltration flow rates and paths *(variables 4 and 5). Figures 1 and 2 provide a visual representation of the short circuit paths that may exist.* Since these values can not be known at design, much less in real time, the development of a spread sheet based design tool is considered to be of little use to the profession.

A similar response was also received from ASHRAE Manager of Research via TC 4.3 (Ventilation Requirements and Infiltration) relative to a similar proposal sent there. The author took these responses as acknowledgment that the real intent of ANSI/ASHRAE Standard 62 cannot normally be met with a single all-air VAV system distributing the ventilation air.

TABLE 1
Papers That Led to the New Paradigm and Their Contribution

Authors and Date	Author's Perceived Contribution of the Publication
Mumma and Wong 1990	Documented the extent that then current all-air systems fail to meet ASHRAE Standard 62-1989. Advanced the concept of return damper modulation for OA control.
Mumma and Hwang 1992	Laboratory and field testing of the proposed method of return damper modulation for OA control.
Mumma and Bolin 1994	Real-time optimization to minimize energy achieved by resetting the critical VAV box(es) minimum(s) up and tempering to avoid overcooling.
Mumma 1995	Further refinement with shutoff VAV boxes.
Ke and Mumma 1996	Extended the multiple spaces equation in ASHRAE Standard 62 to include fan-powered boxes.
Sevigny, Ke, Mumma, and Stanke 1997	Field testing of a method of implementing real-time optimization with a parallel stand-alone computer.
Ke and Mumma 1997	OA redistribution analysis, a critical element in real-time optimization with fan-powered boxes.
Ke and Mumma 1997	Optimized supply air temperature reset, a method yet to see real utilization.
Ke, Mumma, and Stanke 1997	Simulation of eight optimization control strategies for six major climatic zones.
Ke and Mumma 1997	Dynamic occupancy sensing with CO_2, a critical requirement when the ventilation consists of a floor area and an occupant component.
Ke and Mumma 1997	Actual successful field testing of the three most promising optimization controls in a college library.
Mumma and Lee 1998	Further extension of the multiple spaces equation, illustrating the extreme complexity associated with actually complying with ASHRAE Standard 62.

TABLE 2
Variables Needed per Zone to Characterize the IAQ

Input variables per zone					
1	zone #	18	% return from transfer #3	35	% transfer #2 from transfer #4
2	primary air flow, scfm (g/s)	19	% return from transfer #4	36	% transfer #3 from supply
3	plenum air flow, scfm (g/s)	20	% exhaust from supply	37	% transfer #3 from infiltration
4	exhaust air flow, scfm (g/s)	21	% exhaust from infiltration	38	% transfer #3 from transfer #1
5	infiltration air flow, scfm (g/s)	22	% exhaust from transfer #1	39	% transfer #3 from transfer #2
6	transfer #1air flow into zone, scfm (g/s)	23	% exhaust from transfer #2	40	% transfer #3 from transfer #4
7	source zone of transfer #1 air	24	% exhaust from transfer #3	41	% transfer #4 from supply
8	transfer #2 air flow into zone, scfm (g/s)	25	% exhaust from transfer #4	42	% transfer #4 from infiltration
9	source zone of transfer #2 air	26	% transfer # 1 from supply	43	% transfer #4 from transfer #1
10	transfer #3 air flow into zone, scfm (g/s)	27	% transfer #1 from infiltration	44	% transfer #4 from transfer #2
11	source zone of transfer #3 air	28	% transfer #1 from transfer #2	45	% transfer #4 from transfer #3
12	transfer #4 air flow into zone, scfm (g/s)	29	% transfer #1 from transfer #3	46	% plenum air from return
13	source zone of transfer #4 air	30	% transfer #1 from transfer #4	47	occupancy, people
14	% return from supply	31	% transfer #2 from supply	48	OA scfm (g/s)/person
15	% return from infiltration	32	% transfer #2 from infiltration	49	CO_2 (L/s)/ person
16	% return from transfer #1	33	% transfer #2 from transfer #1	50	zone floor area, ft^2
17	% return from transfer #2	34	% transfer #2 from transfer #3	51	OA scfm (g/s)/ft^2
Global Input					
(1) OA CO_2, ppm				(2) % Primary air directly to plenum	

