THERMAL DISTRIBUTION SYSTEMS IN COMMERCIAL BUILDINGS

Rick Diamond, Craig Wray, Darryl Dickerhoff, Nance Matson, and Duo Wang

Lawrence Berkeley National Laboratory

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Executive Summary

Introduction. Previous research suggests that HVAC thermal distribution systems in commercial buildings suffer from thermal losses, such as those caused by duct air leakage and poor duct location. Due to a lack of metrics and data about the potentially large energy savings from reducing these losses, the California building industry has mostly overlooked energy efficiency improvements in this area.

Purpose. The purpose of this project is to obtain the technical knowledge needed to properly measure and understand the energy efficiency of thermal distribution systems in commercial buildings. We expect that this new information will assist the California building industry in designing better thermal distribution systems for new commercial buildings and in retrofitting existing systems to reduce their energy consumption and peak electrical demand.

Project Objectives. The specific technical objectives for this project were to:

1. Develop metrics and diagnostics ("yardsticks" and measurement techniques) for determining the efficiencies of commercial thermal distribution systems.
2. Develop information that the California building industry (e.g., HVAC system design engineers and installers) can use to design new thermal distribution systems, estimate energy efficiency, and prevent or reduce the incidence of problems that have been identified in existing commercial thermal distribution systems.
3. Determine the energy impacts associated with duct leakage airflows in an existing large commercial building, which could be mitigated by applying duct retrofit technologies.

Project Outcomes: Based on the project objectives, the project has three primary outcomes:

Metrics & Diagnostics. The most important metric that we identified and defined characterizes the overall efficiency of the thermal distribution system in large commercial buildings: transport energy (e.g., total energy used to transport air) per unit thermal energy delivered. This metric is useful for comparing the relative performance of various types of thermal distribution systems. We recommend that California's Title 24 compliance process for large commercial buildings include quantification of this metric.

Our field tests of diagnostics focused on measurements of duct leakage airflows, fan airflows, and fan power. In particular, of the two duct leakage diagnostics that we tested, only one reliably determined duct leakage airflows: it involves accurately measuring airflows entering and exiting the duct system—the difference is the duct leakage. With further development and testing, we expect this diagnostic will be useful in developing a database that characterizes the distribution of duct leakage airflows in California's large commercial buildings.

Characterization. Because there has been very little characterization of the actual performance of thermal distribution systems in large commercial buildings, we carried out an extensive characterization of one of these systems. After this characterization, we determined that the test building showed every indication of a "tight" thermal distribution system: good application of mastic, metal bands at joints, and overall high quality. To demonstrate duct leakage impacts, we

installed temporary calibrated leaks and monitored their effects on the system energy consumption and demand.

Energy impacts. The principal outcome from this project is that duct leakage airflows can have a significant energy impact in large commercial buildings. Our measurements indicate that adding 15% duct leakage at operating conditions leads to an increase in fan power of about 25 to 35%. These findings are consistent with the impacts of increased duct leakage airflows on fan power that have been predicted by previous simulations.

Conclusions. The primary outcome from this project is the measured confirmation that duct leakage airflows can significantly increase fan energy consumption in large commercial buildings. In addition, we have defined a new metric for distribution system efficiency, demonstrated a reliable test for determining duct leakage, and developed new techniques for duct sealing. The parallel story in the residential and small commercial sector has shown that from the comparable stage in that research to maturity of technology adoption (e.g., commercialization and inclusion in standards) was approximately ten years. We conclude that a concerted effort will be necessary to make the same—or better—progress for the large commercial sector.

Recommendations. Based on the project findings, our recommendations for further work are:

- Further develop the duct leakage airflow diagnostic and submit it to the American Society for Testing and Materials (ASTM) for adoption as a standard method of test.
- Work with California's Title 24 staff to introduce a requirement for quantifying and reporting the "overall efficiency of the distribution system" metric for new large commercial buildings. Once we have a good understanding of the range of duct system efficiencies from reported data, we could then use these data to set guidelines for minimum acceptable levels.
- Develop specifications for maximum allowable duct leakage airflows and for duct sealing in new construction.
- Continue collaborative work with the U.S. Department of Energy, University of California, and private sector (e.g., Carrier, Eley Associates, Taylor Engineering) to transfer information to the building industry.
- Evaluate the performance of the thermal distribution system at the demonstration building over a heating season, with and without the added duct leakage. The investment of time and equipment at the demonstration building makes it worthwhile to continue monitoring the system in order to look at energy savings over the year.
- Survey additional sites to start a database of duct leakage characteristics in large commercial buildings. This work is currently planned with funding from the U.S. Department of Energy and would benefit from co-funding by the CEC.

Benefits to California. We have identified several benefits that result directly from this study or that will accrue over time as necessary information and infrastructure develops further:

Benefits in electricity savings. The primary benefits from having tight duct systems are electricity savings. We estimate that eliminating duct leakage airflows in half of California's existing large commercial buildings has the potential to save about 560 to 1,100 GWh annually (about $60-$110 million per year or the equivalent consumption of about 83,000 to 170,000 typical California houses), and about 100 to 200 MW in peak demand.

Benefits to future buildings. The identification of a metric for characterizing distribution system performance allows us to recommend its inclusion in the next round of California's Title 24 as a way of characterizing the new building stock. Once we have a good database of new duct system characteristics, we can set reasonable targets for distribution system performance, which will ultimately lead to further energy savings in this sector.

Benefits to new buildings in the UC system. As an outgrowth of our work on this project, we have been working with members of the design team on duct specifications for the new University of California at Merced campus. In particular, we have reviewed the draft design documentation for their duct systems and recommended changes to the specifications. While we do not have a specific energy saving calculated for this work, we see the benefits extending to other UC campuses once others use these specifications.

Benefits in building operations and maintenance. A willing partner in this work has been Thomas Properties, a major manager of buildings, both public and private. Through this study, they have seen the benefits not only of improvements in duct diagnostics, but also in feedback on their HVAC system performance via the EMCS. By working closely with the private sector, we have seen the transfer of knowledge and the improvements in building operation in a major public facility.

Benefits to future engineers. A key aspect of our research team has been the inclusion of numerous students. While they may not have appreciated the long hours at the field site, they are now familiar with a number of important lessons about the performance of thermal distribution systems. As future engineers—not all of who may practice in California—we expect that some will become leaders in this area.

Abstract

Previous research suggests that HVAC thermal distribution systems in commercial buildings suffer from thermal losses, such as those caused by duct air leakage and poor duct location. Due to a lack of metrics and data showing the potentially large energy savings from reducing these losses, the California building industry has mostly overlooked energy efficiency improvements in this area. The purpose of this project is to obtain the technical knowledge needed to properly measure and understand the energy efficiency of these systems. This project has three specific objectives: to develop metrics and diagnostics for determining system efficiencies, to develop design and retrofit information that the building industry can use to improve these systems, and to determine the energy impacts associated with duct leakage airflows in an existing large commercial building. The primary outcome of this project is the confirmation that duct leakage airflows can significantly impact energy use in large commercial buildings: our measurements indicate that adding 15% duct leakage at operating conditions leads to an increase in fan power of about 25 to 35%. This finding is consistent with impacts of increased duct leakage airflows on fan power that have been predicted by previous simulations. Other project outcomes include the definition of a new metric for distribution system efficiency, the demonstration of a reliable test for determining duct leakage airflows, and the development of new techniques for duct sealing. We expect that the project outcomes will lead to new requirements for commercial thermal distribution system efficiency in future revisions of California's Title 24.

Keywords: Buildings, HVAC, ducts, fans, energy, metrics, diagnostics, retrofits

1. Introduction

1.1 Background

Heating, ventilating, and air conditioning (HVAC) equipment in California commercial buildings consumes approximately one quarter of the electrical energy used by these buildings and accounts for about half of their peak electrical demand (Brook 2002). Previous research suggests that the HVAC thermal distribution systems in these buildings suffer from a number of problems, such as thermal losses due to duct air leakage and poor duct location (Xu et al. 1999, 2002). Despite the potential for large energy savings by reducing thermal losses in commercial buildings (Franconi et al. 1998), the California building industry has mostly overlooked this area as a target for energy efficiency improvements. One reason is that metrics for determining the efficiencies of commercial thermal distribution systems are poorly defined. Another reason is that the utility of duct diagnostic and retrofit technologies to identify and reduce duct leakage, duct conduction losses, and associated energy consumption and demand remains undetermined.

As an example of the complexities involved in understanding thermal distribution system performance, consider a variable-air-volume (VAV) HVAC system in a large commercial building. Although the conditioned air that leaks from supply ducts is captured in the return air, and may be regained from a thermal viewpoint, the leakage airflow does not reach the conditioned spaces directly. To maintain the main duct static air pressure at its set point, all leakage upstream of the VAV boxes must be made up by an increase in the supply fan airflow. Leakage downstream of the VAV boxes must be made up by supplying more air to the VAV boxes. To deliver more supply air, VAV box primary air dampers need to open further.

Consequently, to maintain the main duct static pressure at its set point, an increase in the supply fan airflow is also needed to compensate for the downstream leakage airflows. Because the relationship between fan power and airflow is somewhere between a quadratic and cubic function, the increase in supply airflow means that supply fan power consumption increases, with a large fraction of this fan power used just to move the leaking air. Note that some of the thermal losses associated with duct leakage are not entirely recaptured during periods of economizer use, because relief fans discharge some of the return air directly to outdoors to maintain building envelope pressure differentials that would otherwise increase due to the increased outdoor airflows entering the building through the economizer.

The overall goal of this project is to obtain the technical knowledge needed to properly measure and understand the energy efficiency of thermal distribution systems in commercial buildings. We expect that this new information will assist the California building industry in designing better thermal distribution systems in new commercial buildings and in retrofitting existing systems to reduce their energy consumption and peak electrical demand.

This project contributes to the PIER program objective of improving the energy cost and value of California's electricity in two ways. One is by demonstrating through measurements how leaky or poorly designed thermal distribution systems in commercial buildings waste the energy that is used to condition air (e.g., cooling, heating, dehumidification). The other is by developing methods to identify and correct these problem systems. We expect that the knowledge gained from this research will be used to craft new requirements for commercial duct system efficiency in future revisions of California's Title 24.

1.2 Project Objectives

The specific technical objectives for this project are to:

1. Develop metrics and diagnostics ("yardsticks" and measurement techniques) for determining the efficiencies of commercial thermal distribution systems.
2. Develop information that the California building industry (e.g., HVAC system design engineers and installers) can use to design new thermal distribution systems, estimate energy efficiency, and prevent or reduce the incidence of problems that have been identified in existing commercial thermal distribution systems.
3. Determine the energy impacts associated with duct leakage airflows in an existing large commercial building, which could be mitigated by applying duct retrofit technologies.

There are two overall economic performance objectives of this project. One is to lower the cost for building owners of performing diagnostic services on commercial thermal distribution systems, by developing quick field measurement techniques. The second is to lower space conditioning and ventilating costs to commercial-building electric ratepayers, by developing duct retrofit technologies that improve the performance of thermal distribution systems.

1.3 Report Organization

This report presents our findings and recommendations that have resulted from investigating the impacts of thermal distribution systems on energy use in commercial buildings. Most of the work focuses on large commercial buildings rather than on small commercial buildings, because much less is known about thermal distribution system performance in large commercial buildings.

In **Section 2 Project Approach**, we discuss the tasks that we undertook and our approach to the research to accomplish our objectives. In particular, we discuss changes to the testing procedures that we undertook and the need for system modifications during those tests.

In **Section 3 Project Outcomes**, we present the key results from our investigations.

In **Section 4 Conclusions and Recommendations**, we present what we learned from the research and what we recommend for future activities.

Following the **Glossary** and **References**, there are four technical **Appendices**:

> "**Appendix I.** Metrics and Diagnostics" provides a starting point in the development of a set of metrics and diagnostics that describe thermal distribution system performance in both small (thermally dominated) and large (fan-power dominated) commercial buildings. The appendix discusses energy (consumption and demand) metrics, as well as environmental indices (e.g., health, comfort, and safety). It also outlines several one-time and short-term diagnostics that can be used to quantify these metrics.
>
> "**Appendix II.** In-Situ Characterization" provides a general description of the large commercial building and its thermal distribution system that we characterized; a description of our monitoring and diagnostic activities; a summary of our preliminary duct leakage findings; a description of the HVAC system airflow diagnostics that we carried out prior to and during our duct leakage intervention tests; and a summary of our HVAC fan power measurements and other field study findings.
>
> "**Appendix III.** Duct Sealing Techniques" discusses the need for duct sealing techniques that reach duct leaks in existing commercial buildings without having to access and seal every joint manually. It then describes the development and laboratory testing of a mobile aerosol-sealant injection system (MASIS) that can use multiple injectors simultaneously to seal multiple duct sections. To help the reader understand the multiple injector system, the appendix includes a description of the aerosol sealing technology. At the end of the appendix, we also discuss whether there is a need to develop field retrofit techniques for sealing duct system components such as VAV boxes and supply grilles.
>
> "**Appendix IV**. Production Readiness Plan" discusses our production readiness plan for retrofitting thermal distribution systems in large commercial buildings. In particular it summarizes our market transfer work to date, discusses steps needed to commercialize the aerosol sealing technology for use in large commercial buildings, and provides recommendations for future work.

2. Project Approach

In the original project plan, as part of obtaining the technical knowledge needed to properly measure and understand the energy efficiency of thermal distribution systems in commercial buildings, much of the work was intended to focus upon the utility of aerosol-based duct sealing technologies, for both small and large commercial buildings. Technical tasks were divided into three groups that addressed multiple project objectives in some cases:

- Develop metrics and diagnostic approaches for small and large commercial buildings.
- Carry out in-situ characterizations of two commercial buildings (one small and one large) using the metrics and diagnostics approaches defined in the project, before and after sealing the ducts using aerosol-based duct sealing technologies, and determine the energy savings associated with reducing the duct leakage.
- Develop aerosol-based duct sealing technologies, test them in the laboratory, and then apply them in the field to retrofit the duct systems in the buildings selected for the in-situ characterization task.

As in any research effort, the results along the way also shaped the work. Based on input from the PAC team and the PIER Buildings Program team, the project evolved. Changes included:

- We acquired additional funding from the U.S. Department of Energy to develop and test the aerosol-sealing technology itself. This meant that the CEC project could narrow its focus to developing protocols and control algorithms for field applications in large commercial buildings, which require the use of multiple injectors to achieve acceptable sealing efficiencies.
- We acquired funding from the Sacramento Municipal Utility District to carry out a duct leakage intervention – energy impact study on several small commercial buildings. This meant that the CEC project could shift its focus away from small commercial buildings and toward large commercial buildings, where less is known about the performance of their thermal distribution systems.
- Once planning began for the in-situ characterization and duct leakage intervention study in a large commercial building, it quickly became apparent that this effort would consume most of the resources planned for these activities, leaving none to address a second large building. In particular, our experimental design called for extensive characterization of the HVAC system operation on an intervention floor before and after modifying duct leaks on this floor; plus less detailed characterization of a separate floor within the same building (used as a control floor with no changes to the HVAC system, as a basis for comparison to the intervention floor). Based on consultations with the PIER Buildings Program team, the decision was made to study one building in detail and understand it well, rather than diverting resources to two buildings and being less thorough in each building.
- During our in-situ characterization efforts in the large commercial building, we found that the duct leakage diagnostics gave disparate results. In particular, although pressurization tests indicated the duct system was leaky, in reality it was tight (small leakage airflows). This meant that using aerosol sealing to reduce duct leakage was not an option as an intervention method, and that leaks would need to be added to assess the energy impact of duct leakage. To support that effort, additional funding was acquired from the U.S. Department of Energy to add calibrated duct leaks both upstream and downstream of VAV boxes. This meant that we could separately determine the impact of each set of leaks. Given that the multiple injector aerosol-based sealing technology is ready to deploy and may be the most practical method for retrofit duct sealing applications, we had still hoped to seal the installed leaks using aerosol sealing, as a field demonstration of this

technology. However, we were unable to obtain approval for such sealing in the test building.

Appendices I through IV describe our research efforts in detail.

3. Project Outcomes

This section presents the key results from our investigations, in the same order as the three objectives.

Objective #1: Develop metrics and diagnostics ("yardsticks" and measurement techniques) for determining the efficiencies of commercial thermal distribution systems.

Metrics. As described in Appendix I, we identified and defined eleven metrics for characterizing the thermal performance of distribution systems in large commercial buildings. The most important of these metrics characterizes the overall efficiency of the distribution system: transport energy (e.g., total energy used to transport air) per unit thermal energy delivered. This metric is useful for comparing the relative performance of various types of thermal distribution systems. Determining this parameter in a building that has already been built is difficult because measuring the total heating or cooling energy delivered to each zone requires air temperature and flow sensors at every grille and temporally continuous measurements. However, calculating this parameter for a building on paper or using a computer is relatively straightforward, excluding any impacts of improper installation or operation. We recommend that California's Title 24 compliance process for large commercial buildings include quantification of this metric.

Diagnostics. We field tested six diagnostics that can be used to quantify the performance of existing thermal distribution systems, with a focus on diagnostics to measure duct leakage airflows, fan airflows, and fan power. As we discuss in Appendix II, only one of the two duct leakage diagnostics that we tested could reliably determine duct leakage airflows: it involves accurately measuring airflows entering and exiting the duct system—the difference being the duct leakage. A significant barrier to the widespread use of this diagnostic is the need to rapidly measure the flows exiting the duct system at many supply grilles. Through our field tests of several commercially available flow hoods, we found one that can give the same results as our reference research-grade device, within the uncertainty specification of the reference (bias and RMS errors less than 2%). Our tests also indicated that this hood could measure 100 grille airflows in less than two hours, which is rapid enough to make the diagnostic practical. Because hood accuracy depends on grille type, further development and testing is needed, but we expect this diagnostic will be useful in developing a database that characterizes the distribution of duct leakage airflows in California's large commercial buildings.

Objective #2: Develop information that the California building industry (e.g., HVAC system design engineers and installers) can use to design new thermal distribution systems, estimate energy efficiency, and prevent or reduce the incidence of problems that have been identified in existing commercial thermal distribution systems.

Characterization. Because there has been very little characterization of the actual performance of thermal distribution systems in large commercial buildings, we carried out an extensive characterization of one of these systems. Of particular note, we used duct pressurization techniques similar to those described by SMACNA (1985) to determine duct leakage areas. Our initial measurements of leakage area and pressures in the ducts indicated that the duct system downstream of VAV boxes was leaky: about mid-range compared to the leakage (58 to 606 cfm at 1 in. w.c. pressure per 100 ft^2 of duct surface area) of branch ducts that we have tested in other large commercial building systems (Xu et al. 2002). However, this was only part of the story. Using the duct leakage airflow diagnostic described above, we determined that the actual airflow through the duct leaks was small (about 5% of total air-handler supply airflow at operating conditions), and consequently less significant. The test building showed every indication of a “tight” thermal distribution system: good application of mastic, metal bands at joints, and overall high quality.

Objective #3: Determine the energy impacts associated with duct leakage airflows in an existing large commercial building, which could be mitigated by applying duct retrofit technologies.

Energy impacts. Because the duct system in the study building was sufficiently airtight that expected energy impacts would be too small to measure, we installed temporary calibrated leaks to demonstrate duct leakage impacts. Specifically, we added 15% duct leakage to make a total of 20% at operating conditions. We feel that the added leaks represent what we might find in other buildings, but need to validate this assumption by making duct leakage airflow measurements in more buildings.

During the cooling season, the amount of electrical power required by the HVAC system to transport conditioned air includes the power to drive the air-handler supply fans, relief fans, and VAV box induction fans. In reviewing the patterns of fan operation, we found that the supply fan and induction fan airflows and operation are impacted by the introduction of additional duct leakage. The relief fans are operated to maintain building pressure set points and run as needed during pre-cooling and economizer modes. As a result, the relief fans have an irregular operational pattern. We did not see a correlation between duct leakage and relief fan operation. Therefore, we only discuss the impact of duct leakage on the air-handler supply fans and the induction fans, and not on the relief fans.

The principal outcome from this project is that the energy impact of duct leakage in large commercial buildings can be substantial. We found that the added leakage leads to an increase in air-handler supply fan power of about 37%, and an overall increase in total fan power (air-handler supply fans plus induction fans) of about 26%. The total fan power increase is lower because the added duct leakage causes induction fans to operate less often.

Previous simulations (Franconi et al. 1998) have suggested that the energy impacts of 20% duct leakage are larger: on the order of 60% to 70% of the total fan energy consumption. However, the duct leakage fraction in the simulations is normalized by nominal design supply airflow rather than by operating supply airflow. Redefining our duct leakage fraction to match the definition used in the simulations means that the duct leakage that we added was about 10% of the nominal design supply airflow. Given that our leakage fractions were about half of those used in the simulations, and assuming that fan power is somewhere between a quadratic and cubic function of airflow, our measurements are consistent with the simulation results.

4. Conclusions and Recommendations

4.1 Conclusions

The primary outcome from this project is the measured confirmation that duct leakage airflows can significantly increase fan energy consumption in large commercial buildings. In addition, we have defined a new metric for distribution system efficiency, demonstrated a reliable test for determining duct leakage airflows, and developed new techniques for duct sealing. The parallel story in the residential and small commercial sector has shown that from the comparable stage in that research to maturity of technology adoption (e.g., commercialization and inclusion in standards) was approximately ten years. We conclude that a concerted effort will be necessary to make the same—or better—progress for the large commercial sector.

4.2 Commercialization Potential

Market Transfer Efforts. Our project activities have already resulted in market transfer efforts:

- **Codes and Standards**. California's Title 24 currently has no performance criteria for duct systems in large commercial buildings. We have identified a metric for characterizing duct system efficiency that Title 24 reports should include for all new commercial buildings. Once we have a good understanding of the range of duct system efficiencies, we could then set guidelines for minimum acceptable levels.
- **CA Public Sector**. University of California staff is already asking for the development of duct tightness criteria for the new buildings at UC Merced. They are also interested in the measurement and verification procedures to ensure that these criteria have been met. These criteria could be adopted for new construction at University of California and California State University campuses throughout the state.
- **PIER-related activity**. Taylor Engineering, Eley Associates, and the Center for the Built Environment at UC Berkeley have all come on board this project as interested co-participants. The work at the test building in Sacramento has been a fertile test bed for several groups interested in sharing our monitoring capabilities to do unique measurements of HVAC systems in commercial buildings.
- **Synergistic funding with US DOE**. This project has benefited from over $400k of support from the Building Technologies office at the U.S. Department of Energy. DOE plans to continue supporting work in this area, including continued efforts at the test building in Sacramento to further assess duct leakage diagnostics, as well as measurements of duct leakage at different sites.

Production Readiness Plan. In addition to the market transfer work identified above, there is one specific aspect of the study that has large commercialization potential: the aerosol sealing technology. There are some development tasks that may be appropriate for the public sector to pursue, given the lack of R&D that is currently done in the building's sector. Once this work is done, we expect the private sector to fully commercialize this technology. The steps needed for its commercialization are:

- Document the health and safety performance for the aerosol sealant, to eliminate barriers that may prevent adoption of this technology to seal ducts in large commercial buildings.

- Characterize the energy savings potential of existing large commercial buildings, by determining the actual range of leakage distributions in these buildings.
- Demonstrate aerosol sealing in a sample of commercial buildings.
- License technology to the private sector, which could then train contractors and produce equipment to reach the market.

Appendix IV describes these steps in more detail.

4.3 Recommendations

Based on our findings, our recommendations for further work are as follows, arranged in the same order as the three objectives:

Objective #1: Develop metrics and diagnostics ("yardsticks" and measurement techniques) for determining the efficiencies of commercial thermal distribution systems.

> **Recommendation #1**: Further develop the duct leakage airflow diagnostic and submit it to the American Society for Testing and Materials (ASTM) for adoption as a standard method of test.
>
> **Recommendation #2**: Work with California's Title 24 staff to introduce a requirement for quantifying and reporting the "overall thermal efficiency of the distribution system" metric for new large commercial buildings. Once we have a good understanding of the range of duct system efficiencies from reported data, we could then use these data to set guidelines for minimum acceptable levels.

Objective #2: Develop information that the California building industry (e.g., HVAC system design engineers and installers) can use to design new thermal distribution systems, estimate energy efficiency, and prevent or reduce the incidence of problems that have been identified in existing commercial thermal distribution systems.