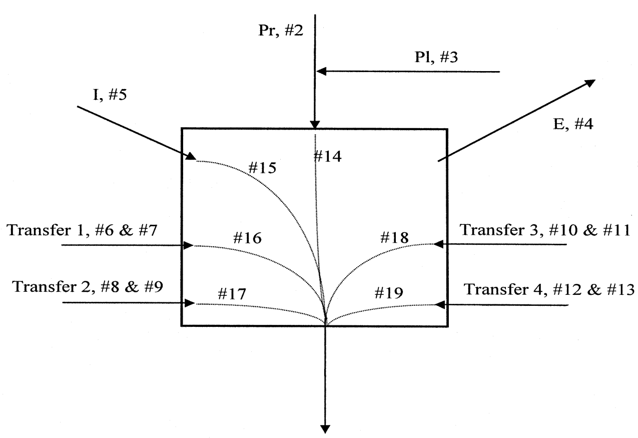

Figure 1 *Short circuit paths from the six sources to the return and Table 2 variable numbers.*

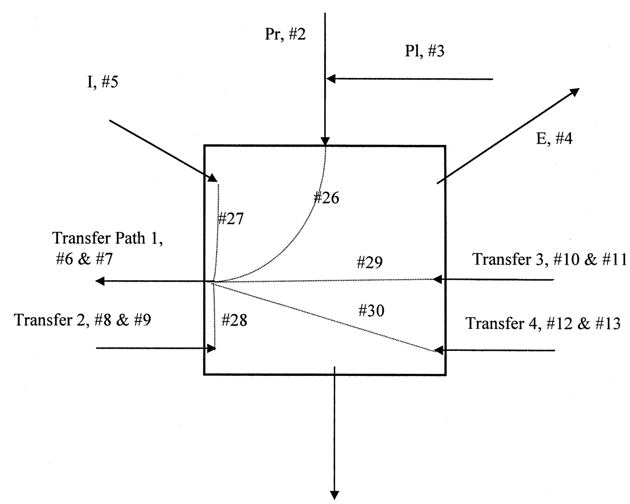

Figure 2 *Short circuit paths from the five sources to the interzonal transfer path 1 and Table 2 variable numbers.*

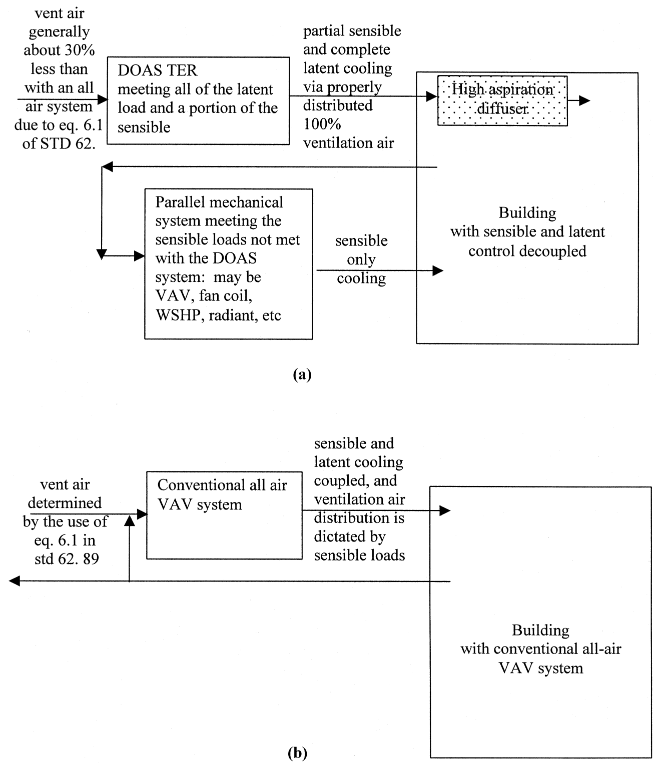

Figure 3 *(a) DOAS/parallel arrangement with decoupled latent control, the new paradigm. (b) Basic arrangement of an all-air VAV system, the current paradigm.*