> **Recommendation #3**: Develop specifications for maximum allowable duct leakage airflows and for duct sealing in new construction.
>
> **Recommendation #4**: Continue collaborative work with the U.S. Department of Energy, University of California, and private sector (e.g., Carrier, Eley Associates, Taylor Engineering) to transfer information to the building industry.

Objective #3: Determine the energy impacts associated with duct leakage airflows in an existing large commercial building, which could be mitigated by applying duct retrofit technologies.

> **Recommendation #5**: Evaluate the performance of the thermal distribution system at the demonstration building over a heating season, with and without the added duct leakage. The investment of time and equipment at the demonstration building makes it worthwhile to continue monitoring the system in order to look at energy savings over the year.
>
> **Recommendation #6**: Survey additional sites to start a database of duct leakage characteristics in large commercial buildings. This work is currently planned with funding from the U.S. Department of Energy and would benefit from CEC co-funding.

The final Project Advisory Committee meeting in November 2002 also generated 11 recommendations for further work. These recommendations are in the form of desired outcomes for improving thermal distribution systems in large commercial buildings, both new and existing, by 2010. Many of these outcomes reflect our recommendations, but they also represent a broader scope. The desired outcomes are as follows:

A. Stock Characterization and Energy Savings Potential

1. *Stock Characterization*. An assessment of thermal distribution systems in the large commercial building stock (e.g., magnitude and location of leakage airflows).
2. *Current Practice*. Characterization of existing practices for duct installation and air sealing.
3. *Energy Impacts*. An expanded understanding of the energy impacts of thermal distribution system characteristics (e.g., impacts related to duct leakage and thermal conduction) and a ranking of the issues that warrant further study.

B. Design and Construction

1. *Design Guides*. Duct design and construction guidelines that focus on the most important issues in terms of their impacts on energy performance.
2. *Simulation Tools*. Mainstream simulation programs that can be used as design tools to predict distribution system performance.
3. *Technology Adoption*. Use of low-leakage duct components and joints, which will reduce or eliminate the need for widespread duct leakage testing.
4. *Specifications*. Specifications for achieving tight ducts within the normal building delivery process.
5. *Design Intent Linkage*. Improved communications between design intent, field construction, and operation.

C. Codes and Standards

1. *Metrics*. Further development of proposed metrics for system characterization (i.e., expanded definitions of what each metric includes, and how each is determined or measured).
2. *Standards*. Defined standards for distribution system installation.
3. *Test Procedures*. A standard test procedure for flow hoods.

D. Operations & Maintenance, Diagnostics, and Commissioning

1. *Commissioning Toolkit*. A toolkit for commissioning ducts.
2. *Real-Time Diagnostics*. A diagnostic method for measuring the energy use of distribution systems during operation, so that building operators can detect and rectify deficiencies in space conditioning energy delivery.
3. *Information Transfer*. Dissemination of our current knowledge to the critical players.

4.4 Benefits to California

We have identified the following benefits that result directly from this study or that will accrue over time as necessary information and infrastructure develops further:

Benefits in electricity savings. The primary benefits from having tight duct systems are electricity savings. Based on CEC Year 2000 estimates (Brook 2002), site electricity consumption for commercial buildings in California was 91,771 GWh that year, with a peak demand of 20,150 MW; 27% (25,185 GWh) of this energy and 51% (10,180 MW) of this peak demand were related to HVAC (heating, ventilating, and cooling) equipment operation in these buildings. The CEC also estimates that 39% (9,822 GWh) of this HVAC consumption and 21% (2,138 MW) of this HVAC demand was associated with fan operation; central system supply and return fans represented 56% (5,460 GWh) of this fan-related consumption and 47% (1,010 MW) of this fan-related demand.

Using CEC Year 2000 estimates (Rohrer 2000), there was about 5,690 million ft^2 of commercial building floor area that year in California. Assuming that the fraction of large commercial buildings in California for the Year 2000 is the same as in the entire US Pacific region for Year 1999 (78%, EIA 2002), then there was about 4,440 million ft^2 of large commercial building floor area in California during the Year 2000. Assuming that central system supply and return fans are only used in large commercial buildings, this means that the average normalized fan power associated with peak demand for these fans was about 0.23 W/ft^2. This value is similar to the peak demand values that we measured for air-handler supply fans in the Sacramento test building (0.20 to 0.25 W/ft^2).

We estimate that eliminating duct leakage airflows in half of California's existing large commercial buildings has the potential to save about 560 to 1,100 GWh annually (about \$60-\$110 million per year or the equivalent consumption of about 83,000 to 170,000 typical California houses), and about 100 to 200 MW in peak demand. It is important to recognize that these potential savings estimates are crude, particularly because we do not have good data yet to define the distribution of duct leakage airflows in the large commercial building sector. Our estimates assume that the duct leakage that can be eliminated ranges from 10 to 20% of the nominal design supply airflow in each building and that the fan power increases associated with this duct leakage are 26% to 70% respectively (eliminating this duct leakage translates to fan power savings of 21 to 41%). The lower bound is based upon our measurements in the Sacramento test building; the upper bound is based upon predictions by Franconi et al. (1998). Dollar savings are based on an electricity price of \$0.10 per kWh. The representation of savings in terms of residential electricity consumption are based upon California residential electricity use projections for Year 2000 (Rohrer 2000), which indicate that the average California house used about 6,740 kWh that year.

Rufo and Coito (2002) assessed technical and economic potentials for 28 energy efficiency measures that could be implemented now in California's commercial buildings. Compared to their estimates of energy consumption and demand savings for the 28 measures (45 to 2,539 GWh, 0 to 769 MW), our energy consumption savings estimates for duct sealing rank somewhere between the 4^{th} and 8^{th} highest savings; our demand savings estimates rank between 6^{th} and 13^{th}.

Benefits to future buildings. The identification of a metric for characterizing distribution system performance allows us to recommend its inclusion in the next round of Title 24 as a way of characterizing the new building stock. Once we have a good database of new duct system characteristics, we can set reasonable targets for distribution system performance, which will ultimately lead to further energy savings in this sector.

Benefits to new buildings in the UC system. As an outgrowth of our work on this project, we have been working on duct specifications for the new University of California at Merced campus with members of the design team. In particular, we have reviewed the draft design documentation for their duct systems and recommended changes to the specifications. While we don't have a specific energy saving calculated for this work, we see the benefits extending to other UC campuses once others use these specifications.

Benefits in building operations and maintenance. A willing partner in this work has been Thomas Properties, a major manager of buildings, both public and private. Through this study, they have seen the benefits not only of improvements in duct diagnostics, but also in feedback on their HVAC system performance via the EMCS. By working closely with the private sector, we have seen the transfer of knowledge and the improvements in building operation in a major public facility.

Benefits to future engineers. A key aspect of our research team has been the inclusion of numerous students. While they may not have appreciated the long hours at the field site, they are now familiar with a number of important lessons about the performance of thermal distribution systems. As future engineers—not all of who may practice in California—we expect that some will become leaders in this area.

Glossary

ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
ASTM	American Society for Testing and Materials
APT	Automated Performance Testing
CEC	California Energy Commission
cfm	Cubic feet per minute
DOE	U.S. Department of Energy
EIA	Energy Information Administration
ELA	Effective Leakage Area
EMCS	Energy management control system
GWh	Gigawatt hours, 10^9 Wh, 10^6 kWh
HVAC	Heating, ventilating and air conditioning
IAQ	Indoor air quality
LBNL	Lawrence Berkeley National Laboratory
MASIS	Mobile aerosol-sealant injection system

MW	Megawatt, 10^6 W
PAC	Project Advisory Committee
PIER	Public Interest Energy Research
RD&D	Research, Development, and Demonstration
RMS	Root mean square
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association
TDS	Thermal distribution system
UC	University of California
VAV	Variable air volume

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Appendices

Appendix I. Metrics and Diagnostics

Appendix II. In-Situ Characterization

Appendix III. Duct Sealing Techniques

Appendix IV. Production Readiness Plan

Appendix I. Metrics and Diagnostics for Characterizing
Thermal Distribution Systems i**n Commercial Buildings**

Tengfang Xu, Mark Modera, Craig Wray, and Rick Diamond
Lawrence Berkeley National Laboratory
Berkeley, California

Proposed Metrics and Diagnostics for Characterizing The Performance of Commercial Thermal Distribution Systems

I.1 Overview

In practice, thermal distribution systems often do not perform as well as intended by the design process, partially because conventional building design processes do not adequately address distribution system performance, and partially because thermal distribution systems are generally not installed or operated according to design.

One common link between these two problems is the need for a set of metrics (or yardsticks) by which the performance of these systems can be characterized. Such metrics are needed both to simplify the inherent complexities associated with the design process, and to digest diagnostic measurements or monitoring results into a manageable number of descriptors. The idea behind metrics is to distill down the complexities of different systems to a single descriptor, or limited set of descriptors, that allows those systems to be compared with each other on a level playing field. One concrete example of a set of metrics is the way that automobiles are characterized and compared, using parameters such as wheelbase (size), horsepower (capacity), and miles-per-gallon (performance, or fuel-efficiency).

ASHRAE Standard 152P already provides energy performance metrics for residential thermal distribution systems. This Standard could serve a similar function for small commercial buildings that use small packaged HVAC systems similar to residential equipment. However, it does not address non-energy performance issues, nor can it be used to address energy efficiency issues in large commercial buildings.

In approaching the issue of metrics for thermal distribution systems in commercial buildings, a key distinction needs to be made between large and small commercial buildings. That distinction stems from two key differences between these two types of buildings, which are the:

1. Fraction of HVAC energy used by the fans and pumps, and
2. Complexity and variability of the systems.

For example, in large buildings, 35 to 50% of HVAC energy use occurs in the fans and pumps, versus approximately 15% in small commercial building systems. The larger fraction of fan and pump energy used by large buildings is largely due to longer distances, higher pressures, and continuous fan operation in the larger buildings.

The goal of this document is to provide a starting point for developing a set of metrics that describe thermal distribution system performance in both small (thermally dominated) and large (fan-power dominated) commercial buildings. This set of metrics needs to include energy (consumption and demand) metrics, as well as environmental indices (e.g., health, comfort, and safety). Because the two types of buildings operate very differently and tend to serve different purposes, we have assembled two distinct sets of metrics, with some overlap between the two sets. In addition, to keep the document scope manageable, most of the metrics and sub-metrics presented here apply only to air distribution systems.

I.2 Full-Sector Commercial Distribution Metrics

I.2.1 Energy Efficiency

At the most global level, the key energy metric for all types of thermal distribution systems is the *overall energy efficiency*, which is defined as:

Sum of the absolute values of all the space loads in the building Divided by the Sum of the absolute values of all heating and cooling energy delivered at heat exchangers for space conditioning plus all fan and pump energy consumption

This metric accounts for both thermal and transport efficiencies, accounts for simultaneous heating and cooling inefficiencies, includes energy added by terminal reheat, and can be used to compare distribution systems that use different transport media (e.g., air versus water). On the other hand, although it is conceptually useful, it is very difficult to calculate this metric for anything but a building on paper or in a computer. In a field situation, measuring the parameters required to calculate this metric generally is not practical.

There are a number of sub-metrics that contribute to the global energy efficiency. One that applies to both air and water systems is the *overall thermal efficiency*, which is the ratio of heating or cooling energy delivered at the supply grilles or radiators to the heating or cooling energy that heat exchangers deliver to the distribution system for space conditioning.

If we limit ourselves to air-only systems, there are a number of other sub-metrics that should apply to all commercial building thermal distribution systems, including:

1. The supply duct fractional air-leakage [%],
2. The normalized fan power [W/cfm, W/ft^2], and
3. The normalized airflow [cfm/ft^2].

I.2.2 Indoor Environmental Performance

It is important to recognize that distribution systems interact in a myriad of complex ways with other components of the building. This is true both with respect to comfort, and with respect to Indoor Air Quality (IAQ). Without even considering the entire building, it is clear that in most cases, the conditioned zones themselves are part of the distribution system, which alone adds enormous complexity. As an example, for an air system, the conditioned space can be seen as the part of the distribution system that connects the supply grilles to the return grilles. The thermal comfort within the space and the energy efficiency would be impacted by paths taken between those grilles, which in turn depend on the location and operating characteristics of the grilles, as well as on the air temperatures and flows through the grilles. Some concrete examples include the impacts of cold air distribution grilles on “dumping” cold air within the room, or the impacts of VAV induction airflows on the degree of mixing within the room. If we wanted to truly compare all types of distribution systems, we would have to address the issue of radiant versus air temperature, which is different for convective versus radiant delivery, and the location of the grilles or radiators relative to the windows. If we wanted to address the problem at this level, we would also need to address the impacts of localized distribution systems, such as user-controlled grilles at workstations. The time requirements needed to determine parameters associated with these complex metrics place practical limits on the scope of the indoor environmental (and even energy efficiency) metrics that should be considered. Based upon these arguments, we have attempted to limit the scope of our comfort and IAQ metrics to issues that can be isolated to the distribution system itself, and that do not involve more complex interactions with other environmental factors.

With respect to thermal comfort, we chose to limit the metrics to quantifying heating or cooling energy delivery at the distribution system entries to each conditioned space, and to ignore any further distribution issues within the spaces (e.g., to ignore how the air travels "down the road" after leaving supply grilles). Our global metric for thermal performance is the thermal uniformity between the spaces being conditioned; more specifically, the *zone-by-zone variance of the room temperatures*. This parameter applies to all systems, and includes the impacts from the weather and building operating conditions. This metric has the opposite problem as compared to the overall energy efficiency – it is more straightforward to determine in the field as compared to predicting the value from building plans or computer simulations.

With respect to IAQ, a thermal distribution system can impact it in a number of different ways, including the distribution system's impacts on:

1. The entry of pollutants,
2. The creation/incubation of pollutants, and
3. The transportation and dilution of pollutants.

Another environmental performance metric that can apply to all types of systems is the *noise* level, which is similar to thermal uniformity in that it is easier to measure in an actual building than to predict. This metric can be defined as the temporal average increase in noise level when the distribution is in normal operation, compared to when the system is not operating.

Because of the limited scope here, some of the metrics that we examined but chose not to utilize include: the Air Diffuser Performance Index (ADPI), which characterizes mixing external to a diffuser, and the set of comfort indices outlined in ASHRAE Standard 55, which are widely used in assessing comfort.

I.3 Small Commercial Air Distribution System Metrics

This section presents a proposed set of metrics to characterize the performance of thermal distribution systems in *small* commercial buildings (thermally dominated systems). This set of metrics should apply to most rooftop packaged units, as long as they do not have zoning or variable air volume controls.

I.3.1 Energy Performance

The key metrics for the energy performance of small commercial buildings come directly from ASHRAE Standard 152P. The measurement and calculation procedures in that proposed standard yield two efficiency metrics:

1. *Seasonal* distribution system efficiency for energy consumption calculations, and

2. *Design* efficiency for equipment sizing/selection and peak-demand calculations.

To determine these efficiencies, the draft ASHRAE standard contains procedures for obtaining a number of sub-metrics, including:

1. The supply and return duct fractional air-leakage,
2. The effective leakage area (ELA),
3. The duct operating pressures, and
4. The distribution effectiveness for thermal losses resulting from heat conduction through duct walls (the fraction of the sensible capacity lost due to conduction through the duct walls).

The key issues associated with using this standard for characterizing small commercial building energy performance are that the standard:

1. Does not address *continuous fan operation* and its impact on energy performance, and
2. Does not address *fan power* in any way.

Adding a simple methodology for calculating the impacts of fan energy use and the thermal efficiency impacts of continuous fan operation can augment the metrics in this standard. To accomplish this, a sub-metric that is required in addition to those in the ASHRAE Standard is the fraction of hours in a year that the fan is running without the air-conditioner or furnace operating. This parameter needs to be calculated separately for the heating season and the cooling season. The seasonal values can then be used to calculate the energy losses while the fan is off, which can then be combined with the equipment-on efficiencies on a fractional time basis. Incorporating this effect means that equipment capacity becomes an important characteristic. A more complete treatment would be needed if variable capacity equipment or a variable speed fan were accommodated.

I.3.2 Indoor Environmental Performance

As noted in Section I.2.2, the environmental-performance metrics described in this appendix are limited in scope. In the case of small commercial buildings, Indoor Environmental Performance can be characterized by six metrics and sub-metrics: four for thermal comfort, and two for IAQ/Safety.

For small commercial buildings, the two thermal comfort metrics are:

1. The ratio of delivered thermal capacity at the supply grilles to the thermal capacity delivered by the HVAC equipment into the distribution system, and
2. The standard deviation of the temperatures in the different spaces being conditioned by the system.

The ratio of delivered capacity to equipment capacity is a comfort measure in that it quantifies the impact of the distribution system on the HVAC system's capability to maintain comfort under design conditions. This metric can be calculated in the same manner as the duct-system energy efficiency under design conditions in ASHRAE Standard 152P.

The standard deviation of the temperatures in the different zones is a measure of how well the distribution system is distributing heating and cooling to the zones that need it. Because of the difficulty associated with quantifying the standard deviation of zone temperatures in the field or from building plans, we suggest that two sub-metrics be used:

1. The standard deviation of the *temperatures delivered at the supply grilles*, and
2. The standard deviation of the *ratio of delivered supply grille airflows to design supply grille airflows*.

The first of these sub-metrics is simply based upon the assumption that there are likely no purposeful means within the design of the system to alter the delivered temperatures on a zone-by-zone basis (e.g., reheat coils), assuming the definition of the types of systems/buildings covered by this section is appropriate. In other words, all rooms should be receiving the same temperature air.

On the other hand, floor-area normalized airflows vary in the design process between rooms, and therefore measured airflows should be compared to design airflows rather than to each other. The problem is that most buildings of this type do not receive zone-by-zone load calculations, or those calculations are impossible to obtain. In cases where no design information is available, the normalized airflow comparison has to be made relative to a nominal airflow per unit floor area (or the building average airflow per unit floor area).

The impact of the distribution system on IAQ in a small commercial building can be quantified in a limited manner based upon the following two sub-metrics:

1. The distribution system impact on conditioned-space and buffer-zone *pressures*, and
2. The distribution system impact on overall *air exchange rates*.

These metrics only address impacts of the distribution system on entry of pollutants and dilution of pollutants. The creation/incubation of pollutants is not addressed at all, and the other two impacts are not treated in an exhaustive manner.

The first sub-metric (*pressure* impacts) can be quantified in terms of two numbers:

1. The maximum *depressurization* of any building zone relative to the outdoors under any normal operating condition, and
2. The maximum *pressurization* of any building zone relative to the outdoors or other building zone under any normal operating condition.

The two values are separated because of their different implications, and because many commercial buildings are purposefully operated at pressures somewhat higher than outdoors.

Depressurization is generally the pressure imbalance that has the most negative impacts, including drawing in hot unconditioned outdoor air and outdoor pollutants (e.g., ozone) through the building shell in the summer, causing exterior doors to be difficult to open, and causing potential combustion-product backdrafting and spillage problems. Backdrafting and spillage are important IAQ concerns, as they cause combustion gases (pollutants) to be brought into occupied spaces. In addition, negative pressures in building spaces often cause pollutants to be carried from one zone to another (e.g., shop areas to office areas).

On the other hand, pressurization can cause doors to blow open, and forces hot humid air through the building shell in the winter, potentially causing moisture damage. In addition, if a zone that contains pollutants is pressurized relative to surrounding zones, the surrounding zones may become contaminated.

The second IAQ sub-metric that can be used for thermal distribution systems in small commercial buildings is their impact on overall *building air exchange rate*. This is the simplest means for describing the impact of these systems on pollutant dilution, as it addresses only their average pollutant dilution impact, or average outdoor-pollutant entry impact. Even this simple metric is complicated by the fact that most commercial buildings use an intentional outdoor air intake coupled to the return air to provide ventilation, and by the fact that unbalanced distribution system airflows do not add linearly to natural ventilation rates. As such, this metric also consists of two numbers, both normalized by conditioned-space volume:

1. The *balanced* leakage airflows through the supply and return ducts (i.e., the amount of leakage airflow out of the supply ducts that could be offset by the amount of leakage airflow into return ducts, or vice versa), and
2. The *unbalanced* leakage airflows through the supply and return ducts (i.e., the leakage airflow differential between the supply and return ducts).

I.4 Large Commercial Air Distribution System Metrics

This section presents a proposed set of metrics to characterize the performance of thermal distribution systems in *large* commercial buildings (fan-dominated systems). This set of metrics should be applied to packaged and built-up systems, whether or not they have Constant Air Volume (CAV) or Variable Air Volume (VAV). The key differences between the metrics for large and small commercial buildings are that the ones for large commercial buildings focus on transport energy use, and a more detailed treatment of zonal performance.

I.4.1 Energy Performance

The working assumption for large commercial buildings is that the distribution systems are generally located within the conditioned space. The only exceptions to this assumption are when the ducts are located outdoors, or when the ducts are located in a top-story plenum that is insulated at the ceiling or vented at the roof deck. If a duct system (or a portion of a duct system) meets either of these conditions, its energy efficiency can be calculated with the same ASHRAE 152P methodology used for small commercial buildings, except that the fan power efficiency should be calculated as outlined in this section.

For large commercial buildings, the primary metric for distribution system performance is the transport energy (e.g., total energy used to transport air) per unit thermal energy delivered ($kW_{transport}$ / $kW_{thermal\text{-}delivery}$). This metric obviously depends on the distance over which thermal energy needs to be transported, and therefore is a function of the size and geometry of the building. On the other hand, by using this as the primary metric, the use of distributed heating and cooling equipment can be compared with central systems. This metric also allows for comparisons of VAV and CAV systems, as well as the impacts of thermal/leakage losses on fan power. Calculating this parameter for a building on paper or using a computer is relatively straightforward, excluding any impacts of improper installation or operation. Determining this parameter in a building that has already been built is somewhat more difficult because measuring the total heating or cooling energy delivered to each zone requires air temperature and flow sensors at every grille and temporally continuous measurements.

It should be noted that this definition implicitly assumes that thermal energy, including heat generated by transport, is only transferred to the conditioned zones through the supply grilles (or radiators for a hydronic system). This assumption would be violated by exposed ductwork within the conditioned space. Similarly, this parameter does not properly account for reductions or increases in cooling or heating loads created by heat exchange with buffer spaces. These two limitations are not a problem for a building in the design process, as the thermal delivery to the zones is by definition the load in those zones. On the other hand, measurements of this parameter in a real building would need to be corrected for these two effects.

There are several common sub-metrics that can be used to characterize fan power in large commercial buildings, including the following three:

1. The specific fan power [W_{fan}/cfm],
2. Fan airflow density [cfm/ft^2], and
3. Normalized fan power [W_{fan}/ft^2].

The limitations and precise definitions of these parameters merit further discussion.

The airflow (cfm) used for *specific fan power* [W_{fan}/cfm] can be defined in terms of the airflow delivered by the fan, or in terms of the airflow delivered to the conditioned spaces. Using the first definition is simply a measure of the efficiency of the fan airflow, which is determined by the airflow resistance of the duct system (leaks in the duct system serve to reduce its resistance). The existence of duct leaks therefore would produce a bias that reduces the value of the specific fan power. The second definition (using the airflow delivered to the zones) incorporates impacts of duct leakage, because leak reduction is excluded in the delivered airflows, thereby increasing the specific fan power. We suggest using the second definition. However, it needs to be made clear that neither definition specifically accounts for the impacts of thermal losses from the ducts (these losses increase the amount of air that needs to be moved, which affects the specific fan power by changing the operating point of the fan).

The *fan airflow density* [cfm/ft^2] is another fairly common metric for describing HVAC systems, however it also needs some clarification. As with specific fan power, the airflow at the fan is not necessarily equal to the sum of the grille airflows. If we use fan airflow in our definition, it implicitly penalizes duct leakage, in that more air needs to be moved to satisfy the load. In contrast, using the airflow at the grilles ignores the impact of duct leakage. To avoid double-counting the impacts of leakage, we define fan airflow density in terms of airflow at the grilles. Unlike specific fan power, both the airflow at the fan and the airflow at the grilles are affected by thermal conduction losses through the duct walls, because thermal losses translate into the need to move more air at both places. By defining this parameter based upon airflow at the grilles, we have a parameter that is not impacted in any significant way by duct leakage, but that does reflect thermal conduction losses.