Hence, the concept of a separate dedicated OA ventilation system was solidified, at least in the mind of the author, as the only reliable method of meeting ANSI/ASHRAE Standard 62, and it is also the simplest method. As a consequence, a new design/integration paradigm was born with the following two basic tenets:

1. Remove all of the outdoor air and space latent cooling loads with the dedicated outdoor air system. For this paper, the specific arrangement envisioned for the dedicated outdoor air system (DOAS) is as depicted in Figure 4. From here on, this collection of hardware will be referred to as DOAS.
2. Remove the sensible cooling loads, primarily independent of the DOAS, with a parallel (not series) mechanical system. See Figure 3 for an illustration of this concept.

ACHIEVING THE REQUIRED OA SUPPLY TEMPERATURE AND DEW-POINT TEMPERATURE ECONOMICALLY IN A DEREGULATED UTILITY MARKET

The need for dry and cool ventilation air from the DOAS was established above. Consequently, when the OA conditions are warm and humid, the dedicated OA unit must provide cooling (i.e., dry-bulb temperature [DBT] around 55ºF

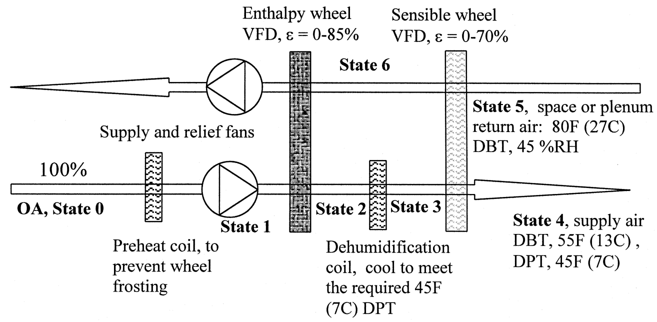

Figure 4 *General arrangement of the DOAS.*

[13ºC]) and deep dehumidification (i.e., DPT in the 45-48ºF [7-9ºC] range). This low supply air dew-point temperature is required when the outdoor air alone removes the entire space latent load and the space dew point is to be maintained around 55ºF (13ºC). Higher space dew-point temperatures result in higher than desired space relative humidities and lead to higher than desired cooling media temperatures for the sensible equipment. If the sensible equipment operates at a temperature below the space dew point, condensation and associated microbial growth is very likely. When it is dry outside, i.e., the DPT of the outside air is below the required supply air DPT, dehumidification is not required, but the air may need humidification and either sensible cooling or heating. A quick survey of some of the many options available to achieve dehumidification, cooling, heating, and humidification will now be presented.

Dehumidification and Cooling

Dehumidification can be accomplished by using either active desiccants (solid or liquid) or cooling coils. As a general rule (Harriman 1990), cooling coils are a better choice when the required DPT is above 40ºF (4ºC), and active desiccants are a better choice when the DPT is below 40ºF (4ºC). An exception to this general rule may occur when abundant inexpensive waste heat, such as from engine driven chillers, can be used for reactivation of the desiccant. Should an active desiccant approach be used, the very high temperatures leaving the desiccant (between 120-160ºF [49-71ºC], depending upon the preconditioning that may be applied to the OA prior to the desiccant dehumidification process) mean that the supply air needs considerable cooling (some may be indirect evaporative cooling) to bring it to even a neutral room air temperature. In general, active desiccants are not the best method for dehumidification in dedicated OA systems, in spite of evidence that desiccants have a beneficial microbial sanitizing influence (Kovak and Heimann 1997). Properly selected and maintained deep chilled-water cooling coils, capable of producing the approximately 45ºF (7ºC) DPT, appear to be the best overall choice. The coil capacity can be modulated to match the transient load conditions. In addition, off-peak air conditioning (ice or water thermal storage) can also be used, shifting energy demand and consumption to less expensive off-peak electrical rates. Direct expansion (DX) deep cooling coils do not offer as good a solution for three reasons. Primarily, at part-load conditions, it is necessary to cycle the compressors, which results in the loss of dehumidification capability, and in fact the wet coil surface becomes a humidifier, when the compressors are cycled off. Secondarily, the availability of chilled water to serve the terminal equipment is necessary for some of the equipment choices. Finally, off-peak utility rates cannot be used. The use of an enthalpy wheel (passive desiccant wheel, with microbial sanitizing effects), operating between the outdoor and relief air streams, offers a powerful tool to reduce both the summer cooling and dehumidification requirements and the winter heating and humidification requirements. For these reasons, the enthalpy wheel is considered a critical element in the design of the DOAS.