The *normalized fan power* [W_{fan}/ft^2] is the most comprehensive and general of these common sub-metrics, because it includes the impact of both leakage and conduction losses, and could apply equally well to air or hydronic distribution systems.

An important point that has generally been ignored in conventional analyses of the energy impacts of duct leakage and conduction losses is their impact on fan power in large commercial buildings. In particular, the issue is that thermal or leakage losses reduce the quantity of energy that is delivered to the zones being conditioned. Thus, to meet the loads in these zones, more energy needs to be transported by the fan to make up for the energy that is lost along the way. As more energy transport is usually accomplished by moving more air, this also translates into larger pressure drops through the ductwork. Thus, as fan power scales with the product of the airflow and the pressure differential, fan power increases dramatically as a result of thermal or leakage losses from the supply ducts[1]. Note that this is true even when the losses are all within the envelope of the building.

One way to conceptualize the fan power impacts of duct losses is to think of leakage and conduction losses as short-circuiting the fan. If the supply ducts are in a ceiling-plenum return, then some of the losses from the supply ducts to that plenum are returned to the supply fan via the return air. Some of these losses are also delivered to the conditioned spaces through conduction across the ceiling tiles, although this is not always beneficial when there are simultaneous heating and cooling loads in different spaces.

The losses from the ducts can be thought of as being partitioned by a "current-splitter", with the larger fraction of the losses going down the path of least resistance. The two competing paths are back to the fan via airflow through the return plenum, and to the conditioned spaces via conduction through the ceiling plenum tiles. The effective conductance of the return air path is its airflow times the specific heat of the air, while the effective conductance of the ceiling tiles is their area divided by the R-value of the tiles. In general, the ratios of these two conductances imply that most of the energy is sent back to the fan.

The sub-metric that can be used to quantify this effect of duct losses on fan power is called the *duct-loss power ratio*, defined as the ratio of the fan power for an airtight, perfectly insulated duct system, to the fan power for the duct system in question. This factor depends upon the leakage and conduction losses from the duct system, and on the ratio of the effective conductance of energy back to the central system to the effective conductance of energy into conditioned spaces. It should be noted that this parameter does not account for the fact that conduction to conditioned spaces is not always beneficial.

One important factor that has to be addressed when calculating the duct-loss power ratio is the impact of VAV boxes, in particular the impact of plenum air induction at these boxes. The issue is that the induction boxes represent a third path for the energy losses from supply ducts to leave the ceiling plenum. They draw in air that was otherwise going to be drawn back to the central fan; they then blow that air into the conditioned zones. In this respect, the induction airflows represent another conductive path between the return plenum and the conditioned spaces. Thus, when calculating the duct-loss power ratio, the product of the induction airflows (less any induction air leakage back into the plenum downstream of the induction fan) and the specific heat of air needs to be added to the conductance of the ceiling tiles.[2]

Some additional sub-metrics that can be used for characterizing energy performance in large commercial buildings are similar to those for small commercial buildings:

1. The supply and return duct fractional leakage airflows,
2. The effective duct leakage area (ELA),

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[1] For example, if we assume that the airflow resistance of a typical duct system results in the pressure differential across the supply fan being roughly proportional to the airflow squared, then a 10% increase in supply airflow causes a 33% increase in supply fan power; a 20% airflow increase causes a 73% power increase. Note that this is an overly simplistic way to assess airflow impacts on fan power, but serves as a rough first order approximation.

[2] To complicate the issue, some of the induction air can leak back into the ceiling plenum downstream of the induction fan if the supply ducts transporting the induction air to the conditioned spaces are leaky.

3. The duct operating pressures, and
4. The distribution effectiveness for thermal losses resulting from heat conduction through duct walls (the fraction of the sensible capacity lost due to conduction through the duct walls).

The difference in large commercial buildings is that these parameters need to be calculated for sections of the ductwork, and not simply for the entire system. For example, the duct leakage needs to be split between before the VAV boxes and after the VAV boxes, as does the distribution effectiveness and the operating pressures.

Most small commercial buildings and CAV systems in large buildings have single operating points, whereas VAV systems have a spectrum of operating points. The operating point impacts the operating pressures and the distribution effectiveness. These parameters should be calculated/measured for VAV systems at design conditions, as well as at a part-load point that is somehow related to the seasonal average value. The seasonal operating condition is not so difficult to quantify for a building on paper, but is likely to be problematic in measurement situations. Determining this condition in a building requires measurements over a reasonable time period to look for trends. For VAV boxes with heating coils, the energy gain/loss induced by the heating coils needs to be included in all distribution effectiveness calculations.

I.4.2 Indoor Environmental Performance

There are several metrics currently in use to describe comfort performance in large commercial buildings, some of which are even focused on the distribution system. One in particular is related to the distribution from the heating/cooling equipment to the supply grilles, rather than distribution within the zones after the airflow leaves the grilles. This metric specifically excludes the metrics that treat room air mixing or temperature mixing, or that try to address zone velocity, humidity, or radiation impacts on comfort. Although our definition of the distribution system ends at the entry points to the zone, the spatial temperature variation of supply grille air temperatures is still a useful surrogate metric for the capability of the distribution system to affect comfort. For each zone, this variation can be represented by the standard deviation of the temporally coincident air temperatures at each supply grille serving that zone. From zone to zone, the spatial temperature variation of supply grille air temperatures can be represented by the standard deviation of the temporally coincident zonal average supply grille air temperatures.

The key difference between the IAQ metrics used in large commercial buildings and those described for smaller buildings is that large commercial buildings usually cannot be treated as single zones; they have different zones with different load requirements, and the dilution of pollutants is affected by local air change rates (e.g., in a room) and transfer air change rates between zones, as opposed to the overall air change rate of the building. In a manner similar to using the standard deviation of normalized supply grille airflow for small systems, one can adopt basic statistics (i.e., max, min, mean, and standard deviation) of supply airflow to characterize the mechanical ventilation impact for large systems under certain operating conditions. In this case, supply airflow means the sum of all supply grille airflows within the same zone, normalized by occupancy or zone floor area. The other key issue with large commercial building metrics is the need to define typical operating conditions for systems that are often very complex (e.g., temporal variation in VAV systems, additional airflow through induction units).

In assessing the performance of an air distribution system and its impacts on comfort and air quality, it is almost impossible to obtain quantitative measurements, but it is relatively easy to obtain qualitative information about a system's design, operation, maintenance, and control. Consequently, a checklist is needed. Some of the characteristics to record include:

- Weather Conditions (temperature, wind speed, humidity)
- Faulty or closed outdoor air dampers
- Failed/damaged ventilation or exhaust fans
- Dirty ducts and/or filters
- VAV terminal box dampers that close off completely
- Failed fire dampers that might close off airflow
- Lack of a purge cycle at system startup
- Faulty fan-tracking control on air handling system
- Faulty building pressure set point (e.g., negative building pressure)

I.5 Diagnostics for Evaluating Thermal Distribution System Metrics

I.5.1 One-Time and Short-Term Diagnostic Measurements for Energy Performance

I.5.1.1 Duct Leakage: Duct leakage characterization includes determining the supply duct air-leakage ratio, which can be performed using two independent methods:

1. Subtracting the sum of supply grille airflows from the measured supply fan airflow, and
2. Calculating leakage airflow from measured duct effective leakage area (ELA) and duct operating pressures. The difference between characterizing large and small systems is that we separate the evaluation of operating pressure and ELA based on main-duct and branch-duct levels, respectively.

   For CAV systems, the diagnostics involve a one-time measurement under fixed operating conditions. For VAV systems, the characterization of air-leakage ratio is limited to one particular operation condition that is likely to be fixed during airflow measurements. However, for the second diagnostic, the operating pressures and supply airflows need to be monitored continuously over a range of normal operating conditions (e.g., days, and perhaps seasonally depending on how induction fans operate).

I.5.1.2 Distribution Effectiveness: Distribution effectiveness (*temperature effectiveness*), which quantifies thermal conduction losses, can be determined by measuring the temperature drop/rise along supply ducts (e.g., between air-handler coils and VAV box inlets, and between VAV boxes and supply grilles). For CAV systems, temperatures and airflows can be measured under steady-state operation, with and without reheat operation. For VAV systems, temperatures and airflows need to be measured over a range of normal operating conditions to account for variable airflow, reheat coil operation, and induction airflows.

I.5.1.3 Duct-Loss Power Ratio: Diagnostics for this metric require further development before they can be practically applied.

I.5.1.4 Equipment Performance: Some relevant characteristics in small commercial buildings are the capacities of the cooling and heating equipment and the fan, including total fan airflow (cfm). In addition to measuring airflows through the equipment, determining the cooling and heating capacities requires measurements of temperature and humidity upstream and downstream of the equipment, so that enthalpy changes can be calculated.

I.5.1.5 Fan Operation: This metric is used for small CAV systems. One can determine the fan's on-time by recording when the space conditioning unit is on and off and then calculate the ratios of fan operation hours when the unit is off to total fan operation hours during the cooling and heating seasons, respectively. Although short-term data are useful, data from a whole-year of monitoring can provide additional important information on energy saving implications due to seasonal effects.

I.5.1.6 Cycle-Average Distribution Efficiency: This metric is used for small CAV systems. It can be determined on a minute-by-minute basis throughout compressor and furnace cycles by short term monitoring of air temperatures and flows.

I.5.1.7 Transport Energy Ratio ($kW_{transport}/kW_{thermal\text{-}delivery}$): This metric is the total energy used to transport air ($kW_{transport}$) per unit thermal energy delivered ($kW_{thermal\text{-}delivery}$), and is only used for distribution-system performance in large commercial buildings. Determining this parameter in a building that has already been built tends to be impractical because measuring the total heating or cooling energy delivered to each zone requires air temperature and flow sensors at every grille and temporally continuous measurements.

I.5.1.8 Specific Fan Power (W_{fan}/cfm): This metric does not account for the thermal losses along the ductwork. One of the required measurements is the fan power, either measured one-time (CAV) or through short-term monitoring (VAV). The cfm is the total delivered airflow, which is obtained by measuring the supply grille airflows under a typical operating condition (heating, cooling, or mechanical ventilation mode). This parameter is relatively constant for CAV systems, changing only as ductwork gets dirty or begins to leak, or as fans wear out or become dirty. For VAV systems, this parameter changes as the system operating point changes. In addition, for VAV systems with induction units, it is practically impossible to separate the airflow delivered by different components. In general, determining this metric is impractical for VAV systems with or without induction units, unless for diagnostic purposes the system is placed in a fixed operating mode and the induction fans are turned off.

I.5.1.9 Fan-Airflow Density (cfm/ft^2): The airflow per unit floor area in each zone can be determined by measuring the conditioned floor area and the total airflow delivered to each zone under certain operating conditions. As described in Section I.4.1, this metric reflects the impacts of thermal conduction losses while not discounting duct leaks in a significant way. Like specific fan power, the metric is impractical for VAV systems, unless for diagnostic purposes the system is placed in a fixed operating mode and the induction fans are turned off.

I.5.1.10 Normalized Fan Energy (W_{fan}/ft^2): As described in Section I.4.1, this metric reflects the combined impacts of duct leaks and thermal conduction losses, and can be examined for large commercial systems, especially CAV systems, by measuring fan power and the conditioned floor area. A one-time measurement of power is needed for a CAV system, whereas short-term monitoring over a range of normal operating conditions is needed for a VAV system.

I.5.2 One-Time and Short-Term Diagnostic Measurements for Environmental Quality Performance

I.5.2.1 Duct Loss Location: Inspecting ducts involves qualitative (e.g., visual) and quantitative measurements. This includes inspecting the ceiling plenum to determine the location of insulation and venting, and conducting pressure and temperature measurements to determine the location of the air and thermal boundaries. In addition, for large systems located in multi-story buildings, it is necessary to examine the ceiling plenum for the top floor in this regard. These diagnostics can be conducted on a one-time basis.

I.5.2.2 Duct-System Airflow Resistance: Determining airflow resistance involves measuring airflows and operating pressures in supply and return ducts, including at a minimum the pressures in the equipment plenums and at the supply grilles. For CAV systems, this can be a one-time measurement in a desired mode (e.g., heating, cooling, and/or mechanical ventilation). For VAV systems, continuous monitoring is required over a range of normal operating conditions (e.g., several days).

I.5.2.3 Balanced and Unbalanced Leaking Airflow: This applies to small systems only. It involves measuring the leakage airflows for the supply and return ducts.

I.5.2.4 Zonal Pressure Distribution: The zonal pressure distribution can be measured to the nearest 0.1 Pascal under different operating conditions, and can include measurements of both occupied and buffer zones, with reference to outdoor air or to an adjacent zone. Zone pressurization and depressurization data when coupled with zonal leakage area measurements can provide useful information about the potential for unintended airflows that might affect comfort and air quality.

I.5.2.5 Comfort Capability Index: As described in Section I.3.2, this metric is the ratio of delivered thermal capacity at the grilles to the thermal capacity delivered by the HVAC equipment. It can be determined in the same manner as the duct-system energy efficiency under design conditions in ASHRAE Standard 152P.

I.5.2.6 Spatial Temperature Variation: The spatial distribution of air temperatures can be determined by measuring the temperature of air exiting each supply grille. This metric includes a combination of standard statistical parameters (i.e., maximum, minimum, mean, and standard deviation), which can be calculated using the measured temperature data. For small systems, the statistical parameters are based on all the supply grilles; for large systems, the metrics should be presented at two levels: one within each zone; the other between zones based on the mean air temperature at all supply grilles for a zone. Note that measuring the supply air temperature at every grille in a large commercial building is impractical (particularly for a building with a VAV system); in this case, sampling techniques need to be used to obtain representative information.

I.5.2.7 Spatial Airflow Variation: The spatial distribution of supply airflow can be determined by measuring airflows at all supply grilles normalized by the served floor area (cfm/ft^2) or occupancy (cfm/person), and compared to the associated design supply airflows (cfm/ft^2, or cfm/person). The relevant metric for a single zone is the standard deviation of the ratios of actual airflows to the design airflows. For CAV systems, this involves a one-time measurement. In general, determining this metric is impractical for VAV systems with or without induction units, unless for diagnostic purposes the system is placed in a fixed operating mode and the induction fans are turned off.

I.5.2.8 Other Qualitative Measurements to Be Recorded: See the **c**hecklist described in Section I.4.2.

APPENDIX II. IN-SITU CHARACTERIZATION OF THE THERMAL DISTRIBUTION SYSTEM IN A LARGE COMMERCIAL BUILDING

Craig Wray, Darryl Dickerhoff, Nance Matson, Duo Wang, and Rick Diamond
Lawrence Berkeley National Laboratory
Berkeley, California

In-Situ Characterization of the Thermal Distribution System in a Large Commercial Building

Summary

This appendix describes our recent in-situ characterization activities in a 25 story office building in Sacramento, California; we carried out this work as part of the PIER “Thermal Distribution Systems in Commercial Buildings” project. Our review of the literature and our previous research has found very little characterization of the actual performance of thermal distribution systems in large commercial buildings. Our initial duct leakage assessment using conventional industry methods indicated that the duct system was leaky, but this was only part of the story. By conducting more sophisticated duct leakage airflow diagnostics, we determined that the actual airflow through these leaks was small (about 5% of air handler flow at operating conditions), and consequently less significant. Visual inspection of our test building showed every indication of a “tight” thermal distribution system: good application of mastic, metal bands at joints, and overall high quality installation. In order to demonstrate the impact of duct leakage, we introduced measured leaks and monitored the impact of these controlled leaks to determine their impact on the HVAC system energy consumption.

The principal outcome from this project is that the energy impact of duct leakage in large commercial buildings can be substantial. By adding 15% duct leakage to make a total of 20% at operating conditions, we found that the added leakage leads to an increase in air-handler supply fan power of up to 37%, and an overall increase in total fan power (air-handler supply fans plus induction fans) of up to 26%. The total fan power increase is lower because the added duct leakage causes induction fans to operate less often.

Previous simulations (Franconi et al. 1998) have suggested that the energy impacts of 20% duct leakage are larger: on the order of 60% to 70% of the total fan energy consumption. However, the duct leakage fraction in the simulations is normalized by nominal design supply airflow rather than by operating supply airflow. Redefining our duct leakage fraction to match the definition used in the simulations means that the duct leakage that we added was about 10% of the nominal design supply airflow. Given that our leakage fractions were about 50% of those used in the simulations, and assuming that fan power is somewhere between a quadratic and cubic function of airflow, our measurements are consistent with the simulation results.

The remainder of this appendix provides a general description of the test building and its thermal distribution system; a description of our monitoring and diagnostic activities; a summary of our preliminary duct leakage findings; a description of the HVAC system airflow diagnostics that we carried out prior to and during our duct leakage intervention tests; and a summary of our HVAC fan power measurements, and other findings.

Building and Thermal Distribution System – General Description

For our large commercial building characterization and duct leakage intervention study, we selected an office building in Sacramento, California. Eley Associates is also studying this building for a different PIER project (New Buildings Institute, Element 3: Integrated Design of Large Commercial HVAC Systems), so it offers the potential for collaboration between our efforts. The building, first occupied in April 2001, has 25 stories and a total floor area of 955,000 ft^2. Our study focused on two floors with similar occupancy and use (each approximately 29,000 ft^2 in floor area). We extensively characterized the HVAC system operation on the intervention (17^{th}) floor before installing duct leaks on this floor; we used the 16^{th} floor as a control floor (i.e., with no changes to the HVAC system as a basis for comparison to the intervention floor).

Each floor has four separate air-handlers, with two nominal 15,000 cfm, 15 hp supply air-handlers per floor and two nominal 10,000 cfm, 5 hp relief air-handlers per floor. Each pair of supply and relief air-handlers is located in a separate mechanical room at the northeast and northwest corners of each floor, and each air-handler uses an EMCS-controlled variable-frequency-drive. Each supply air-handler is a draw-through packaged unit equipped with an air mixing chamber, a filter section, a hot-water air preheat coil, a chilled-water air cooling coil, and a backward-curved plug fan. Each relief air-handler uses a backward-curved tube-axial centrifugal fan. A central plant with boilers and chillers supplies the appropriate air-handler coils with cold and hot water.

Together, the two supply air-handlers on each floor serve a single-duct VAV system supply loop that in turn serves 34 VAV boxes on the intervention floor and 38 boxes on the control floor (see Figure II-1). The difference between the number of zones on the two floors is due to slight changes in room configuration, and does not affect our findings. A single duct-static-pressure-sensor in each loop is located at the farthest point from the air-handlers. The 13 perimeter VAV boxes on the intervention floor and the 14 perimeter boxes on the control floor have discharge electric reheat coils (750 to 2,500 W, staged) and are parallel-fan-powered (1/6 and 1/4 hp induction fans), with the fans drawing their induction air from the ceiling plenum return through a pleated filter and discharging into the primary air section of the box through an adjustable fixed-stop gravity backdraft damper. The core VAV boxes have no reheat and no induction fans. Each VAV box inlet has a flow grid located immediately upstream of its EMCS-controlled primary air damper.

In total, the VAV boxes on the intervention floor serve 103 supply grilles, each with a manual volume damper located near the branch takeoff. Most supply grilles use 2’ x 2’ perforated-face grilles and discharge in multiple directions; exceptions are the wall grilles in the two electrical rooms, a discharge with no grille in the communications equipment room, and the linear slot diffusers in the two main elevator lobbies. The 2’ x 2’ grilles sit in the ceiling between T-bar sections, with a small gap between the grille edges and the T-bar sections.

With the exception of the elevator lobbies (portions of the slot diffusers also serve as return grilles), ceiling returns are 2’ x 2’ perforated-face grilles. The mechanical rooms are each connected to the ceiling space through a short return transfer duct, and serve as a large plenum from which the supply air-handler draws its return air through EMCS-controlled return dampers.

Outdoor air is ducted to each supply air-handler mixing box from a wall louver and through two parallel EMCS-controlled dampers: a minimum outdoor air damper and a larger economizer damper. Return air is exhausted directly from each mechanical room to outdoors by the relief air-handler, as needed to control indoor-outdoor pressure difference for the floor. The indoor pressure appears to be referenced to the outdoor pressure at the building roof.

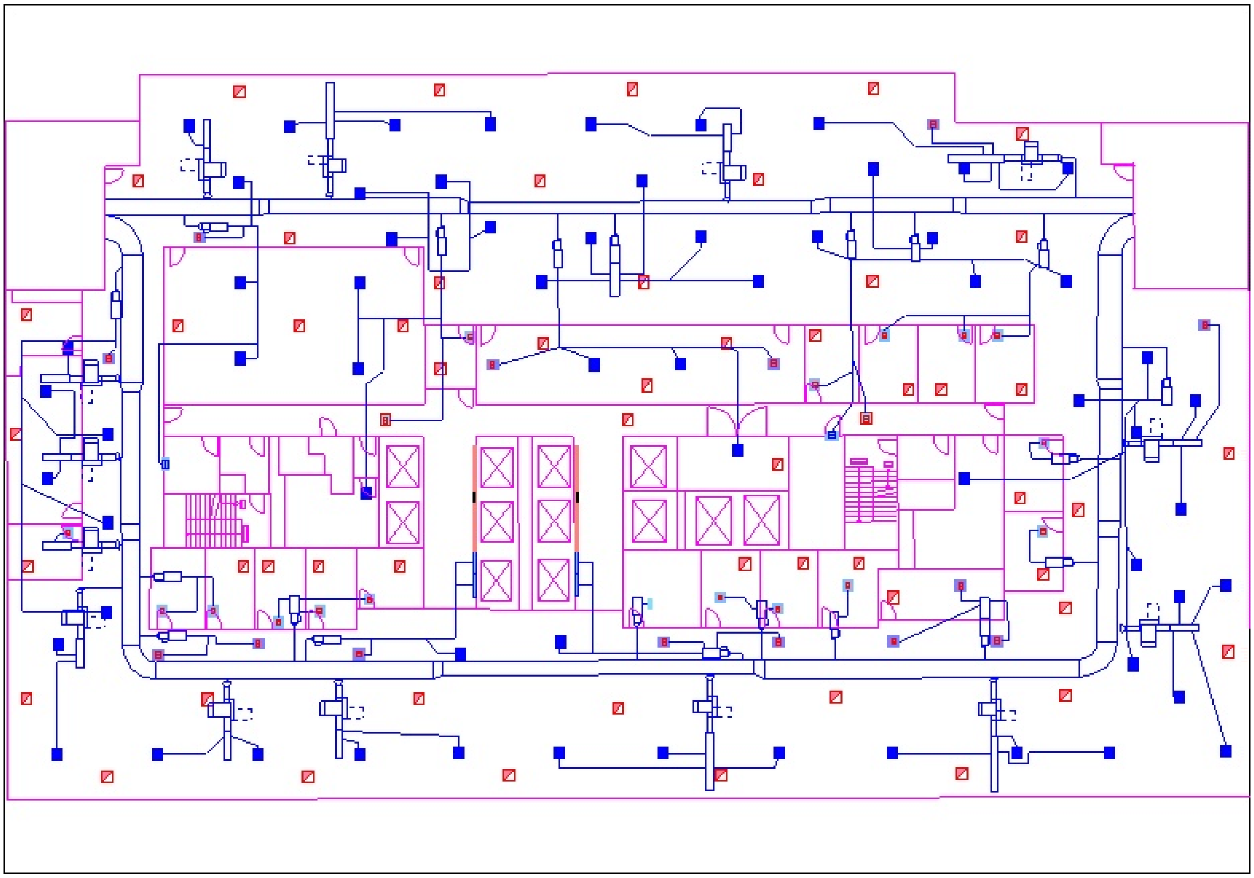

Figure II-1. Duct layout for intervention floor. The control floor's duct layout is similar.

All ducts are constructed from sheet metal, except for the flexible duct connections to the supply grilles. All joints are covered externally with mastic. We also found mastic covering the inside of branch duct connections to the VAV box plenums. Except in the mechanical rooms and in the ceiling space near those rooms where there is internal insulation, external insulation with a vapor barrier covers the ducts.

HVAC System Operation - Cooling Mode: The building operated in cooling mode during our duct leakage intervention study. In this mode, the EMCS puts each floor's air handling system through a short smoke control system check between about 2:00 a.m. and 2:30 a.m.