Reheat

In the event that the required supply DPT is lower than the required DBT and the dehumidification is accomplished with a cooling coil, reheat will be necessary. Many options are available to the engineer, including

- fuel consuming reheat coils,
- reheat from waste heat, for example from an engine driven chiller,
- runaround heat recovery, designed to remove heat from the OA (thus precooling the air and reducing the cooling coil load) prior to the cooling coil and reheating the air after the cooling coil, or
- sensible heat transfer between the return air and the supply air, using either a plate type heat exchanger, heat pipe, or a sensible wheel.

The last two appear to hold great promise.

Winter Humidification

Winter humidification is often overlooked in current design practice. However, since it is recommended that the space RH be maintained above 40% for optimal thermal comfort and to minimize microbial related health risks (Sterling et al. 1985), humidification is important. It is particularly important at the current OA flow rates required by ANSI/ASHRAE Standard 62, which can lead to a very low indoor air relative humidity when humidification is not used. Engineers have at their disposal two general methods of humidification: hot or cold (Mumma et al. 1997) steam or water addition or recovery of moisture in the relief air stream with an enthalpy wheel. In the steam (approximately an isothermal process) and water (approximately an adiabatic saturation process) addition processes for humidification, energy is required to either produce steam or preheat the air prior to the adiabatic cooling caused by liquid water injection (sprays, ultrasonic, pads, etc.). From an energy consumption point of view, the passive desiccant wheel is by far the best option and is the author’s choice. Further, the passive desiccant wheel does not introduce air stream/moisture mixing problems and or potential microbial problems as the steam and water humidifiers do, making it a much better choice.

In conclusion, it is recommended that the DOAS consist of the following components: a chilled water coil and a heat recovery system (consisting of a sensible wheel, providing reheat between the relief air and the supply air) and a passive desiccant wheel (operating between the relief air and the OA before the cooling coil) as illustrated in Figure 4. The Mumma and Shank (2001) paper develops the details and supports this conclusion. Another paper addresses the control requirements for the DOAS (Mumma 2001).

DETERMINING THE OPTIMUM OA SUPPLY DRY-BULB AND DEW-POINT TEMPERATURES FOR ALL SEASONS

Determining the optimum OA supply dry-bulb and dew-point temperatures for all seasons is not a simple and straightforward endeavor. The two variables, DBT and DPT, can be interrelated. Notwithstanding the interrelationship, they will be discussed independently, starting with the DBT. Implicit in the new design paradigm advocated in this paper, the latent loads will be completely removed by the dry ventilation air delivered to the space. The parallel terminal equipment will only need to accommodate the sensible loads not met by the ventilation air. Clearly, the lower the supply air DBT, the smaller the terminal sensible cooling equipment need be (and smaller first cost) and the more sensible cooling is transferred to the DOAS. Offsetting the reduction in first cost of the terminal equipment creates the potential for an increase in the terminal reheat energy that may be necessary to avoid overcooling the spaces with the ventilation air. Generally, a low supply air temperature is justified for spaces with fairly high internal loads, as is the case in most commercial, educational, and institutional buildings. Detailed treatment of this subject is found in the paper by Shank and Mumma (2001). It may be stated now, however, that supplying the dry ventilation air is the correct concept, but supplying it at a neutral temperature (70-75ºF [21-24ºC]) as recommended by representative manufacturers is certainly not the optimum. It can be made to work, however.