If pre-cooling is not needed, the HVAC systems are put into occupied mode around 4:00 a.m. and the systems run to maintain zone temperature conditions until 6:00 p.m. If pre-cooling is needed (dictated by building and outdoor temperatures measured at midnight), the corresponding HVAC system is put into economizer mode (outdoor air dampers fully open) and the air-handler supply fans are operated for pre-cooling. The relief fans run as needed to maintain building pressures.

During the occupied mode, the discharge duct temperature measured at the outlet of the air-handlers is used to control the heating and cooling valves serving the coils upstream of the air handler supply fan. The temperatures measured by the EMCS zone thermostats are used to determine the corresponding amount of primary air needed from the main loop through the VAV box to each zone. The primary air damper is adjusted accordingly. For the powered boxes, the induction fans are energized if the primary airflow is less than 40% of the damper's throttling range maximum set point.

Building Monitoring

We extensively monitored the intervention and control floors to characterize HVAC system operation and to determine the impact of duct leakage on fan energy consumption. The monitoring occurred over the period from November 2001 to early August 2002. Much of the monitoring in advance of our duct leakage intervention tests during the summer of 2002 was used to support our characterization diagnostic tests, but was also useful to troubleshoot the operation of our monitoring equipment and to validate the data being collected using the building's Energy Management Control System (EMCS). Our monitoring using the EMCS involved recording data for 232 measurement points. In addition, we installed 96 temperature, relative humidity, pressure, and power monitoring points. Table II-1 summarizes these 328 points.

Table II-1. Monitoring point summary.

	EMCS Monitoring	*LBNL Installed Monitoring*
Supply Fans		Electricity consumption, pressure, airflow
Relief Fans		Electricity consumption
Outdoor Air Supply	Minimum outdoor airflow; economizer damper position	Air temperature; relative humidity
Return Air	Damper position; air temperature; relative humidity	Airflow, air temperature; relative humidity
Air Handler Cabinet	Supply air temperature (after fan)	Supply air temperature and relative humidity (after supply fan and before heating coil); air temperature between cooling coil and supply fan
Zones (All)	Zone air temperature, primary airflow	
Zones (All with Induction Fans and Heaters)	Induction fan status (off / on); box reheat status (off / 1st stage / 2nd stage)	
Zones (Detailed)		Air temperature and static pressure before and after VAV box; total pressure after box; air temperature in VAV box before heater (where applicable); air temperature at inlet to induction fan (where applicable); supply air temperature at the farthest grille
Zones (Partial Detail)		Supply air temperature at the farthest grille
Outdoor Conditions	Air temperature; relative humidity	
Miscellaneous Temperatures and Pressures	Static pressure in supply loop (one location per floor); indoor-outdoor static pressure difference	Static pressure in far supply loop corners; ceiling plenum air temperature (two locations, intervention floor)

Zones: On the zone level, the building's EMCS was set to record zone temperatures and all VAV box primary airflows on the control and intervention floors. For the VAV boxes with induction fans, the EMCS also recorded induction fan status (on / off) and box heater status (off / stage 1 / stage 2). We measured the induction fan power as a function of VAV box primary airflow reported by the EMCS. The EMCS primary airflow and fan status data were then used to calculate induction fan energy over the test period. We also measured the heater power for each powered VAV box.

We monitored four VAV boxes on the intervention floor (North and South locations, two with induction fans and two without) in detail using Onset Hobo Pro temperature sensors and two Energy Conservatory 8-channel Automatic Performance Testing (APT) systems. Temperatures were measured before the VAV box, inside the box before the heater (where applicable), after the VAV box, and at the farthest supply grille. The APTs were used to measure and record duct static pressures before and after the VAV boxes, as well as downstream duct total pressure.

We also installed Onset Hobo Pro temperature sensors at the farthest supply grille for ten other VAV boxes: four on the intervention floor, representing East and West orientations, with and without induction fans and heaters; and eight on the control floor, two on each of the four orientations, with and without induction fans and heaters.

Mechanical Rooms: Onset Hobo Pro temperature/relative humidity (RH) sensors were installed to monitor thermal conditions in the outdoor air intakes, at the return dampers (for calibrating EMCS data), before the heating coils, immediately after the cooling coils (before the system fans), and after the system fans. These thermal data were primarily collected to support our airflow diagnostics and monitoring, but are also useful for future evaluations of floor-by-floor heating and cooling coil demand imposed on the central plant. Pressure transducers (Setra 264, Modus T10, and Validyne DP103) were used to translate the supply grid and return grid pressures and the supply and relief fan static pressures to millivolt signals for the Datataker 50 data logger. Ohio Semitronics WL45R electric power transducers were installed and connected to the Datataker 50 to monitor supply and relief fan electricity usage. Each electric power transducer produces a pulse output that the data logger records on a minute by minute basis.

Outdoor Conditions: The EMCS recorded outdoor temperature and relative humidity, as well as calculated enthalpy, during the test period. We also installed two Onset Hobo Pro temperature/RH sensors in the outdoor air plenums on the intervention floors so that we could later compare the EMCS data to our measurements.

Additional Data Collection for Others: Following our monitoring period, Eley Associates and Taylor Engineering plan to conduct part of their PIER research project in the same building. To better facilitate their data needs, we installed additional measuring equipment in the intervention and control floor mechanical rooms: pressure (static across the supply and relief fans) and relief fan airflow. We also added zone damper positions to the points being monitored by the EMCS.

Data Synchronization: We used the duct static pressure signal at the system test time (~2:30 am) to synchronize the EMCS data to the data logger supply fan power and airflow data. The EMCS data were recorded at one, two, or five minute intervals, depending on the point being monitored. We interpolated to estimate missing values. For longer time periods (up to 60 observations once or twice a week during EMCS data download by building staff), we interpolated based on two values: the average of the observations in the hour before the missing data and the average of the observations in the hour after the missing data. This approach did not change the results significantly. We used the factory calibration for the power transducers and calibrated the airflow measuring equipment as discussed in the “Characterization of HVAC

System Airflows" section, which follows. The outdoor air temperatures and zone temperatures discussed in this report are as reported by the EMCS system.

Diagnostic Tests: Besides downloading and retrieving monitoring data, our main efforts during our approximately 30 field visits to the building included a number of diagnostic tests to characterize the system and study operational modes. These tests included:

Tracer gas measurements:

- Air-handler supply fans
- VAV box primary airflow grids

Pressure Diagnostics:

- Carried out static pressure traverses of air-handler supply fan plenum to determine the optimal location for static pressure taps used to measure static pressure rises across the supply fan (for Eley/Taylor project).
- Determined main loop static pressure profiles at different operating conditions to assess uniformity of pressure and to identify limits for VAV box opening that allow HVAC system to maintain loop set point static pressure.
- Measured pressure differences across doors to corridors, bathrooms, electrical rooms to validate airflow diagnostic anomalies.
- Measured static pressures in VAV boxes and at installed downstream leakage sites as a function of the EMCS-reported VAV box primary airflow.
- Measured operating pressures at installed downstream and upstream leakage sites to support leakage airflow calculations.

Flow Hood Measurements:

- Used powered flow hoods to determine the total airflow exiting the supply grilles (sum of supply grille airflows).
- Used powered and commercially available flow hoods on a sample of grilles to test flow hood repeatability and to assess hood usability and accuracy for a rapid duct leakage screening method.

Leakage Diagnostics:

- Carried out component and total duct leakage tests at all detail and partial detail VAV boxes on the intervention floor. This included determining component leakage areas of the VAV box, induction fan damper, ducts, and grilles.
- Measured duct leakage upstream of the VAV boxes by fully closing all VAV primary air dampers and using a calibrated fan and flowmeter at the air-handler supply fan plenum.
- Conducted HVAC system and VAV box inspections and measurements to develop plans to add calibrated leakage and to demonstrate leakage sealing using the multiple compact aerosol injector.
- Tested effectiveness of backer rod insulation to seal leaks at supply grille edges.

Diagnostics Using the EMCS:

- Set the individual VAV box primary airflow set point to zero: record the primary airflow that the EMCS reports and visually determine the actual position of all primary air dampers.
- Measured induction fan and heater power as a function of VAV box primary airflows (Base leakage case and downstream leakage case) (using Elite Pro power monitor).
- Carried out VAV box airflow capacity tests to determine the range of primary airflow set points (% of cooling maximum set points) that correspond to maintaining 1" w.c. in the main loop.

Diagnostics for Others:

- Modulated outdoor air / return air dampers for airflow tests. (Federspiel Controls)
- Maintained constant duct pressure for duct traverses. (Eley Associates / Taylor Engineering)
- Taught Taylor Engineering how to download monitoring data for their PIER project.

System Layout & Equipment Inspections:

- Verified and noted changes to duct layout as shown in plans.
- Developed detailed duct maps of four VAV boxes that we studied in detail and four partial zones on the intervention floor (lengths, sizes, location).
- Inspected equipment to verify measurement results.

Miscellaneous Calibrations:

- Calibrated pressure transducers.
- Installed physical filters on relief fan static pressure taps.
- Calibrated EMCS thermostat data with Onset Temperature Hobos (data collected to be analyzed as part of a future thermal analysis project).
- Calibrated analog airflow gauges on air-handler supply fans in support of a rapid duct leakage diagnostic.
- Used an aspirated temperature sensor to estimate the radiant component of temperature sensors downstream of powered VAV boxes, based on primary airflow, induction fan, and heater status.

Preliminary Assessment of Duct Leakage

To determine whether there was adequate leak-sealing potential for our retrofit study, we used duct pressurization techniques similar to those described by SMACNA (1985) to measure the leakage of six sample duct branches on the intervention floor, as well as the leakage of the main loop duct system on that floor. Table II-2 summarizes the results from those tests, using several different metrics to express the leakage. Duct surface areas in Table II-2 are based on field measurements of the ducts as they are actually installed.

Table II-2. "As Found" duct leakage based on duct pressurization tests.

*VAV Box**	*Number of Supply Grilles*	*ELA_{25} (cm^2)*	*Duct Surface Area (ft^2)*	*ELA_{25} / Surface Area (cm^2/m^2)*	*Leakage Class, C_L (cfm/100 ft^2)*	*Leakage Flow @ 1" w.c. (cfm)*
1702	3	53	198	2.9	128	252
1704	2	53	152	3.7	183	278
1708	2	54	139	4.1	200	278
1712	3	51	243	2.3	120	292
1720	5	67	321	2.2	114	367
1729	5	163	291	6.0	282	821
*Average***		56	210	3.1	149	294
Loop		52	5,076	0.1	6	309

* VAV boxes 1702 through 1712 are parallel fan-powered and have backdraft dampers at the fan discharge into the VAV box; the other two boxes (1720 and 1729) are unpowered and have no backdraft dampers.

** The averages in Table II-2 exclude Box 1729. That VAV box has two different, much leakier supply grilles (4 ft. long linear slot diffusers instead of 2 ft. square perforated-face multiple-throw diffusers).

Table II-2 indicates that the leakage class (C_L) for the tested branches (downstream of the primary air dampers, and including VAV box leakage at the induction fan air intake dampers) was approximately 114 to 282 cfm at 1 in. w.c. pressure per 100 ft^2 of duct surface area. This leakage is about mid-range compared to the leakage (C_L = 58 to 606 cfm per 100 ft^2) of branch ducts that we have tested in other large commercial building systems (Xu et al. 2002).

The supply loop is very tight (C_L = 6 cfm / 100 ft^2) in comparison to the branch ducts. This finding was not surprising given that the building is new and the main supply loop was well sealed with mastic. The supply loop includes all ducts between the supply fans and primary air dampers. Other systems that we have tested (Xu et al. 2002) have been leakier (C_L = 34 to 121 cfm/100 ft^2).

Component Leakage Tests: Component leakage tests that we conducted on the six branches indicate that induction fan backdraft damper leakage, grille edge leakage, and slot diffuser boot leakage are the most significant components of the branch leakage area. Excluding Box 1729, which has the slot diffusers with higher leakage areas, the backdraft dampers are about 7 to 13% of the leakage area (ELA_{25} of 4 to 7 cm^2), while the grille edges are about 54 to 75% of the leakage area (ELA_{25} of 29 to 51 cm^2). The larger fractions for the dampers correspond to boxes supplying fewer grilles; the larger fractions for the grille edges correspond to branches with more grilles. Accounting for the number of grilles in each branch, the average backdraft damper and grille edge leakage area (ELA_{25}) is about 5 cm^2 and 12 cm^2 respectively. Due to the long perimeter of each grille (8 feet), the grille edge leakage is very sensitive to grille seating on the T-bar supports: this leakage can easily vary by a factor of two due to poor seating.

The two slot diffuser boots attached to one takeoff from Box 1729 are very leaky. With the edges of the other three grilles sealed, 78% of the leakage remained (ELA_{25} of 128 cm^2). Even after considerable sealing of the supply boots to the slot diffuser with tape, 44% of the leakage still remained. Almost half of the slot diffuser boots are difficult or impossible to seal, because of poor access to parts of the assemblies that are located adjacent to elevator shaft walls. Fortunately, there are only four slot diffusers on the entire floor.

The component leakage tests that we have carried out do not address duct leakage airflows. Those airflows depend on leakage area and pressure across the leak. Leaks at the induction fan backdraft dampers are at much lower pressure differences than in the main loop, and leaks at the grille edges are likely at even lower pressure differences. Based on our monitoring, the loop operates at a pressure of about 250 Pa; the branch pressures in the plenums downstream of the VAV boxes vary widely from about 0.2 Pa to 84 Pa, more so for some boxes than others, and depend on the positions of the primary air dampers and whether the induction fans are operating. Such a wide range of pressures is not helpful in determining duct leakage airflows. For example, using the pressure difference range of 0.2 to 84 Pa with the component leakage areas that we have measured and extrapolating to the entire system indicates that the total system leakage could be anywhere from 500 to 7,000 cfm (about 60% in the loop in the lower case, and about 60% at the 99 grille edges in the higher case, with only about 10 to 15% from the backdraft dampers and slot diffusers in either case).

If the grille edges have significant leakage airflows, then it makes sense to try to seal them. Unfortunately, the pressure differences across the grille edge leaks are practically impossible to directly measure in the field, so one cannot determine the grille leakage airflows based on leakage area. To circumvent this problem, we conducted diagnostic tests to assess the impact of sealing the grille edges on the airflow leaving the grilles. Using a powered flow hood, we measured grille airflows before and after sealing the grille edges for all five grilles of one unpowered VAV box (using closed-cell-foam round backer rod as a seal; taping the joints produces the same effects, but the rod is faster to install). The airflows from the grilles increased 23 cfm from an initial total of 1,507 cfm (1.5% increase); each grille airflow increased 2 to 8 cfm. The total increase of 1.5% is not significant, as it is within the measurement precision of our flow hood. In contrast, the leakage area of the VAV box downstream section (including the grille edges) decreased from 69 cfm_{25} to 23 cfm_{25} when the grille edges were sealed (67% reduction). Based on these results, it appears that even though the grille edges have significant leakage area, the effective pressures at the grille edges are very low, and there is no significant leakage airflow across these edges. Therefore, it does not seem worthwhile to further pursue sealing techniques for grille edges.

To further understand opportunities for duct leakage reduction, we carried out laboratory diagnostics on one fan-powered VAV box to evaluate VAV box component leakage areas. Based on our measurements, about 30% (5 cfm_{25}) of the box leakage is across the partition separating the primary air path from the induction fan inlet, with about 70% (3 cfm_{25}) of that at the fan backdraft damper. Our component leakage field tests indicated damper leakage could be 2 to 3 times greater than this value in other boxes, but it seems that damper leakage may not be a significant issue, particularly when there are only a few VAV boxes with such dampers. In any event, field retrofits to reduce backdraft damper leakage would be difficult because of limited access and aerosol sealing cannot be used to seal the damper edges (the damper needs to open when the induction fan is on). Providing a better sealing for the backdraft damper and partition appears to be a design and manufacturing issue more so than an installation or field retrofit issue.

Characterization of HVAC System Airflows

Although air leaks from supply ducts, is captured in the return air, and may be regained from a thermal viewpoint, the leakage airflow does not reach the conditioned spaces directly. To maintain the main duct static air pressure at its set point, all leakage upstream of the VAV boxes must be made up by an increase in the supply fan airflow. Leakage downstream of the VAV boxes must be made up by supplying more air to the VAV boxes. To deliver more supply air, VAV box primary air dampers need to open further. Consequently, to maintain the main duct static pressure at its set point, an increase in the supply fan airflow is also needed to compensate for the downstream leakage airflows. The increase in the supply fan airflow in turn requires the supply fan speed to increase. Because the relationship between fan power and airflow is approximately a cubic function, the increase in supply airflow means that a large fraction of the supply fan power is used just to move the leaking air. Note that some of the thermal losses associated with duct leakage are not entirely recaptured during periods of economizer use, because relief fans discharge some of the return air directly to outdoors to maintain building envelope pressure differentials that would otherwise increase due to the increased outdoor airflows entering the building through the economizer.

Duct leakage airflows depends not only on leak size, but also on leak location, because the pressure difference at the leaks varies throughout a VAV system, particularly downstream of the VAV boxes. The pressure difference at each leak is determined by the airflows through the ducts and the airflow resistance of the duct system. Consequently, to understand duct leakage, one also needs to understand how airflows vary throughout the duct system. There are many HVAC system airflows of interest in this study: supply airflows, return airflows, outdoor airflows, VAV box primary airflows, VAV box induction airflows, supply grille airflows, and duct leakage airflows. The following sections describe in more detail how we determined each type of airflow.

HVAC System Airflow Calibrations and Diagnostics

Few HVAC system airflows on the intervention and control floors could be monitored using the Energy Management Control System (EMCS). The only airflows that could be monitored this way were the minimum outdoor airflows and the primary airflows entering each VAV box. We installed equipment to monitor supply fan and return airflows (the difference being total outdoor airflow, which includes economizer airflow). In particular, we installed equipment to measure the pressure difference across existing airflow grids (air-handler supply airflow) and across flow grids that we installed (return airflow). These pressure differences were then correlated to airflows by applying calibration equations that we developed.

We also made detailed airflow measurements for four VAV boxes: two have induction fans operating in parallel with the primary air; the other two are non-powered and have no induced airflow. We calibrated the primary inlet flow grid for each box and, for the boxes with fans, also measured the actual amount of induction airflow while varying the primary airflows. Obtaining these airflows with and without added downstream duct leakage was advantageous because the resulting data will support future analyses of the efficiency of the VAV units and their downstream duct sections under varying operating conditions and leakage configurations. The flow grid data have already been useful in assessing and rejecting various duct leakage sampling techniques that might use the EMCS airflow data to determine duct leakage airflows.

Calibration of the Supply Fan Flow Grids via Tracer Gas: Each supply fan (two on the control floor and two on the intervention floor) has an inlet airflow grid to measure the supply airflow, but none were connected to the EMCS system; each grid was instead connected only to an analog gauge that allows building engineering staff to determine airflow on a “snapshot” basis. To monitor the supply airflow continuously (minute by minute), we installed pressure sensors connected to our data loggers and recorded the pressures that the airflow grid sees. Figure II-2 is a picture of a supply fan inlet with its flow grid.

Figure II-2. Air-handler supply fan inlet. Note the two vertical copper tubes that comprise the airflow station. A tube for lubricating the shaft bearing can also be seen, as well as one of the temperature sensors that we installed.

For each fan, to determine the relationship between the pressure difference across the supply fan flow grid and the airflow through the supply fan, we first calibrated the flow grid using a tracer gas technique that determines the airflow. The approach was to set the system for 100% outdoor air (with closed and sealed return air dampers). Sulfur hexafluoride[3] tracer gas was then injected into the duct system at a constant and measured rate immediately upstream of the cooling coil (which is upstream of the supply fan and downstream of the return and outdoor air inlets). A diffuse injection was accomplished by using a "soaker hose" spread out across the coil. The tracer gas concentration was monitored downstream of the fan before any duct branching, using a tracer gas analyzer calibrated at the measurement site. The airflow through the fan is given by Equation 1:

$$Q_{fan} = I/C \tag{1}$$

where Q_{fan} is the fan airflow ($[m^3_{air}]/s$), I is the tracer gas injection rate ($[m^3_{gas}]/s$), and C is the concentration of the tracer gas ($[m^3_{gas}]/[m^3_{air}]$).

[3] This tracer gas has been used extensively in past studies of other buildings. The engineering staff in the test building approved the tracer gas use after reviewing its "Material Safety Data Sheet" and after smelling the gas (no smell).

The major obstacle to tracer gas measurements of fan airflow (apart from equipment costs) is potentially poor mixing of tracer gas in the air stream between the tracer injection point and the downstream location where the tracer gas concentration is measured. Good mixing in our tests was confirmed by collecting and analyzing samples from multiple downstream locations inside the duct.

Uncertainties in the supply airflow measurements are due to uncertainties in the tracer gas injection rate, uncertainties in the tracer gas concentration, and uncertainties caused by sampling imperfectly mixed tracer gas in the air. With proper calibration and operation of the instruments, both uncertainties in the tracer gas injection rate and concentration can be as low as approximately 1% individually. Uncertainties due to an imperfect characterization of the well-mixed tracer concentration downstream of the injection point were about 2.5%.

By repeating the tracer gas tests at several different fan airflows, a curve fit was developed to determine the correlation between measured airflow (from the tracer gas) and grid pressure difference The uncertainty in the curve fit is 2 to 3% of estimated fan airflow at the 95% confidence level. Consequently, the uncertainty of an airflow predicted from a measured pressure difference varies from 3 to 4%. Figure II-3 shows a typical calibration for a supply flow grid.

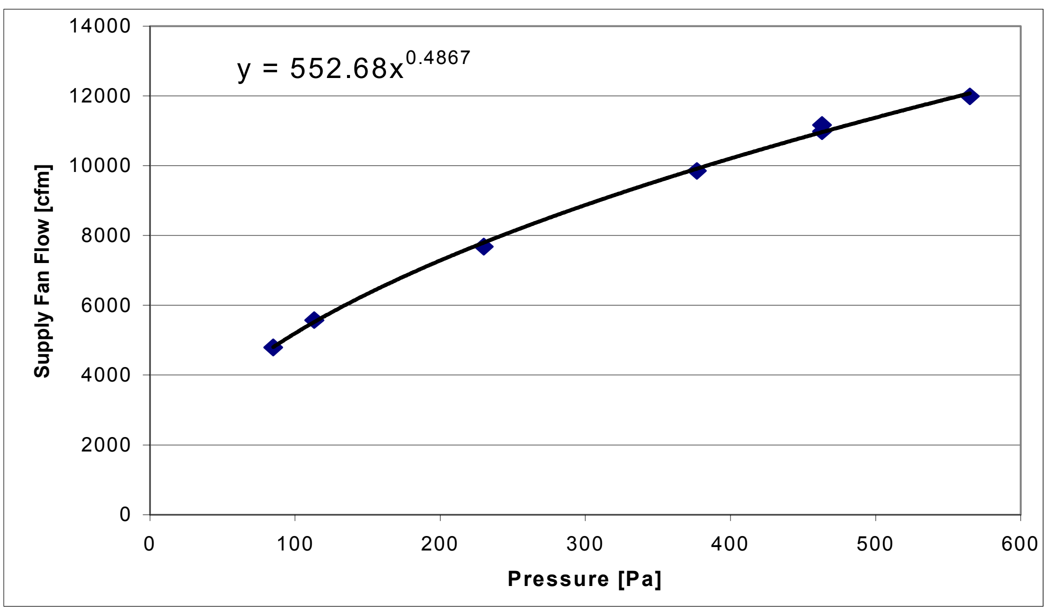

Figure II-3. Calibration for the intervention floor's West supply flow grid.

Construction and Calibration of the Return Flow Grids: Although there was an airflow sensor connected to the EMCS to measure the minimum outdoor airflow for each supply fan, there was no existing measurement equipment in the building to measure return airflow or the total outdoor airflow entering the mixing box upstream of each supply fan (sum of the economizer airflow and the minimum outdoor airflow). Therefore, we monitored the return airflows by building, installing, and calibrating a new flow grid. We chose to measure the return airflow rather than the outdoor airflow because the return airflow was less likely to be affected by airflow non-uniformities: it did not have upstream duct work (the outdoor air ductwork had a 90 degree turn just upstream of the measurement section) and it was possible that wind gusts at the outdoor air intake louvers might affect airflow patterns at the measurement section.

The flow grid that we constructed consisted of a wooden box fitted around the return damper assembly, with a set of five tubes having three equally spaced small holes facing into the airflow and another set of five tubes downstream having similarly located and sized holes facing away from the airflow. Each upstream hole was drilled into a dimple that we hammered into the grid tube; this dimpling reduces the angular sensitivity of aligning the holes with the airflow. No dimpling was used on the downstream holes because they are less sensitive to the airflow approach angle. We used downstream holes rather than measuring the static pressure in the flow grid section so that we could obtain larger pressure difference signals. For each airflow direction, the 15 holes in the five tubes are manifolded together and therefore linearly "averaged". Each supply fan had two such devices placed over the two adjacent sets of return air damper assemblies. Equal length tubing was used to manifold and therefore average the pressures from the corresponding sections of the two return grids. Figure II-4 shows one of the return flow grids installed on the top of an air-handler mixing box (the right hand damper assembly, at the edge of the picture, had not yet received its flow grid).

Figure II-4. An installed return flow grid assembly.

We calibrated the return flow grid by closing and sealing the outdoor air dampers. In this configuration, all the supply air flowed through the return flow grid. With the supply fan operating at various speeds, we monitored the pressure differences across the return and supply grids. Then, we calculated the supply airflow using the correlations described in the previous section and developed a correlation between the resulting airflow and the monitored return grid pressure difference data. We also collected data at several different return damper positions to confirm that there was no influence on the calibration for different damper positions. Figure II-5 shows a typical calibration result.

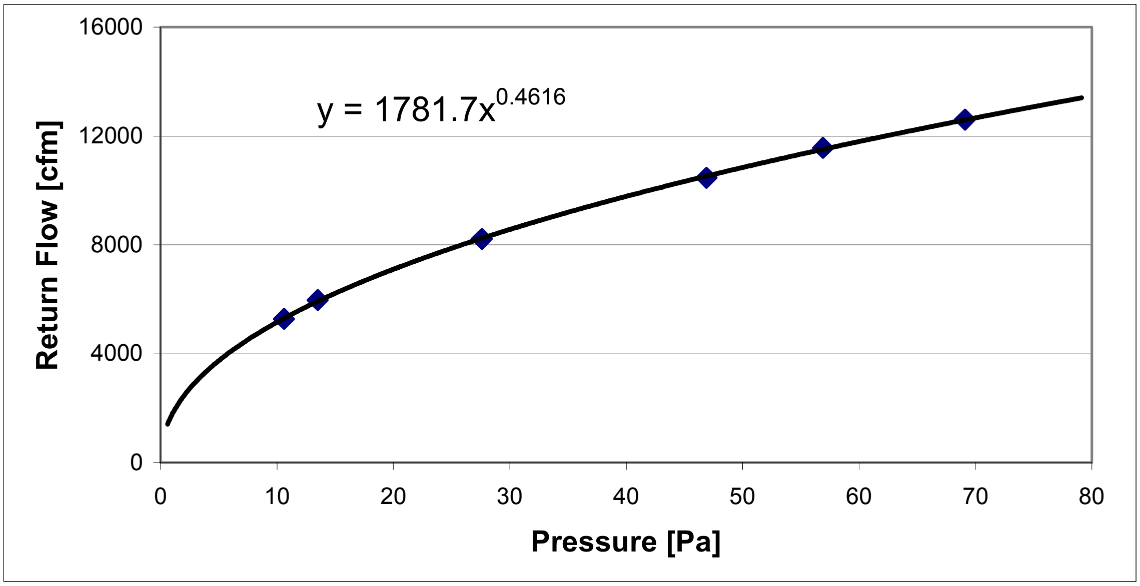

Figure II-5. Typical return flow grid calibration.

Calculation of Outdoor Airflow: The calculation of total outdoor airflow is simply the difference between the total supply airflow and the return airflow.

Calibration of VAV Box Primary Airflow Grids: Each VAV box has a factory installed radial flow grid. The EMCS monitors the primary airflow entering each VAV box using this grid and an adjacent pressure sensor. However, these flow grids and pressure sensors were apparently not calibrated in the installed configuration. This was evident in cases when the EMCS values indicated airflow when the air-handler supply fan was off. We confirmed this problem by turning the air-handler supply fans off and manually closing the primary air damper for the VAV box. For the four VAV boxes that we monitored in detail, we calibrated these flow grids using the same tracer gas method that we used to calibrate the air-handler supply fan flow grids.

Calibration of Detailed VAV Induction Fan Airflows: The induction fans in the powered VAV boxes have gravity dampers at their outlet into the boxes. During testing and balancing, the maximum opening of these dampers can be adjusted to “fine tune” the induction fan airflow after coarse adjustments are made using fan motor speed taps. We found that many of these dampers were set at more than 50% closed. In addition, VAV box pressures (which create a backpressure for the induction fan) change as the primary airflow changes. Therefore, it is unlikely the induction fans could operate at their rated airflow. To determine these airflows, we measured the airflows directly using a calibrated fan and flow meter device (similar to the device used to measure supply grille airflows). We found that the fans have a nearly constant airflow regardless of the primary inlet airflow. It was not possible to use the tracer gas method to calibrate these airflows, because there was no practical way to fully mix the injected tracer gas before the ducts split off from the plenum downstream of the VAV box.

Determining Duct Leakage Airflows and Adding Additional Leakage

We determined the existing leakage airflows at a reference condition. This measurement involved measuring the airflow out of each supply grille during a fixed and known supply fan airflow, and comparing the sum of the grille airflows to the supply fan airflow; the difference is the leakage. We also estimated the distribution of "base case" (pre-intervention) duct leakage airflows by correlating duct leakage area with observed pressure differences across the duct system walls. To determine the leakage areas, we used fan pressurization techniques on the upstream duct section and on a sample of six downstream sections to measure leakage area. Pressure differences were monitored using pressure taps and sensors that we installed.

As discussed later in the "Results of HVAC System Airflow Tests" section, the total "base case" leakage airflow of the invention floor was low. Consequently, rather than sealing a leaky duct system to determine energy savings associated with duct leakage (as we originally intended), we instead added duct leaks in both the upstream section and downstream sections to test the hypothesis that duct leakage will lead to increased fan power.

"Base Case" Upstream Leakage: We estimated the "base case" duct leakage airflows upstream of the VAV boxes on the intervention floor by using a duct pressurization test. Although this test is a poor indicator of leakage airflow for duct sections where pressure differences are highly variable (the test only determines leakage area), we were able confirm that the pressure difference was essentially the same throughout the upstream section of the duct system. It is probable that this constant pressure assessment would not be true if only one of the two supply fans was used or if there was a very large difference between the VAV airflows on the East and West sides.

To carry out the duct pressurization test on the upstream duct section, we manually closed all 34 VAV box primary air dampers and sealed both air-handler supply fans at their inlets on the intervention floor. A panel with a calibrated fan and flow meter device was then installed in place of one supply fan access door. We monitored duct pressures at both supply fan plenums; these were essentially equal throughout the test. The resulting leakage curve includes the leakage of the VAV primary air dampers, which we estimated using laboratory measurements of one VAV box (0.6 cm^2 ELA_{25} per damper). The upstream leakage is then the difference between the leakage curve we measured and the leakage curve of the leakage from all the VAV inlet dampers combined. The VAV inlet damper leakage was about 34% of the total measured upstream leakage.

Although the total leakage airflow, which includes primary air damper leakage, is well determined by this pressurization test (±5% of leakage airflow, <±1% of supply fan maximum airflow), it is difficult to estimate the error in the upstream leakage part with the dampers excluded, because we only measured the VAV inlet damper leakage of one unit.

"Base Case" Downstream Leakage: We estimated the total "base case" duct leakage airflows downstream of the VAV boxes from the difference between the total airflow supplied to the boxes and the sum of all the supply grille airflows. The total airflow supplied to the boxes is simply the difference between the fan airflow and the upstream leakage. To make these measurements, we needed a stable, reproducible system configuration. This configuration was to set all the VAV box dampers fully open (EMCS set to request 5,000 cfm for each VAV box, which is larger than the maximum airflow of any box), to fully close all the outdoor air dampers, to fully open the return dampers, to set the supply fans to maximum airflow, and to set the relief fans to off. We call this configuration the "sum of supplies" mode of operation.

To measure the supply grille airflows, we used our “powered flow hood”. Laboratory results (Walker et al. 2001, Wray et al. 2002) have shown this flow hood to be very accurate. However, it is also quite slow and cumbersome to use (it took five people 12 hours to measure the airflows from all 103 supply grilles).

Because this “sum of supplies” test was carried out over several nights, one supply grille was measured repeatedly in this configuration to check the system stability and repeatability. The airflow at this grille was repeatable to within ±2% for the various days that we set the system in this configuration. It should be noted that during one night of very high winds, the airflows varied widely and we were unable to use any results from that night. We found that repeatable measurements could be made when wind speeds were less than 15 mph.

To extrapolate this leakage from the “sum of supplies” mode to normal operation conditions, we assume that the downstream background leakage airflow is a constant fraction of the supply airflow delivered to the VAV boxes. Consequently, the “base case” downstream leakage at operating conditions is the fraction of background downstream leakage during the “sum of supplies” mode multiplied by the total airflow delivered to the VAV boxes at operating conditions.

Because of the difficulties in using our powered flow hood for this test, we also carried out field tests using five commercially available flow hoods to determine if they could be used to measure the grille airflows as accurately, but more rapidly. The tests were performed on four different multi-branch subsections of the VAV system, using one supply grille on each section (there were two to five grilles for any one subsection). We selected four grilles to cover a range of nominal grille airflows from 50 to 200 L/s (100 to 400 cfm). The grilles were all 2 ft square, 4-way throw, with a perforated face (3/16" holes, 1/4" on center). The VAV system was set in the “sum of supplies” mode as described in this section to provide constant airflow during the test period. On each grille, the five commercially available hoods were used in sequence to measure the airflows. The reference airflow for each grille was measured using our powered flow hood. Manufacturer's instructions for hood operation were followed in each case (e.g., the use of relief vents or low-flow plates). We found that it was essential to follow instructions properly because it was easy to use the wrong operating mode and get large errors (e.g., we found an error of 38% by using incorrect vent modes for one of the flow hoods).

The results of these field tests are summarized in Table II-3. The results show that the hoods exhibited similar trends, with under-prediction of low airflows. Overall, Flow Hood 1 had the best performance with bias and RMS errors less than 2%. This RMS difference is close to the accuracy of the powered flow hood itself, which shows that for this grille type, Flow Hood 1 can give the same results as our reference device within the uncertainty specification of the reference. Flow Hoods 3 and 5 were a little worse, with RMS errors approaching 5%. Flow Hoods 2 and 4 exhibited significant biases and under-predicted airflows by more than 10%.

Table II-3. Summary of field test results for five commercially available flow hoods on four commercial grilles.

Flow Hood	Bias Error, L/s (cfm)	Bias Error, %	RMS Error, L/s (cfm)	RMS Error, %
1	1 (3)	1	2 (4)	2
2	-14 (-29)	-11	17 (36)	11
3	6 (12)	4	7 (15)	5
4	-9 (-20)	-11	10 (21)	14
5	-3 (-7)	-2	6 (12)	4

Our tests indicated that 20 grille airflows could be measured in 35 minutes using Hood 1. This means that two people each using a hood could measure 100 grille airflows in less than two hours. Because hood accuracy depends on grille type, further work is needed to determine if these results can be extended to other buildings.

Installed Upstream Leaks: We installed duct leaks in the main duct loop upstream of the VAV boxes at five locations. The leaks consisted of a perforated plate (similar to the perforated plates that are used in the supply grilles), which we mounted in an end cap covered with cardboard. All these leaks were installed on ten inch diameter ducts that protruded out from the existing main loop by 1 to 3 feet. Figure II-6 shows one of these installed leaks.

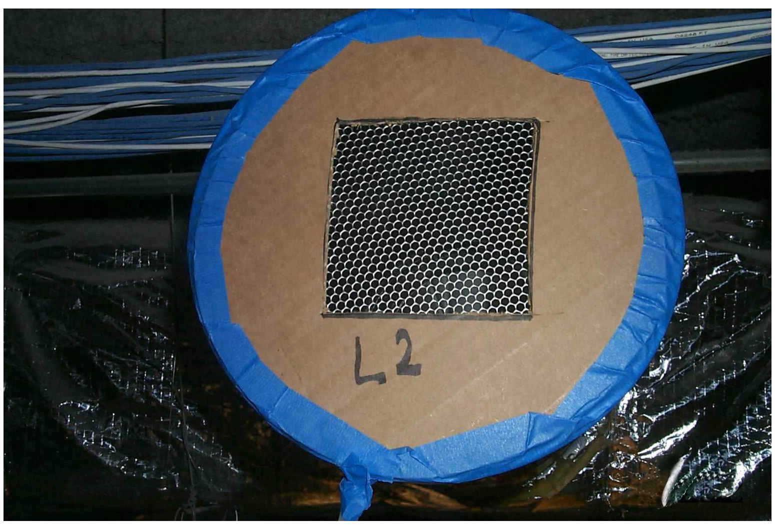

Figure II-6. Example of an installed upstream leak.

The amount of leakage was adjusted by cutting out a section of the cardboard. We had calibrated the perforated plate in the laboratory to determine its leakage characteristic per unit area. The pressure difference seen by the upstream leakage was found to be the same at all five leak sites over a wide range of total supply airflows, so we only needed to monitor one pressure difference to determine all the leakage airflows through the upstream leaks that we added.

Installed Downstream Leaks: We also installed leaks in six locations downstream of the VAV boxes. To determine which VAV boxes had enough capacity to accommodate the leakage airflows and still provide the design airflow to the occupants, we set each VAV box in turn to its maximum airflow at "design static pressure" and recorded these airflows. The branches that we selected were from all four sides of the building and included both core and perimeter zones.

The installed downstream leaks used the same material as the installed upstream leaks. An example is shown in Figure II-7.

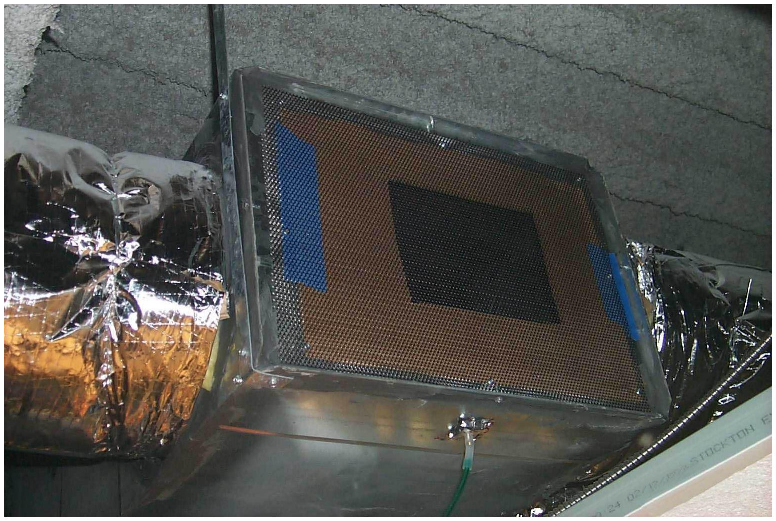

Figure II-7. Example of an installed downstream leak at the end of a plenum. Ducts can be seen exiting two sides of the plenum. A static pressure sensor is mounted adjacent to the installed leak.

A correlation between the pressure difference across each installed downstream leak and the corresponding upstream primary airflow was obtained by fitting curves to pressure differences that we measured during diagnostic tests at various VAV primary airflows (obtained from the EMCS), with and without the induction fan operating (if the box had a fan). A separate relationship was needed when induction fans were used, because the induction fans add airflow downstream of the VAV box primary airflow grids.

Analysis of Results

To evaluate the effects of upstream and downstream duct leakage separately and in combination, we monitored the performance of the intervention floor system over four test periods, defined as:

1. Base (pre-intervention conditions)
2. Upstream (leaks added to the main duct loop)
3. Downstream (leaks added to the ducts downstream of the VAV boxes)
4. Upstream and Downstream (leaks added to the loop and downstream of the VAV boxes)

For comparison, we conducted similar monitoring of the control floor during the same test periods. However, we made no changes to the ducts on that floor. The test periods ranged from one to two weeks each and took place during the summer of 2002. We looked at overall impacts (full day analysis – 3 am to 6 pm) as well as utility peak period impacts (2 pm to 6 pm). To compare a variable for the same time period for both the control and intervention floor, we included only those observations when air-handler supply fans were operating on both the control and intervention floors.

Results of HVAC System Airflow Tests

Air-Handler Supply Airflows: Table II-4 lists the average supply fan airflows for the intervention floor during the various test periods, without corrections for weather differences.

Table II-4. Average supply fan airflow (cfm) during testing period (intervention floor, not weather corrected).

	Leakage Configuration			
Operating Period	"Base"	"Up"	"Down"	"Up & Down"
All Day	10,700	11,900	12,800	14,100
Peak Hours	11,300	11,900	12,000	13,900

*All day refers to the 3 am to 6 pm period while peak refers to the 2 pm to 6 pm period.

As Figure II-8 shows, daily average air-handler supply fan airflows for the control (16th) and intervention (17th) floors varied in response to weather-induced thermal loads driven by the outdoor air temperature. In particular, these airflows generally increased over the entire measurement period, with larger airflows occurring in the hotter months. For all seasons, the airflows for the control (16th) floor were larger than the airflows for the intervention (17th) floor.

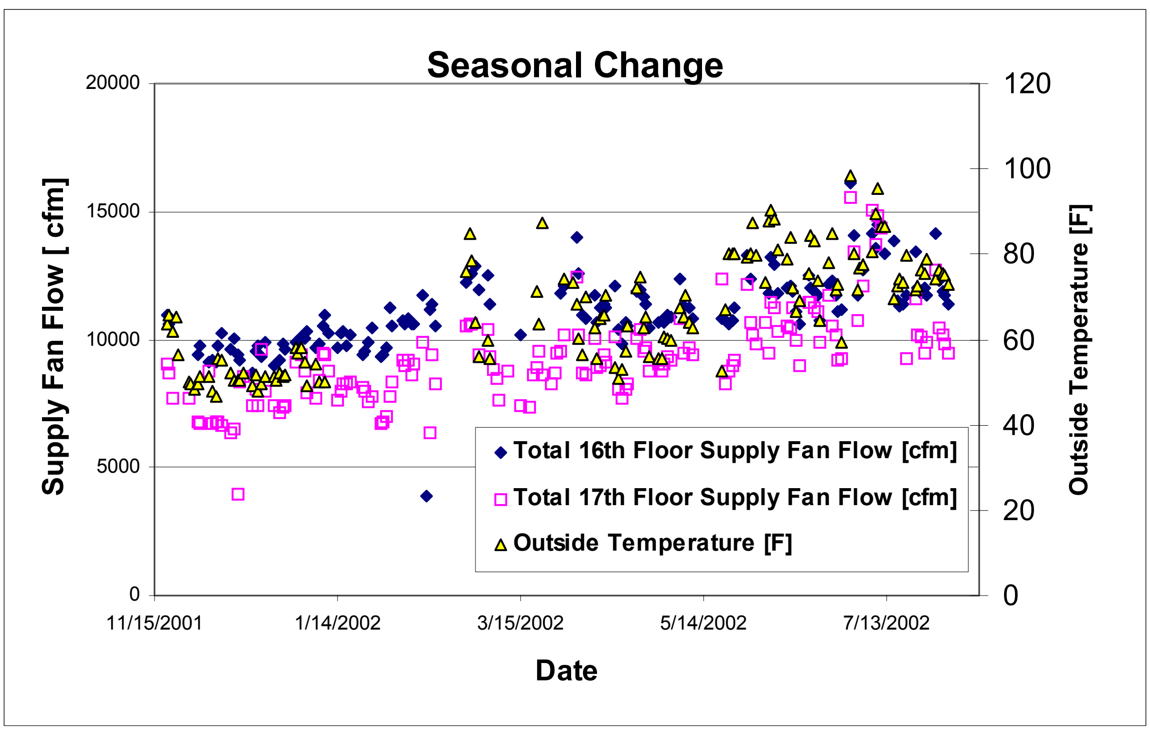

Figure II-8. Daily average supply fan airflow and outdoor temperature for the entire measurement period.

Although the four leakage intervention study periods took place during the summer of 2002 with similar occupancy patterns on both floors, the weather conditions during each period were somewhat different from each other. Table II-5 lists the average outdoor temperatures for each test period, as recorded by the EMCS.

Table II-5. Average outdoor temperature (°F).

Leakage Configuration	All Day (3 am to 6 pm)	Peak Period (2 pm to 6 pm)
Base	77	90
Up	78	80
Down	81	93
Up & Down	89	102

It is important to note that the supply airflows exiting the air-handlers during operating periods were not constant, even over the course of a day. Figure II-9 shows an example of the daily behavior of air-handler supply airflows for the "Base Case" condition.

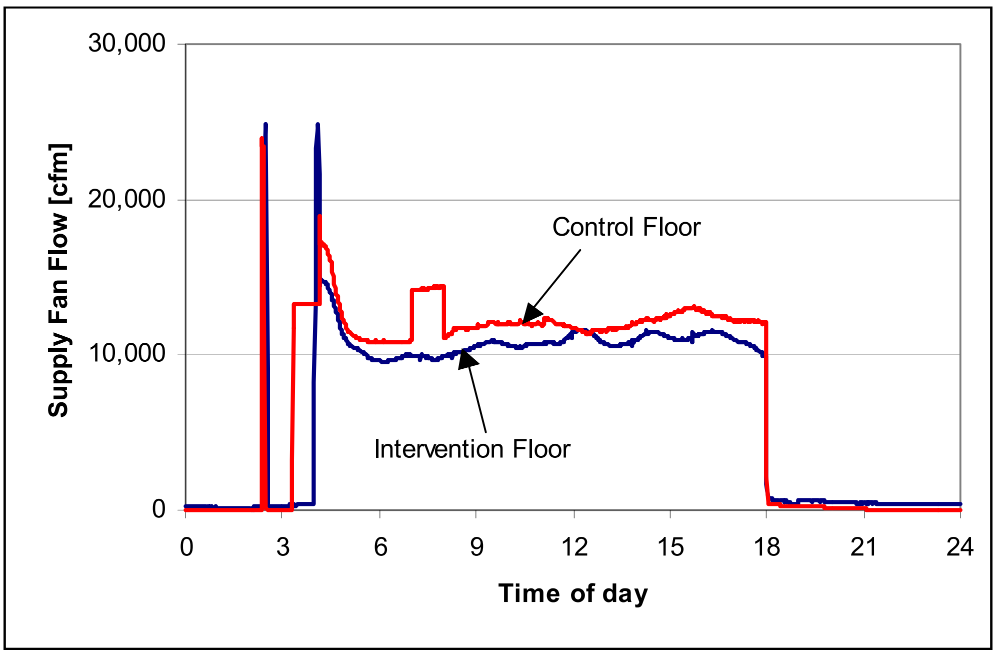

Figure II-9. Supply fan airflows from a sample "Base Case" day.

Figure II-9 shows a number of interesting features:

- At about 2:30 a.m. every day, the fans are turned on for a five minute long "system check".
- On the example day shown (July 23, 2002), one of the supply fans on the control floor is turned on at about 3:30 a.m. for about half an hour to precool the floor.
- The abrupt increase in the control floor supply airflows at about 7 a.m. is typical behavior during the summertime evaluation period. It is not know why this increase occurs.

- Generally ,the supply airflows are larger than average during the first hours of “normal” operation starting at 4 a.m.
- The HVAC system is turned off at 6 p.m. every day.

Weather Normalization: To take the temperature differences and corresponding operational differences into account for different periods, a normalization procedure can be used to compare the airflows (or other variable such as power) measured during the leakage case test periods to that measured during the base case period. The fractional change in value is calculated as follows:

$$\text{Fractional} \cdot \text{Change} = \frac{\overline{X}_{IL} - \overline{X}_{IB}}{\overline{X}_{IB}} \times \frac{\overline{X}_{CB}}{\overline{X}_{CL}} \qquad (2)$$

where:

X	=	the variable being studied for the time period specified,
IL	=	intervention floor, leakage case time period,
IB	=	intervention floor, base case time period,
CL	=	control floor, leakage case time period, and
CB	=	control floor, base case time period.