Determining the optimum supply air DPT is much more difficult, and it appears that nothing has been published to establish a clear procedure for determining the optimum DPT to date. Nonetheless, the optimum DPT is a function of at least the following:

- water-side economizer, if used,
- terminal equipment pumping first and operating costs,
- terminal equipment piping first costs, and
- terminal equipment size and operating costs.

Rather than determining the optimum, a workable DPT is generally established. A working DPT is a function of the following:

- the space latent loads,
- the space design DPT, which defines the lower limit of the sensible equipment surface temperatures to avoid condensation (the heat transfer rate per unit area at the terminal equipment is strongly influenced by the temperature difference between the space temperature and the equipment surface temperatures, suggesting space DPTs no warmer than 55ºF [13ºC]), and
- the required ventilation air.

Further work on this important topic is currently underway by the author.

MEETING THE SPACE SENSIBLE LOADS

As a general observation, the ventilation air may be supplied with temperatures ranging from 75ºF to 45ºF (24ºC to 7ºC). In this range of supply temperatures, for typical ventilation air flow rates, from 0% to 30% of the space sensible loads can be accommodated by the OA. The balance of the space sensible loads must be accommodated with many optional equipment choices. Those choices include

- a parallel all-air VAV system without airside economizer, for simplicity, as advocated by Bill Coad (1999);
- packaged unitary water-source heat pumps;
- fan coil units;
- packaged unitary equipment;
- radiant ceiling panels (in this case, the diffusers selected for the DOAS need to be of the high aspiration type to produce the required air diffusion performance index [ADPI] for air movement and comfort).

All of the sensible cooling equipment choices can result in good workable designs; however, the author has the most interest in the radiant cooling application, since this technology is scarcely used in the United States. For more engineering details about applying this approach, refer to the paper by Conroy and Mumma (2001). The small ventilation air ductwork and parallel radiant cooling panel terminal equipment offer a significant opportunity to reduce the required floor-to-floor dimension.

INTEGRATING THE SPRINKLER FIRE SUPPRESSION TRANSPORT SYSTEM WITH THE THERMAL TRANSPORT SYSTEM

Gershon Meckler, who did the work to make this approach acceptable to the National Fire Protection Authority (NFPA) (NFPA 13 1999), first introduced the author to this concept in 1984. In his case, uninsulated sprinkler piping was used to transport energy in a water-loop heat pump system. However, the concept is equally applicable for terminal sensible equipment where the space DPTs are maintained below the transport fluid temperatures. This physical and functional integration is a large first cost saving item. See the Janus (2001) paper for details on the following:

- Codes
- Pipe sizes/insulation savings
- Classes of buildings that use dry and wet sprinkler systems (i.e., buildings where the two transport systems can be physically and functionally integrated)
- Integrating the transport system layouts
- Controls of the thermal transport pumps to ensure no condensation on the pipes or the terminal heat exchangers (HTX)
- Controls for the fire emergency mode
- Economic benefits
- Impact of the integration on plenum depth
- Methods of integrating both the design and trades jurisdictional areas
- Case studies

CONCLUSIONS AND RECOMMENDATIONS

The new paradigm shift in the design of building comfort control systems is in its early stages. The author has written this paper to help facilitate an understanding of what is believed to be the wave of the future. The requirements of the new paradigm include

- separating the OA system from the space-conditioning systems to ensure proper ventilation in all occupied spaces;
- conditioning the OA to handle all of the space latent load and as much of the space sensible load as is economically feasible without excessive reheat;
- maximizing the cost-effective use of energy recovery equipment;
- integrating the fire suppression and energy transport systems; and
- employing radiant ceiling heating and sensible cooling for occupant thermal control where appropriate.

As more and more of these new OA design systems are placed in service, with superior results in environmental control, IAQ, and cost performance, the skeptics will not be able to honestly say "when in doubt leave it out" but must say "let's just do it."

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