The first part of Equation 2, based on the intervention floor data, calculates the intervention floor fractional change in a value due to the additional duct leakage installed. The second part of Equation 2 provides a normalization factor using the control floor data. The result is the fractional change in the variable studied (e.g., airflows, power, induction fan on-time).

This equation can be further derived to estimate what the airflows (or other variable) would be if the intervention floor was operated in a given leakage case mode during the base period. The derivation of this equation is:

$$\overline{X}_{IBL} = \overline{X}_{IB} \times (1 + \text{Fractional} \cdot \text{Change}) \qquad (3)$$

where:

$\overline{X}_{IBL}$ = Estimate of what the variable average value would be if the corresponding case took place during the base case time period.

Unless otherwise noted, these equations were used to calculate the normalized changes due to duct leakage for the two time segments (all day and peak). In order to compare a variable for the same time period for both the control and intervention floor, we included only those observations when both control and intervention floor air-handler supply fans were on.

Using these normalization techniques for each leakage case, we have estimated what the average air-handler supply airflow would be if we ran the system in a given leakage case, but during the base case weather period instead. As shown in Table II-6, the base case average airflows range from 10,700 cfm (all day) to 11,300 cfm (peak). If the system were to operate in a leakage case over the same time period, we estimate these values would range from 11,900 (all day and peak) to 13,700 (all day) and to 13,800 (peak). Due to higher supply fan airflows during the morning cool-down period, the all day average impacts are greater than that found for the peak period. In both all day and peak period analyses, the impact of the downstream leakage case (18% all day and 7% peak) is greater than that of the upstream leakage case (11% all day and 6% peak). The greatest impact is when there are both upstream and downstream leaks (28% all day and 22% peak period).

Table II-6. Normalized average airflow (cfm) for air-handler supply fans.

Leakage Configuration	*% Change in Airflow*		*Estimated Average Airflow for Base Case Test Period (cfm)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	10,700*	11,300*
Up	11%	6%	11,900	12,000
Down	18%	7%	12,600	12,100
Up & Down	28%	22%	13,700	13,800

*base case averages.

VAV Box Primary Airflows: Most VAV primary air inlet airflows quickly settled down to nearly constant values for the day; the exception being the behavior of VAV Box 1712. The reason for the different behavior is unknown. Figure II-10 shows a sample "Base" day of the VAV box primary airflows for the four boxes that we monitored in detail.

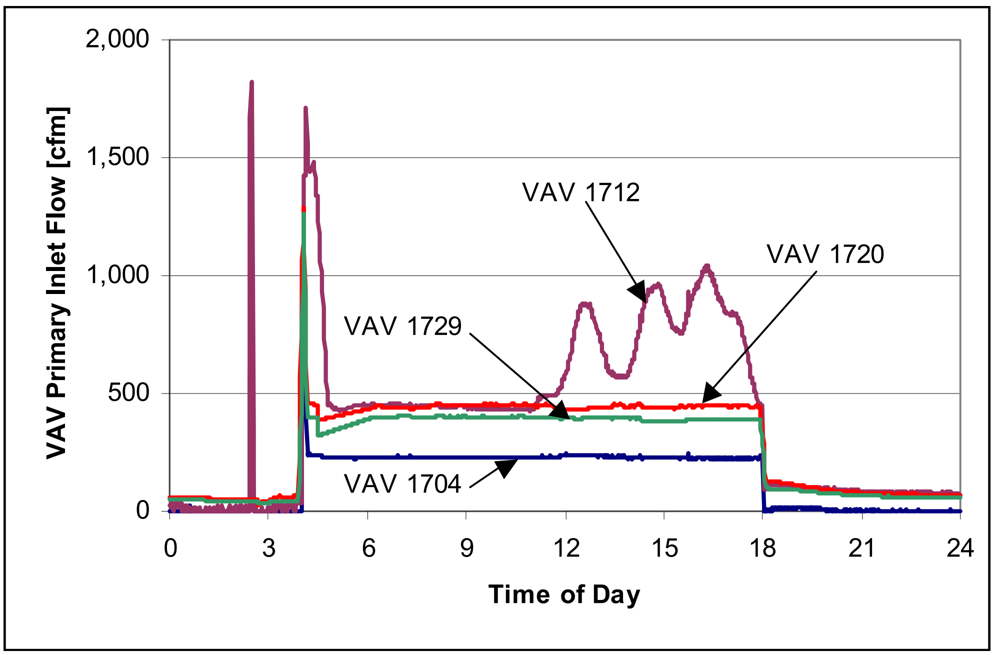

Figure II-10. Four selected VAV box primary airflows for a sample "Base Case" day.

Duct Leakage Airflows at Operating Conditions: Table II-7 lists the average upstream, downstream, and total leakage fractions and airflows during the various leakage configurations.

Table II-7. Average upstream, downstream, and total leakage fractions and leakage airflows at operating conditions.

		Leakage Configuration							
		"Base"		"Up"		"Down"		"Up & Down"	
Operating Period	Leak Location	Leakage (%)	Flow (cfm)	Leakage (%)	Flow (cfm)	Leakage (%)	Flow (cfm)	Leakage (%)	Flow (cfm)
All Day	Upstream	3.2	340	14.6	1,737	2.7	340	12.3	1,738
	Downstream	2.3	245	2.0	239	8.0	1,022	7.4	1,042
	Total	5.5	585	16.6	1,976	10.7	1,362	19.7	2,780
Peak Hours	Upstream	3.0	340	14.6	1,737	2.8	340	12.5	1,737
	Downstream	2.3	258	2.0	239	8.3	990	7.4	1,022
	Total	5.3	598	16.6	1,976	11.1	1,330	19.9	2,759

The leakage fractions in Table II-7 are determined as a percentage of the total supply fan airflow measured during the corresponding operation and leak configuration period (not weather corrected). Note that the leakage fraction of the "Up & Down" case is not the sum of the "Up" and the "Down" case, because the background leakage would be counted twice and because the total supply fan airflow changes for each case. However, upstream and downstream leakage fractions combine directly in any particular case and equal the total leakage fraction in that case.

Figure II-11 shows how the duct leakage airflows vary over a sample day for the "Up & Down" configuration.

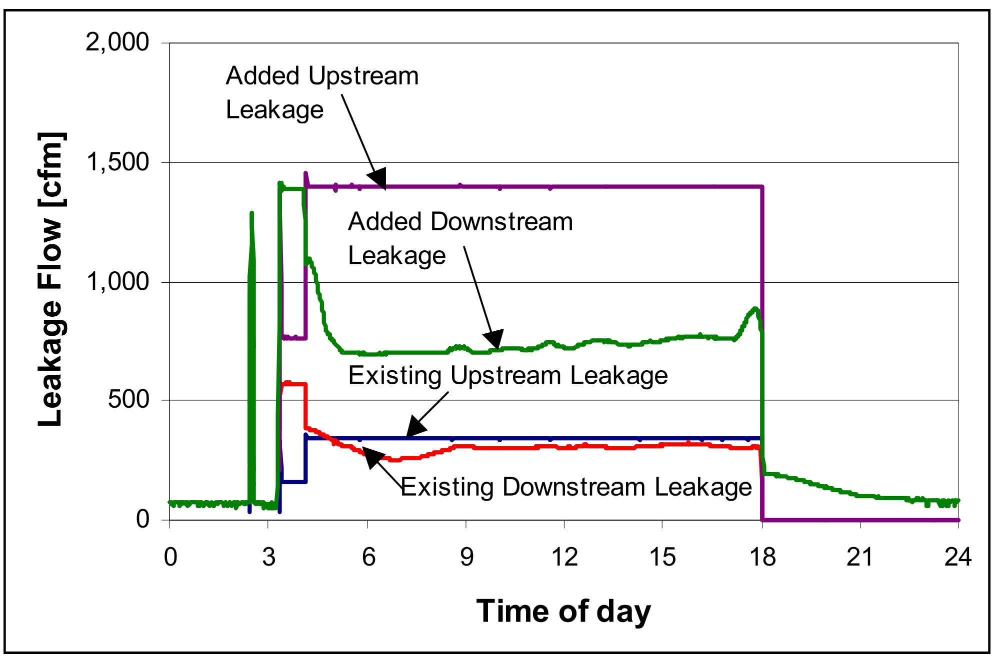

Figure II-11. Duct leakage airflows for a sample "Up & Down" day.

Regardless of the leakage configuration and supply fan airflow, the upstream leaks are at an almost constant pressure (the "design static pressure") and therefore have a nearly constant leakage airflow throughout the "normal operating" hours. This is a result of the system design: the pressure in the main duct where the upstream leaks are located is controlled to maintain the "design" set point of 1 in. w.c. (250 Pa). In contrast, the downstream leakage airflow varies as each of the six VAV boxes that had leaks installed vary their primary airflow and as induction fans start and stop.

Duct Leakage Airflows at "Design" Conditions: The duct leakage airflows discussed in the previous section are the actual values found at operating conditions during our measurement period. If one wants to specify a reproducible leakage requirement for a performance standard, then the system configuration will need to be specified.

We define the leakage performance standard conditions to be:

- The air-handler supply fan should be at 100% of its full rated speed.
- The air-handler supply fan airflow shall maintain the maximum designed static pressure.
- The VAV boxes are uniformly open (all are at the same fraction of their design maximum airflow) to the maximum degree possible while the first two conditions are met.
- All induction fans, terminal heaters, and relief fans are off.
- The return dampers are fully open and the outdoor air dampers are closed as much as possible.

These conditions apply to the type of control system that was used in our test building. There may be additional specifications needed for other system layouts and controls.

In our test building, we produced these conditions during the "Base Case" test period by adjusting the VAV primary airflow set points until we were just able to maintain the design static pressure of 1 in. w.c. (250 Pa). This operating point corresponds to the VAV boxes being set at 75% of their "cooling maximum" airflows and the total supply fan airflow at 24,400 cfm. We call this airflow the "design static airflow". Note that the system never operated in this leakage performance configuration during normal operation.

In Table II-8 for each of the leakage configurations, we list the upstream, downstream, and total duct leakage airflows themselves, and as a fraction of the "design static airflow".

Table II-8. Leakage airflows and leakage fraction of supply fan airflow at the "design static airflow" of 24,400 cfm.

		Leakage Configuration							
		"Base"		"Up"		"Down"		"Up & Down"	
Operating Period	Leak Location	Leakage (%)	Flow (cfm)	Leakage (%)	Flow (cfm)	Leakage (%)	Flow (cfm)	Leakage (%)	Flow (cfm)
"Design"	Upstream	1.4	340	7.1	1,737	1.4	340	7.1	1,737
	Downstream	2.3	567	2.2	534	6.5	1,595	6.2	1,507
	Total	3.7	907	9.3	2,271	7.9	1,935	13.3	3,244

Economizer Operation: An economizer can be used to mix outdoor air with return air to reduce cooling coil loads when the enthalpy of the outdoor air is lower than the enthalpy of the return air. The economizer mode can also be used to precool the building; building engineering staff advised us that the EMCS has been configured to enable economizer operation for precooling based on an EMCS survey of temperatures at midnight. During the summer evaluation period, the economizer was often used for precooling, but there was no consistency between usage on the control and intervention floors. In particular, the time that the control floor economizer would start operating varied from day to day; the economizer on the intervention floor sometimes started at a different time, or not at all. This suggests that the economizer may have been regulated by means other than the stated EMCS control sequence, such as an optimal start routine.

The daily variation in outdoor airflows entering the supply fan is complex, as shown in Figure II-12. This figure shows airflows for the West side of the control floor. The supply airflow is mostly composed of outdoor air with some abrupt drops in total outdoor airflow and corresponding abrupt increases in the minimum outdoor airflow. These changes can happen at any time during the normal occupied hours. Also interesting is the time from about 3 to 4 a.m. when the East side fan is on for precooling. In these conditions, the East fan pressurizes all of the duct system, including the West side air-handler, resulting in airflow going out of the West outdoor air intake.

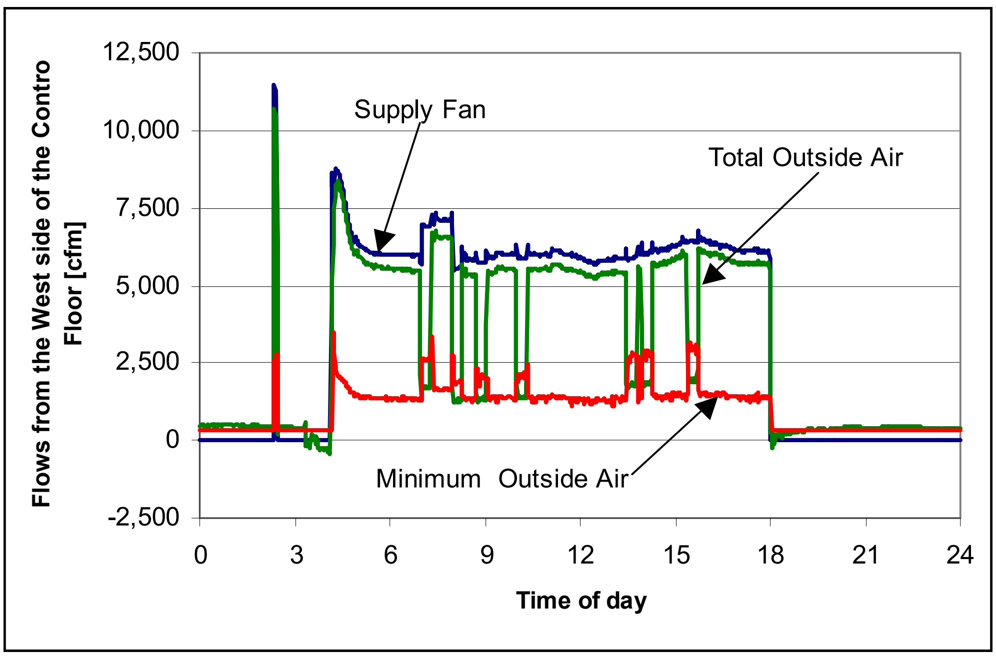

Figure II-12. West supply fan and outdoor airflows on the control floor, for a sample "Base Case" day.

The fluctuating behavior of the economizer operation, particularly in the precooling mode, makes it difficult to normalize and compare economizer behavior for the different leakage cases. For example, Table II-9 shows the average outdoor air fraction of supply airflow without normalization for each case. During the base case all day operating period (3 am to 6 pm), there is a significant difference in the outdoor air fractions for the control and intervention floors (66% vs. 38%), when precooling sometimes occurs. However, during the peak cooling hours (2 pm to 6 pm) when there is no precooling, there is almost no difference in these fractions (44% vs.

48%). This influence can also be seen in the variation of the outdoor air fractions during the different leakage cases. Because the economizer operational strategy is not clear, we cannot disaggregate the impact of duct leakage on economizer operation.

Table II-9. Average outdoor air fraction of supply airflow.

	Leakage Configuration							
	"Base"		"Up"		"Down"		"Up & Down"	
Operating Period	Control Floor	Intervention Floor	Control Floor	Intervention Floor	Control Floor	Intervention Floor	Control Floor	Intervention Floor
All Day	66%	38%	58%	35%	60%	35%	32%	29%
Peak Hours	44%	48%	46%	45%	35%	44%	20%	32%

Fan Power Impacts of Duct Leakage

A significant objective of this project is to determine whether the presence of duct leakage changes the amount of electricity required to operate the distribution system fans. We studied this by measuring and calculating fan power during the four test periods. For comparison, we measured and calculated fan power for the control floor during the same test periods.

Fan Power due to Increased Duct Leakage: During the cooling season, the amount of electrical power required by the HVAC system to transport conditioned air includes the power to drive the air-handler supply fans, relief fans, and VAV box induction fans. In reviewing the patterns of fan operation, we found that the supply fan and induction fan airflows and operation are impacted by the introduction of additional duct leakage. The relief fans are operated to maintain building pressure set points and run as needed during pre-cooling and economizer modes. As a result the relief fans have an irregular operational pattern. We did not see a correlation between duct leakage and relief fan operation. As such, we will only discuss the impact of duct leakage on the air-handler supply fans and the induction fans, and not on the relief fans.

Air-Handler Fan Power: Table II-10 summarizes the increase in air-handler supply fan power for each of the leakage cases compared to the base case.

Table II-10. Normalized average fan power (kW) for air-handler supply fans.

Case	*% Change in Fan Power for Air-Handler Supply Fans*		*Estimated Average Fan Power for Air-Handler Supply Fans for Base Case Test Period (kW)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	5.4*	5.8*
Up	16%	10%	6.3	6.4
Down	26%	11%	6.8	6.4
Up & Down	37%	28%	7.4	7.4

*base case averages.

The increase in air-handler supply fan power is driven by the increased airflow required to meet thermal conditions for a given leakage case. Following the trend of the weather-corrected changes in supply airflow required (Table II-6), the greatest increase in fan power is for the upstream and downstream leakage case (37% all day and 28% peak). The downstream leakage case (26% all day and 11% peak) requires a greater increase in fan power than the upstream leakage case (16% all day and 10% peak). The base case average fan power ranges from 5.4 (all day) to 5.8 kW (peak). If the system were to operate in the leakage cases over the same time period, we estimate that the average fan power would range from 6.3 to 7.4 kW (all day) and from 6.4 to 7.4 kW (peak).

Induction Fan Power: The EMCS adjusts the amount of air delivered to a given zone via the VAV box's primary air damper, based on the zone temperature reading and throttling range set point. If the amount of primary airflow entering the VAV box is lower than a specified threshold, in this case less than 40% of the box's cooling maximum airflow, the EMCS turns on the induction fan in the VAV box. The induction fan pulls air from the ceiling plenum (return plenum) into the VAV box.

Table II-11 summarizes the average fraction of induction fans operating during each test period. When there is a higher amount of downstream leakage, the powered VAV boxes operate at higher primary airflows to compensate for the leakage and the induction fans do not run as often. This decrease in induction fan on-time contributes to lower induction fan power required to meet thermal conditions.

Table II-11. Normalized average fraction of induction fans operating.

Case	*% Change in Fraction of Induction Fans Operating*		*Estimated Average Fraction of Induction Fans Operating for Base Case Test Period*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	0.72*	0.63*
Up	0%	3%	0.72	0.65
Down	-37%	-23%	0.45	0.49
Up & Down	-43%	-36%	0.41	0.40

*base case averages.

On average during the base case, the induction fans were running 72% (all day) and 63% (peak). If we were to operate the system in the leakage cases over the base case time period, we estimate that the percentage of induction fans running would increase slightly for the upstream case (72% all day and 65% peak), but decrease for the downstream case (45% all day and 49% peak) and the upstream and downstream case (41% all day and 40% peak).

The induction fans (on average, 1.3 kW all day and 1.2 kW peak for the base case) use about one-fifth as much fan power as the air-handler supply fans. The reduction in the fraction of induction fans operating translates directly into lower induction fan power needed to meet the thermal requirements of each leakage case (see Table II-12). The downstream case requires 38% (all day) and 24% (peak) less induction fan power and the upstream and downstream case requires 45% (all day) and 38% (peak) less induction fan power than the base case. For the base case period, this reduces the induction fan power to an estimated 0.8 kW (all day and peak) for the downstream case and an estimated 0.7 kW (all day and peak) for the upstream and downstream case.

Table II-12. Normalized average fan power (kW) for induction fans.

Case	*% Change in Induction Fan Power*		*Estimated Average Induction Fan Power for Base Case Test Period (kW)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	1.3*	1.2*
Up	0%	3%	1.3	1.2
Down	-38%	-24%	0.8	0.8
Up & Down	-45%	-38%	0.7	0.7

*base case averages.

Air-Handler Supply and Induction Fan Power: The increases in air-handler supply fan power, seen in Table II-10, are tempered by the decrease in induction fan power, seen in Table II-12, resulting in a net increase in air-handler supply and induction fan total power of 13% (all day) and 9% (peak) for the upstream case, 17% (all day) and 6% (peak) for the downstream case, and 26% (all day) and 19% (peak) for the upstream and downstream case (see Table II-13).

Table II-13. Normalized average fan power (kW) for air-handler supply and induction fans.**

Case	*% Change in Air-Handler Supply and Induction Fan Power*		*Estimated Average Air-Handler Supply Fan and Induction Fan Power for Base Case Test Period (kW)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	6.7*	6.9*
Up	13%	9%	7.6	7.5
Down	17%	6%	7.8	7.3
Up & Down	26%	19%	8.4	8.2

*base case averages.

**Because the total supply and induction fan power values are the averages based on the sum of the coincident fan power measurements, the sum of the average individual supply and induction fan power may not add to be the average total supply and induction fan power results.

For the base case, the net average air-handler supply fan and induction fan total power is 6.7 kW (all day) and 6.9 kW (peak). We estimate that, by operating the leakage cases in the base case time period, the net average air-handler supply fan and induction fan total power would increase to 7.6 kW (all day) and 7.5 kW (peak) for the upstream case, 7.8 kW (all day) and 7.3 kW (peak) for the downstream case, and 8.4 (all day) and 8.2 (peak) for the upstream and downstream case.

Fan Power Metrics: As described in Appendix I, there are several metrics that can be used to characterize the performance of thermal distribution systems. We have focused here on those related to system fan power. These include specific fan power (W/cfm), normalized fan power (W/ft^2), and fan-airflow density (cfm/ft^2).

Specific Fan Power (W/cfm): The specific fan power (W/cfm) is calculated, for constant volume systems, as the total fan power divided by the total delivered airflow at the supply grilles. This metric accounts for the fan power only and does not reflect any thermal losses in the duct system. As such, it quantifies the amount of fan power required to deliver a given quantity of air through the duct system to the space or building. This parameter is relatively constant in a constant-air-volume system, because the fan speed and the airflow to the grilles remain constant for a given operating state (heating, cooling, or ventilation).

This is not true for the test building, which has a variable-air-volume system with variable frequency drive fans and powered induction boxes. In this system, the amount of air delivered to each grille and the amount of power required to drive the supply, relief, and induction fans can vary widely throughout the day depending on zone conditioning load requirements. Although we have measured air-handler supply and induction fan power on a minute by minute basis, measuring the airflow delivered to each grille on the same basis would require a level of airflow monitoring that is beyond the scope of this project. Instead, we calculated the specific air-handler fan power value as the power for the air-handler supply fans divided by the airflow through the air-handler supply fans (See Table II-14). This parameter can be used to compare air-handler fan power for our various leakage cases.

Table II-14. Specific fan power (W/cfm) for air-handler supply fans.**

Case	*% Change in Specific Fan Power*		*Estimated Specific Fan Power for Base Case Test Period (W_{fan}/cfm)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	0.50*	0.51*
Up	4%	4%	0.52	0.53
Down	7%	5%	0.54	0.54
Up & Down	9%	6%	0.55	0.54

*base case averages.

**based on airflows through the air-handler supply fans and not the airflows at the grilles.

Because the air-handler supply fan power increases as the air-handler supply fan airflow increases, the changes in specific fan power for the air-handler supply fan are relatively low, increasing 4% (all day or peak) for the upstream case, 7% (all day) and 5% (peak) for the downstream case, and 9% (all day) and 6% (peak) for the upstream and downstream case.

For the base case, the specific fan power is 0.50 W/cfm (all day) and 0.51 W/cfm (peak). The specific fan power increases slightly to range from 0.52 to 0.55 W/cfm for the leakage cases.

Normalized Fan Power (W/ft^2): This metric reflects the impacts of duct leaks and thermal losses and is primarily used as a metric for constant volume systems. For this building with a VAV system, we have evaluated this metric based on how it was operated during our test periods.

Based on the operation of the building during our test periods, Tables II-15 through II-17 summarize normalized fan power for the air-handler supply fans, the induction fans, and the net supply and induction fan totals. The base case normalized fan power is 0.18 W/ft^2 (all day) and 0.20 W/ft^2 (peak) for the air-handler supply fan, 0.05 W/ft^2 (all day) and 0.04 W/ft^2 (peak) for the induction fans, and 0.23 W/ft^2 (all day) and 0.24 W/ft^2 (peak) for the net total of the air-handler supply and induction fans. These values change in the same way as the fractional changes for the fan powers (Tables II-10, II-12, and II-13) and the estimated values range from 0.21 to 0.26 W/ft^2 for the air-handler fans, 0.02 to 0.05 W/ft^2 for the induction fans, and 0.25 to 0.29 W/ft^2 for the net total air-handler supply and induction fans.

Table II-15. Normalized fan power (W/ft^2) for air-handler supply fans.

Case	*% Change in Normalized Fan Power for Air-Handler Supply Fans*		*Estimated Normalized Fan Power for Air-Handler Supply Fans for Base Case Test Period (W/ft^2)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	0.18*	0.20*
Up	16%	10%	0.21	0.22
Down	26%	11%	0.23	0.22
Up & Down	37%	28%	0.25	0.26

*base case averages.

Table II-16. Normalized fan power (W/ft^2) for induction fans.

Case	*% Change in Normalized Fan Power for Induction Fans*		*Estimated Normalized Fan Power for Induction Fans for Base Case Test Period (W/ft^2)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	0.05*	0.04*
Up	0%	3%	0.05	0.04
Down	-38%	-24%	0.03	0.03
Up & Down	-45%	-38%	0.03	0.02

*base case averages.

Table II-17. Normalized fan power (W/ft^2) for air-handler supply and induction fans.

Case	*% Change in Normalized Fan Power for Air-Handler Supply Fans and Induction Fans*		*Estimated Normalized Fan Power (Air-Handler Supply Fans and Induction Fans for Base Case Test Period (W/ft^2)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	0.23*	0.24*
Up	12%	9%	0.26	0.26
Down	17%	6%	0.27	0.25
Up & Down	26%	19%	0.29	0.29

*base case averages.

Fan-Airflow Density (cfm/ft^2): For a constant-air-volume system, the fan-airflow density metric (cfm/ft^2) can be determined by measuring the floor area and the total airflow delivered to each zone under certain operating conditions and by calculating the airflow per unit floor area in each zone. This metric reflects the impacts of thermal losses while not discounting duct leakage in a significant way.

For a VAV system, the amount of monitoring required to calculate this metric during occupied hours would be excessive. Rather than report an artificial value calculated by setting the system into a constant volume mode, we have redefined the fan-airflow density metric here to be the airflow measured at the air-handler supply fans divided by the floor area. As can be seen in Table II-18, this value is affected by the amount of duct leakage seen by the system and increases at the same rate as the air-handler supply fan airflows listed in Table II-6: 11% (all day) and 6% (peak) for the upstream case; 18% (all day) and 7% (peak) for the downstream case; and 28% (all day) and 22% (peak) for the upstream and downstream case.

Table II-18. Fan-airflow density (cfm/ft^2) for air-handler supply fans.

Case	*% Change in Fan-Airflow Density for Air-Handler Supply Fans*		*Estimated Fan-Airflow Density for Air-Handler Supply Fans for Base Case Test Period (cfm/ft^2)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*	*All Day (3 am to 6 pm)*	*Peak (2 pm to 6 pm)*
Base	-	-	0.37*	0.39*
Up	11%	6%	0.41	0.41
Down	18%	7%	0.44	0.42
Up & Down	28%	22%	0.47	0.48

*base case averages.

For the base case, the fan-airflow density is 0.37 cfm/ft^2 (all day) and 0.39 cfm/ft^2 (peak). Our estimates for the leakage cases operating during the base case range from 0.41 cfm/ft^2 to 0.48 cfm/ft^2.

Zone Air Temperatures: The presence of additional duct leakage did not change the average zone air temperatures significantly, as determined by the thermostat data obtained using the EMCS (see Table II-19). Looking at the six zones in which we added duct leakage downstream of the VAV boxes, we found that temperatures remained within 0.1°F between cases and well within thermal comfort requirements. This is due to the additional fan power available to move conditioned air to meet the comfort requirements of the space.

Table II-19. Normalized average zone air temperatures (°F).

Case	*Estimated Average Zone Temperature for Base Case Test Period (°F)*	
	All Day (3 am to 6 pm)	*Peak (2 pm to 6 pm)*
Base	73.9*	73.9*
Up	73.8	73.8
Down	73.9	73.8
Up & Down	73.9	73.8

*base case averages.

Fan Power Trends: Figures II-13 and II-14 show general daily fan power trends using two sample days: a "Base Case" day (no added duct leaks) and an "Up & Down" day (with upstream and downstream duct leaks installed). Note that one might consider assessing duct leakage effects on fan power by simply comparing the daily fan power trends for the two cases. This approach does not account for occupancy and weather differences between cases.

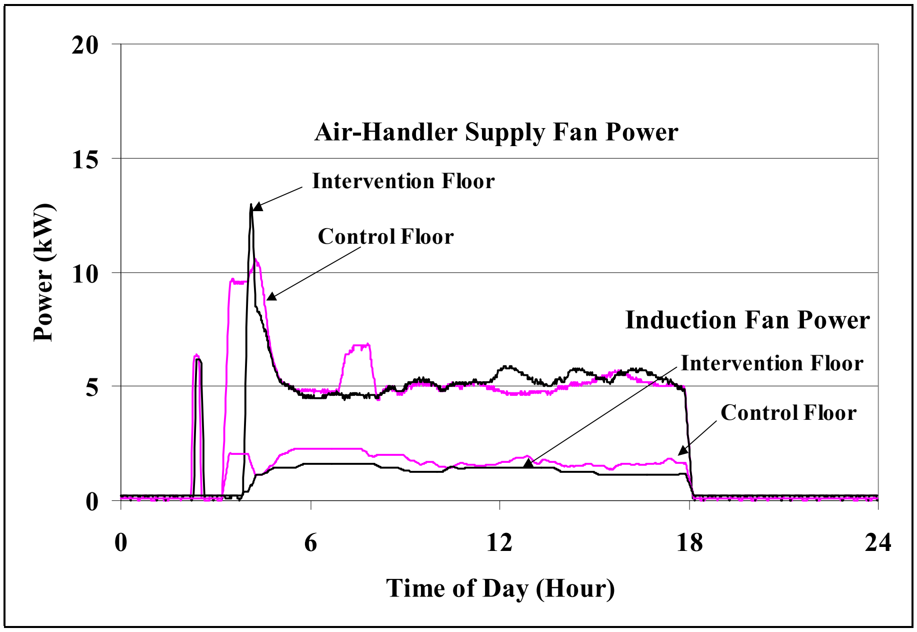

Figure II-13. Air-handler supply fan and induction fan power for a sample "Base Case" day.

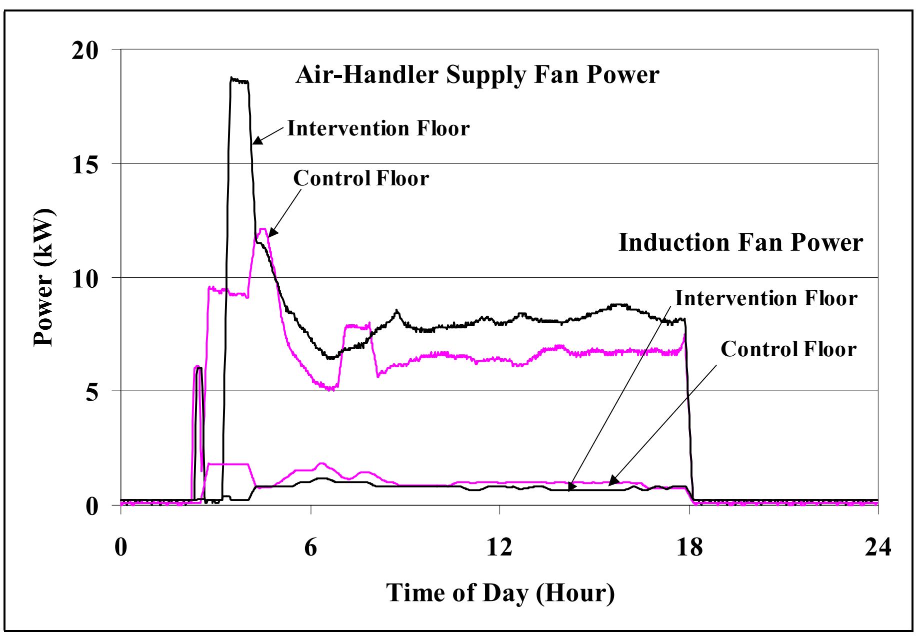

Figure II-14. Air-handler supply fan and induction fan power for a sample "Up & Down" day.

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Appendix III. Duct Sealing Techniques for Large Commercial Buildings

Duo Wang, Craig Wray and Rick Diamond
Lawrence Berkeley National Laboratory
Berkeley, California

Ducts Sealing Techniques for Large Commercial Buildings

Introduction

Our duct leakage intervention study (described in Appendix II) showed that the energy impact of upstream and downstream duct leakage in large commercial buildings with variable-air-volume (VAV) HVAC systems can be substantial. Therefore, if duct systems in this large commercial building population are leaky, then it is important to have effective duct sealing methods that can be readily commercialized and applied.

As found, the one-year old test building that we studied in this project showed every indication of a "tight" thermal distribution system: good application of mastic, metal bands at joints, and overall high quality installation. We found that ducts both upstream and downstream of VAV boxes were sealed, with a total fractional leakage flow of about 5% at operating conditions. An example of the sealing detail was that mastic had even been applied to the inside of plenum takeoff joints, downstream of the VAV boxes.

We have also measured fractional leakage flows in four other California buildings (Xu et al. 1999, 2002). Two of the buildings were built in 1996 and had leakage fractions similar to the one-year old test building. However, the other two were much more leaky: one built in 1979 had a leakage fraction of about 17%; the other (built in 1989) had a leakage fraction of about 25%. The duct sealants used and their location in the four older buildings are not well documented, but we know that mastic was used on some duct joints in the building that was built in 1989.

Although this set of leakage flow data is very limited, it has two significant implications for California buildings:

1. The *tight* ducts are evidence that at least some HVAC contractors in the California building industry already know how to effectively seal ducts in new large commercial buildings, even though specifications for duct leakage airflows tend to be poorly defined for new construction (especially for the lower-pressure-class duct sections that are located downstream of VAV boxes).
2. The *leaky* ducts are evidence that the California building industry needs to consider sealing ducts in existing large commercial buildings. However, sealing ducts in existing buildings is more challenging than in new buildings, because of reduced access to ducts after ceiling panels are in place and spaces are occupied. Remote sealing techniques that reach duct leaks without having to access and seal every joint manually are preferable in these cases.

Over the past decade, Lawrence Berkeley National Laboratory has developed an internally applied aerosol-based technology to seal leaky ducts remotely. As a commercialized technology, contractors have actively used it to seal thousands of duct systems in houses and hundreds of duct systems in small commercial buildings. Since 1997, we have begun investigating how to apply similar technology in large commercial buildings (Modera et al. 2001, Carrie et al. 2002).

The aerosol sealing process involves the separate but simultaneous injection of carrier air and aerosol sealant into the duct system, with intentional duct openings (e.g., grilles, fans) blocked off and sensitive components isolated. An integrated fan and airflow meter connects to the duct system to provide and measure carrier airflow from the room to the duct system. A sealant injector is inserted into the duct system downstream of the carrier air injection point. The injector's sealant pump supplies room temperature liquid sealant[4] through tubing to the injector; a separate stream of heated compressed air mixes with the liquid sealant at the injector nozzle discharge. Sealant particles are produced by atomization and drying of the liquid stream exiting the injector nozzle. The carrier airflow transports aerosol sealant particles to the leaks, where the particles collide with leak edges, accumulate, and ultimately seal the leaks.

Continuously monitoring the duct wall pressure difference and carrier airflow allows one to continuously determine the duct effective leakage area (ELA_{25}) and sealing rate[5], and to track sealing progress. As leaks are sealed, the ELA_{25} decreases (duct flow resistance increases), the duct wall pressure difference increases, and the carrier airflow (exiting through the leaks) decreases. The sealing rate also decreases, because the particle transport in the carrier air decreases. Changes in these parameters also serve to identify problems. For example, sudden changes in duct pressure difference and airflow can indicate that a grille covering or isolation blockage has failed.

When the ELA_{25} is relatively small (with associated high duct wall pressure difference), or the sealing rate becomes low and stable, sealing should be terminated. For example, aerosol sealing in residential or light commercial duct applications is usually terminated when the duct wall pressure difference reaches 500 Pa, at which point the ELA_{25} is often between 10 to 20 cm^2. We expect that similar termination criterion would be used when sealing ducts in a large commercial building.

Applying the existing single-injector aerosol technology to seal ducts in large commercial buildings is problematic, because these duct systems are much larger and more complex than duct systems in residential and small commercial buildings. For example, a typical supply-air duct system in a large commercial building has a fan blowing air into a main trunk duct, with a VAV box connected to each trunk outlet; ducts downstream of each VAV box branch off to supply grilles. The result is that the large commercial supply duct systems are long (several hundred feet) and it is difficult to inject sealant efficiently from a single point all the way to leaks near the end of the duct system. In addition, there are sensitive components such as heater coils and backdraft dampers inside VAV boxes. To maintain their functionality, these components should not be exposed to aerosol sealant.

Consequently, to reduce the distance that sealant must travel from injection to leakage sites, and to avoid spraying sealant into VAV boxes, the aerosol-sealing technology needs to use two injection stages. One stage involves sealing the main trunk of the duct system with the primary air inlets to the VAV boxes closed and with multiple injectors located along the length of the duct. The other stage involves sealing the numerous duct branches downstream of the VAV boxes. Both these configurations mean that being able to seal multiple sections simultaneously and being able to move injectors quickly from one injection site to the next would reduce the time required to seal the entire system.

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[4] The sealant is a water-based vinyl-acetate polymer liquid adhesive, with a concentration of 120 µg of solid adhesive per mL of liquid.

[5] ELA_{25} is defined as the cross-sectional area of a perfect nozzle that would produce the same flow as the total measured airflow through the leaks (carrier airflow), but at a reference duct wall pressure difference of 25 Pa. The sealing rate is indicated by the reduction in ELA_{25} per unit time.

The remainder of this appendix describes the development and laboratory testing of a mobile aerosol-sealant injection system (MASIS) that can use multiple injectors simultaneously to seal multiple duct sections. To help the reader understand the multiple injector system, we have included a description of the aerosol sealing technology. DOE funded the development of the injector.

The three key elements in the development and laboratory testing of MASIS were:

1. Develop protocols (plans) to seal ducts upstream and downstream of VAV boxes.
2. Refine injection system components for use in field applications.
3. Test the automated monitoring system that measures process pressures and airflows.

At the end of this appendix, we also discuss whether there is a need to develop field retrofit techniques for sealing duct system components such as VAV boxes and supply grilles.

Multiple-Injection Aerosol Sealing Technology

Technology Overview: We initially developed the mobile aerosol-sealant injection system as a laboratory prototype. Each aerosol injection station of MASIS consists of a cart with a liquid sealant tank, a peristaltic pump, and an electrical control box; a sealant tube attaches the pump to an aerosol sealant injector. A compressed air hose and electrical wiring for the injection heater are also attached to the injector.

Figure III-1 shows a schematic of the multiple injection system. Figures III-2 and III-3 show the cart and control box, as well as a schematic of the electrical circuitry in the control box.

Aerosol Sealant Injector Details: Figure III-4 shows an aerosol sealant injector; Figure III-5 illustrates the injector components schematically.

The aerosol injector stem is a copper pipe ($1^3/_8$ inch O.D.), which contains the liquid sealant line and the compressed air line with its electrical heater (110 V, 400 W). The top of the stem has a cap on which the injector nozzle is fastened. The purpose of the heater is to heat the compressed air so that it will evaporate the water in the atomized liquid sealant.

The injector nozzle is an external-mix atomizing nozzle (Model 970 S4, Düsen-Schlick GmbH). Figure III-6 shows a nozzle, and includes schematics that show the parts of the nozzle in more detail. Liquid sealant flows through the center of the nozzle's liquid insert; heated compressed air flows between the air cap and the outside of the liquid insert. The inner diameters of the liquid insert and air cap discharges are 1.0 mm and 2.6 mm respectively; the liquid insert protrudes 1.2 mm from the air cap. The spray angle of the nozzle is 10 to 15 degrees; this small spray angle is important because it reduces the amount of sealant deposition on the duct walls and increases the sealing rate.

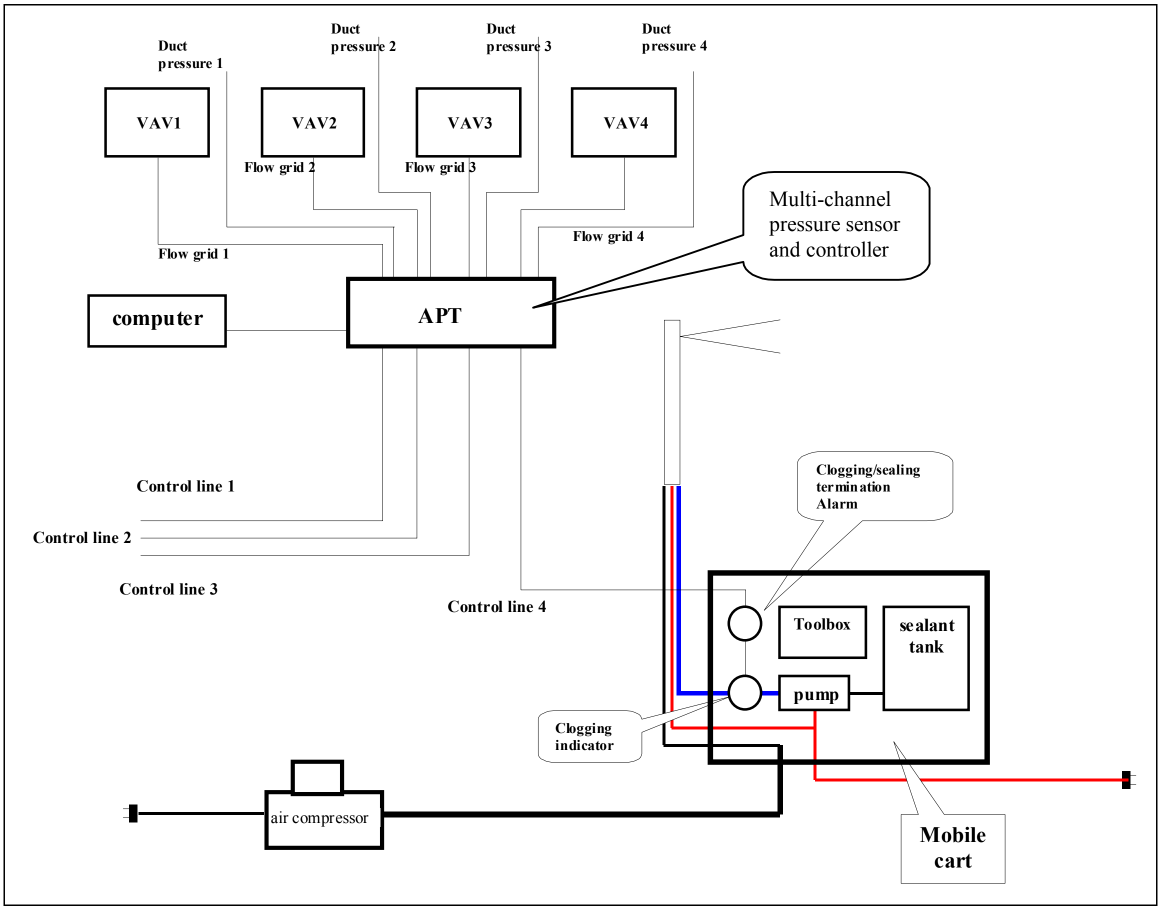

Figure III-1. Schematic of MASIS. This figure shows only one cart (lower right), but the schematic assumes that up to four injectors (each with a cart) could be used simultaneously, all controlled by a single computer.

Figure III-2. MASIS cart and injection system control box.

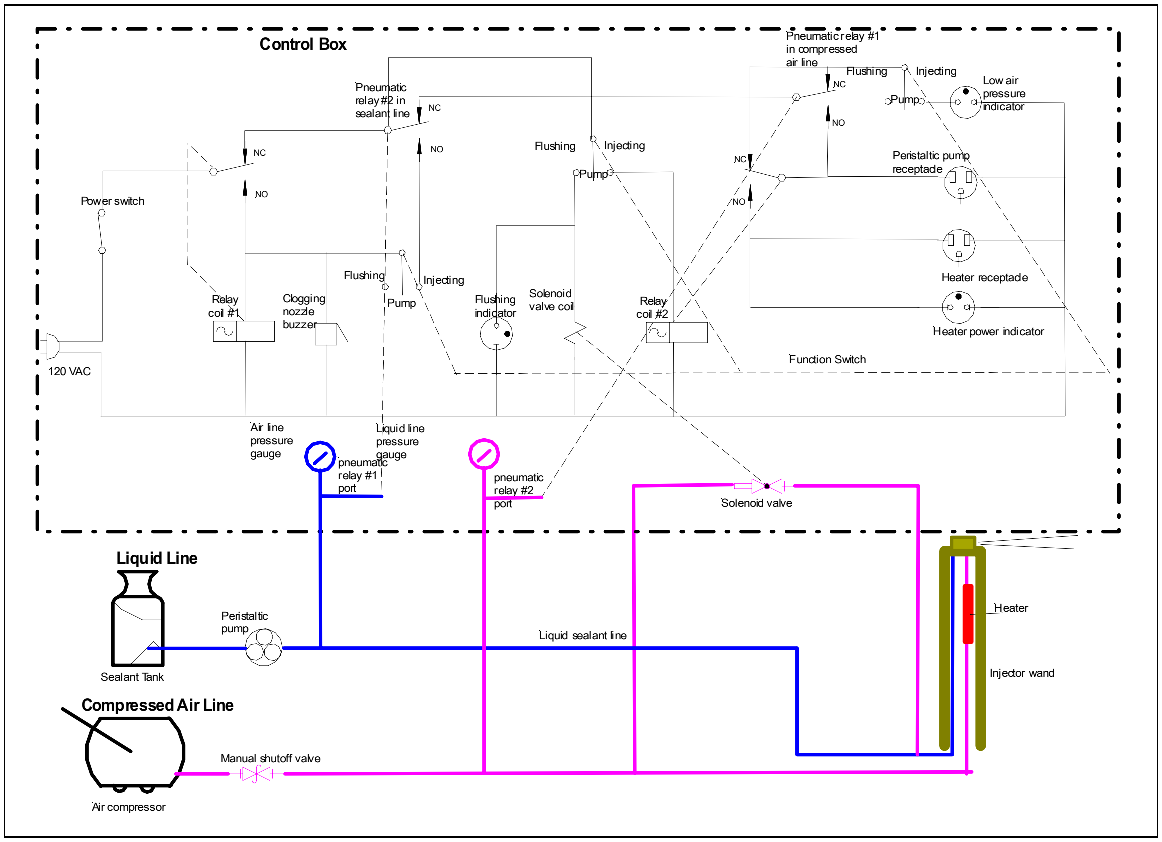

Figure III-3. Schematic of MASIS control box electrical circuitry, and connections to other injection system components.

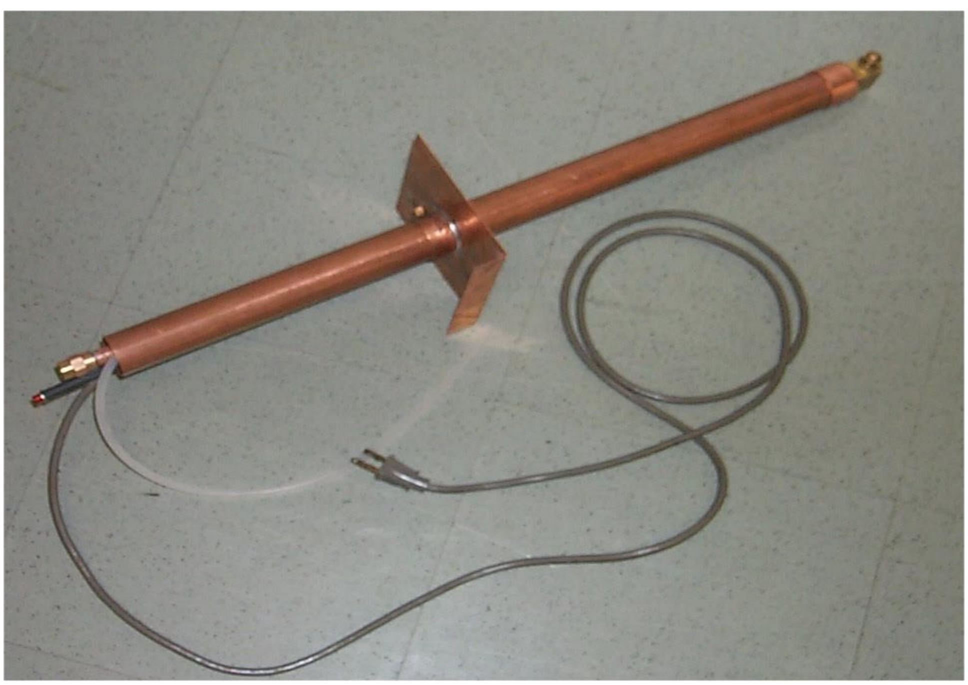

Figure III-4. Aerosol sealant injector.

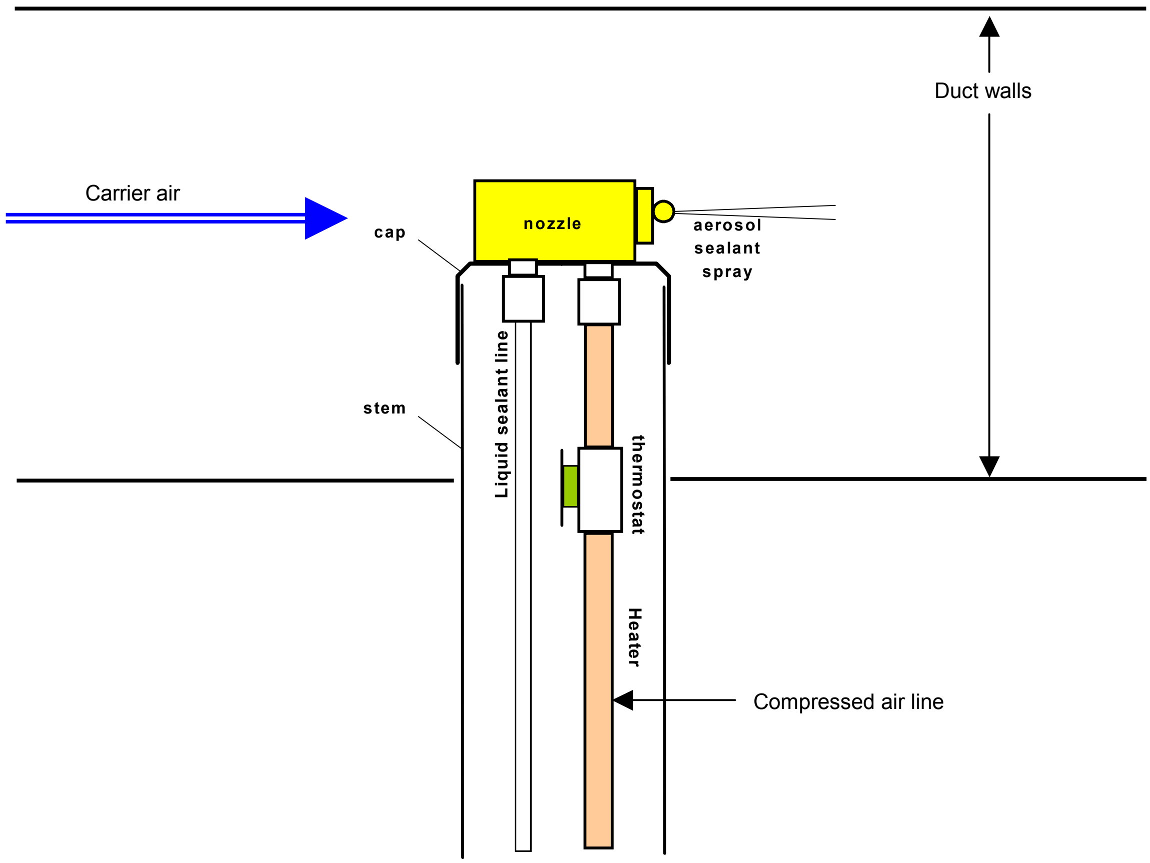

Figure III-5. Schematic of aerosol sealant injector, shown installed in a duct.

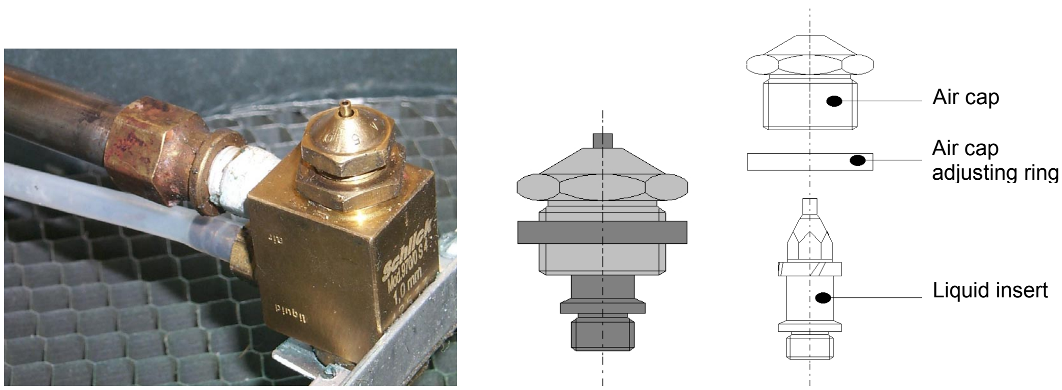

Figure III-6. Aerosol sealant injector nozzle, with schematics of components.

Site-Specific Planning for Duct Sealing

- Generate a duct map, which shows the location of all the VAV boxes and the location of all grilles associated with each VAV box.
- Develop a protocol to isolate the system fan and VAV boxes from the trunk; if the VAV box primary air dampers need to be closed manually, develop a protocol for that process.
- With assistance from the building engineer, identify and locate all sensitive components (e.g., smoke sensors, dampers); develop a protocol for protecting the components.
- Determine where and how to install the sealing fan with airflow meter.
- Determine how many aerosol injectors are needed and where to install them.
- Mark the locations of components requiring intervention or related to the sealing process (duct system and injection system) on duct map; also mark the related ceiling access panels.

Protocol for Sealing the Trunk Upstream of VAV Boxes

1. Turn off the system fan (if on) and isolate the system fan and VAV boxes from the system.
2. Protect sensitive components as needed.
3. Connect the sealing fan and airflow meter to the trunk system.
4. Setup the eight-channel Automated Performance Testing system (APT) and a computer to monitor the airflow and the duct wall pressure difference for the trunk duct.
5. Using the sealing fan with its airflow meter, determine the trunk ELA_{25}.
6. If the trunk is very leaky, inspect the trunk for large leaks and manually seal any that are found. Determine the ELA_{25} again if any sealing was carried out.
7. If further sealing is needed, install the aerosol sealant injectors. Otherwise, skip forward to the downstream sealing process.
8. Turn on all the injectors, monitor the duct wall pressure difference and carrier airflow changes, and continuously determine ELA_{25} as sealing proceeds.
9. Terminate the sealing once the trunk ELA_{25} stabilizes.

Protocol for Sealing Branch Ducts Downstream of VAV Boxes

Sealing multiple VAV branch ducts means there are multiple independent but simultaneous tasks. In this stage of the sealing process, MASIS can seal up to four branches at the same time. For each branch, all the associated grilles are temporarily covered and the associated primary air damper is opened. The sealing fan supplies the carrier airflow through the associated VAV box to the downstream leakage sites, and an aerosol injector injects sealant downstream of the box. The ELA_{25} of the branch duct is based on the airflow entering the associated VAV box (measured using the box flow grid); it is not based on the carrier airflow, which is the sum of the airflows supplied to the multiple downstream sections that are being sealed. The pressure difference across the duct wall is still measured in the trunk.

As part of this process, it is necessary to calibrate the VAV box flow grids. Assuming that all flow grids behave similarly and that relative changes in ELA_{25} are more important than the absolute values, then it is necessary to only calibrate one grid and the same calibration can be used for all other grids. The simplest calibration involves a one-point test. The test involves opening one primary air damper after the trunk is sealed; the airflow through the box grid will then be the carrier airflow minus the leakage flow of the trunk. That leakage flow is estimated using the trunk ELA_{25} and duct wall pressure difference. Note that pressure-dependent VAV boxes do not have flow grids installed; a probe to determine box airflow will need to be inserted in each branch being sealed.

The following describes the protocol for sealing up to four downstream sections at a time. This protocol is repeated on other downstream sections as needed.

1. Cover all of the grilles on the downstream sections being sealed.
2. Open the primary air damper of one VAV box; calibrate the airflow through the associated box flow grid.
3. Open the other primary air dampers for the downstream sections being sealed.
4. Determine the ELA_{25} of each branch duct being sealed.
5. If a branch duct is very leaky, inspect the duct for large leaks and manually seal any that are found. Determine the ELA_{25} again if any sealing was carried out.
6. For each branch duct being sealed, if further sealing is needed, install the aerosol sealant injector downstream of the VAV box. Otherwise, skip forward to Step 9 of the downstream sealing process.
7. Turn on the injectors. As sealing proceeds, monitor the duct wall pressure difference and the branch duct airflow changes, and continuously determine the ELA_{25} for each branch duct being sealed.
8. For each branch duct, when its ELA_{25} stabilizes, terminate the sealing, uninstall the injector, close the primary air damper, and uncover the associated grilles.
9. Move to the next unsealed branch. Repeat the protocol from Step 3 onward. If all branches are sealed, continue to Step 10.
10. Restore the system and turn on the system fan (if it was found on).

Refining MASIS for Field Use

The use of the MASIS injection system in the laboratory and field differs in two ways:

1. Lab tests use short compressed air hoses connected to nearby fixed, high-capacity air supplies; field use requires a portable compressed air supply and longer air hoses.
2. In a lab, it is easy to clean clogged injector nozzles with solvents and the time to do so is less critical; field cleaning of nozzles is more difficult because of limited facilities to accommodate solvents and the time available for cleaning is often limited.

The following describes our refinements to MASIS for field use.

Compressed Air Supply: For field use, each injection station requires a portable air compressor. The compressed air pressure for liquid sealant atomization is 50 psi and the airflow is 3 to 5 cfm. The compressor needs to operate on the 110 V electrical supplies commonly available in buildings and is limited to single-stage devices for portability. Compressors that meet these requirements are commercially available, but our flow and pressure requirements are near their capacity limits.

The compressor could be mounted on the injection cart to improve system portability. However, compressors are noisy, so locating the compressor 50 to 100 feet away from the work area is preferable. This means that compressed air hoses of this length with a low pressure drop are needed. To determine an appropriate hose size for the compressed air lines, we calculated the pressure drops for the flow range that we use, based on a roughness factor 0.0004 feet. For 3/8" I.D. air hose, the pressure drops range from 5 to 13 psi per 100 feet of hose. For 1/2" I.D. air hose, the pressure drops are much lower: 1 to 2 psi per 100 feet of hose. To take advantage of the smaller pressure drops, we selected 1/2" I.D. hose and limited the length to 50 feet for portability reasons.

Nozzle Clogging: The heated compressed air that dries the sealant after it leaves the injector nozzle has an undesirable side effect: it also heats up the whole nozzle assembly. The increased nozzle temperature causes sealant to gradually deposit on the wall of the liquid insert inside the nozzle. When enough sealant accumulates in the insert, it becomes clogged and the sealant flow stops.

Lab tests under our DOE research program addressed the clogging issue. One technique that we identified and applied under that program was to use compressed air to flush the nozzle after an injection sequence is complete. A bypass valve was installed between the liquid and compressed air lines. To flush the nozzle, it is necessary to turn off the sealant pump and heater and open the bypass valve simultaneously. To avoid rupturing the sealant lines and to prevent backflow through the pump from the compressed air introduced into the sealant lines, we upgraded the peristaltic pump and lines. The new pump can provide 63 ml/min of sealant flow at 100 psi. The pump has a metering capability and can be set to any desired flow, within a 10:1 turndown ratio. Our lab tests indicate that the pump and flushing system are effective in providing the desired range of sealant flows (20 to 50 ml/min) for double the number of injection sequences before clogging (10 sequences instead of 5). Further work funded by DOE is underway to develop a nozzle that is less susceptible to clogging.

Testing the Sealing Process Monitoring System

MASIS uses an auto-zeroing eight-channel pressure transducer (Automated Performance Testing system, APT) to measure the duct wall pressure differences, as well as the flow meter pressure differences used to determine leakage airflows. Using custom software that we developed to monitor up to four simultaneous sealing processes, a laptop computer connected to the APT displays the duct wall pressure differences and carrier airflows, calculates the effective leakage areas (ELA_{25}), and determines the sealing rates versus elapsed time. Both the upstream and downstream sealing stages use the same centralized monitoring system. Although sealing each of the VAV branches in the second stage is independent and separate monitoring stations could be used, it is advantageous for capital cost and efficiency reasons to use the same centralized monitoring system in both stages. We have extensively tested the software and monitoring system in our laboratory and determined that this system is fully functional. Figure III-7 shows sample results from a single-branch duct-sealing test in our laboratory.

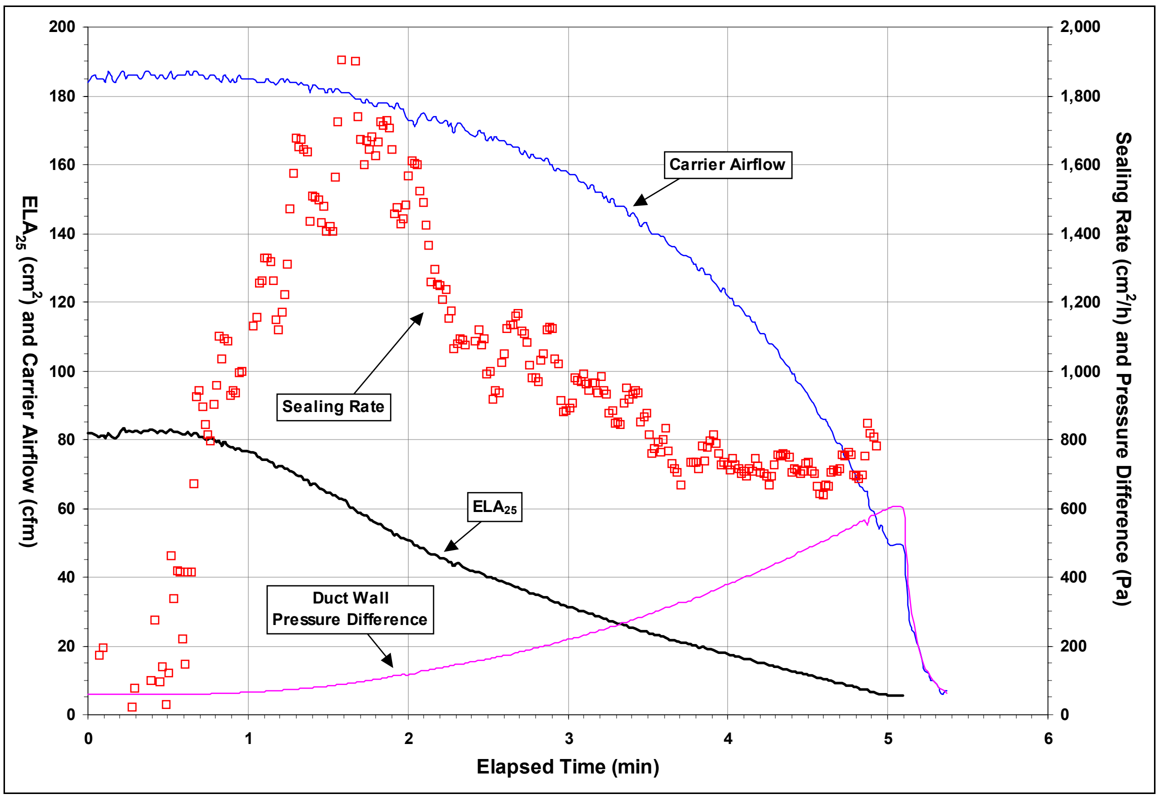

Figure III-7. Sample results from a single-branch duct-sealing laboratory test. The injector was turned off after running about 5 minutes; the carrier airflow fan was turned off about 5 seconds later. The effective leakage area (ELA_{25}) of the duct section was reduced about 94% over the 5-minute period.

Other Duct Sealing Issues

We considered other duct sealing issues in this project, but after further laboratory and field investigations, discounted them in terms of field retrofits. One example involves

VAV box air leakage. Our laboratory tests of one VAV box with an induction fan indicated that about 30% of the box leakage is across the partition separating the primary air path from the induction fan inlet, and about 70% of that is associated with induction fan backdraft damper leakage. Our field tests indicated damper leakage could be even larger (2 to 3 times greater), but it seems that damper leakage may not be a significant issue when compared with other duct leakage. In any event, field retrofits to reduce backdraft damper leakage would be difficult because of limited access and aerosol sealing cannot be used to seal the damper edges (the damper needs to open when the induction fan operates). Providing better sealing for the backdraft damper and partition appears to be a design and manufacturing issue more so than an installation or field retrofit issue.

Another example involves air leakage at supply grille edges. Our component leakage tests in the test building indicated that a substantial fraction of the leakage area in downstream duct sections is located at the supply grille edges. However, our leakage flow tests with and without grille edge seals indicated that this leakage area is of little concern, likely because operating pressures at these leaks are very low.

Next Steps

Given the definition of aerosol sealing protocols here and the positive results from our lab tests, we are ready to deploy the multiple injector aerosol-based sealing technology to seal duct leaks in large commercial buildings. The next step is to demonstrate its performance in a sample of large commercial buildings.

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Appendix IV. Production Readiness Plan for Retrofitting Thermal Distribution Systems in Commercial Buildings

Rick Diamond, Craig Wray, Darryl Dickerhoff, Nance Matson and Duo Wang

Lawrence Berkeley National Laboratory
Berkeley, California

Production Readiness Plan for Retrofitting Thermal Distribution Systems in Large Commercial Buildings

The primary outcome from this project is the measured confirmation that duct leakage airflows can significantly increase fan energy consumption in large commercial buildings. In addition, we have defined a new metric for distribution system efficiency, demonstrated a reliable test for determining duct leakage, and developed new techniques for duct sealing. The parallel story in the residential and small commercial sector has shown that from the comparable stage in that research to maturity of technology adoption (e.g., commercialization and inclusion in standards) was approximately ten years. We conclude that a concerted effort will be necessary in order to make the same—or better—progress for the large commercial building sector.

Market transfer work to date:

1. **Codes and Standards**. California's Title 24 currently has no performance criteria for thermal distribution systems in large commercial buildings. We have identified a metric, transport energy per unit thermal energy delivered, for characterizing distribution system efficiency that the Title 24 compliance process should include for all new large commercial buildings. Once we have a good understanding of the range of distribution system efficiencies, we could then set guidelines for minimum acceptable levels.
2. **CA Public Sector**. University of California staff is already asking for the development of duct tightness criteria for the new buildings at UC Merced. They are also interested in the measurement and verification procedures to ensure that these criteria have been met. These criteria could be adopted for new construction at University of California and California State University campuses throughout the state.
3. **PIER-related activity**. Taylor Engineering, Eley Associates, and the Center for the Built Environment at UC Berkeley have all come on board this project as interested co-participants. The work at the test building in Sacramento has been a fertile testbed for several groups interested in sharing our monitoring capabilities to do unique measurements of HVAC systems in commercial buildings.
4. **Synergistic funding with US DOE**. This project has benefited from over $400k of support from the Building Technologies office at the U.S. Department of Energy. DOE plans to continue supporting work in this area, including continued efforts at the test building in Sacramento to further assess duct leakage diagnostics, as well as measurements of duct leakage at different sites.
5. **Work with the building services sector**. The engineering staff at the public building where we did the fieldwork for this study has been very interested and supportive of our work. They plan to use the findings from this study to better operate their showcase building.

In addition to the market transfer work identified above, there is one specific aspect of the study that has large commercialization potential: the aerosol sealing technology. We are currently negotiating a license agreement with a major commercial HVAC manufacturer for the use of the aerosol sealant in large commercial buildings. However, there are some development tasks that may be appropriate for the public sector to pursue, given the lack of R&D that is currently done in the building's sector. Once this work is done, we expect the private sector to fully commercialize this technology. The steps needed for its commercialization are as follows:

1. **Documentation of health and safety performance for the aerosol sealant**. UL has tested and approved the aerosol sealant for safety (e.g., fire, smoke) and there has been a preliminary review of the literature for possible health effects from the sealant material. A more extensive health review is planned, but not currently funded. Without this documentation of the health impacts, there may be reluctance on the part of building owners and managers to use the aerosol technology. Our attempt to demonstrate aerosol-sealing technology developed in this project encountered such a barrier.
2. **Characterization of the energy savings potential of existing buildings**. Very few buildings have had their distribution systems fully characterized. We know the potential savings from computer simulations, but these models have been based on assumptions of leakage amount and distribution. There is a need for measuring a sample of large commercial buildings to determine the actual range of leakage distributions. The U.S. Department of Energy currently plans to fund this work in 2003.
3. **Demonstration of aerosol sealing in a sample of commercial buildings**. The aerosol sealant has been demonstrated in residential and small commercial buildings. This project has led to further development of the equipment suitable for testing in large commercial buildings. The next step is to demonstrate the performance in a sample of commercial buildings.
4. **Transfer to private sector**. The University of California currently holds the patents for the aerosol sealing technology. If private interests license the technology, they will then carry out the necessary training of contractors and production of equipment to reach the market.

Our recommendations for future work are as follows:

Recommendation #1: Further develop the test method for determining leakage airflows in large commercial duct systems and submit it for adoption to ASTM (e.g., "Test standard for determining duct leakage flow in large commercial building systems").

Recommendation #2: Work with California's Title 24 staff to introduce a requirement for quantifying and reporting the "overall efficiency of the distribution system" metric for new large commercial buildings. Once we have a good understanding of the range of duct system efficiencies from reported data, we could then use these data to set guidelines for minimum acceptable levels.

Recommendation #3: Develop specifications for maximum allowable duct leakage airflows and for duct sealing in new construction.

Recommendation #4: Continue collaborative work with the U.S. Department of Energy, University of California, private sector (e.g., Carrier, Eley Associates, Taylor Engineering) to transfer information to the building industry.

Recommendation #5: Evaluate the performance of the thermal distribution system at the demonstration building over a heating season, with and without the added duct leakage. The investment of time and equipment at the demonstration building makes it worthwhile to continue monitoring the system in order to look at energy savings over the year.

Recommendation #6: Survey additional sites to start a database of duct leakage characteristics in large commercial buildings. This work is currently planned with funding from the U.S. Department of Energy and would benefit from co-funding by the CEC.

The final Project Advisory Committee meeting in November 2002 also generated 11 recommendations for further work. These recommendations are in the form of desired outcomes for improving thermal distribution systems in large commercial buildings, both new and existing, by 2010. Many of these outcomes reflect our recommendations, but they also represent a broader scope. The desired outcomes are as follows:

A. Stock Characterization and Energy Savings Potential

1. *Stock Characterization*. An assessment of thermal distribution systems in the large commercial building stock (e.g., magnitude and location of leakage airflows).
2. *Current Practice*. Characterization of existing practices for duct installation.
3. *Energy Impacts*. An expanded understanding of the energy impacts of thermal distribution system characteristics (e.g., impacts related to duct leakage and thermal conduction) and a ranking of the issues that warrant further study.

B. Design and Construction

1. *Design Guides*. Duct design and construction guidelines that focus on the most important issues in terms of their impacts on energy performance.
2. *Simulation Tools*. Mainstream simulation programs that can be used as design tools to predict distribution system performance.
3. *Technology Adoption*. Use of low-leakage duct components and joints, which will reduce or eliminate the need for widespread duct leakage testing.
4. *Specifications*. Specifications for achieving tight ducts within the normal building delivery process.
5. *Design Intent Linkage*. Improved communications between design intent, field construction, and operation.

C. Codes and Standards

1. *Metrics*. Further development of proposed metrics for system characterization (i.e., expanded definitions of what each metric includes, and how each is determined or measured).
2. *Standards*. Defined standards for distribution system installation.
3. *Test Procedures*. A standard test procedure for flow hoods.

D. Operations & Maintenance, Diagnostics, and Commissioning

1. *Commissioning Toolkit*. A toolkit for commissioning ducts.

2. *Real-Time Diagnostics*. A diagnostic method for measuring the energy use of distribution systems during operation, so that building operators can detect and rectify deficiencies in space conditioning energy delivery.
3. *Information Transfer*. Dissemination of our current knowledge to the critical players.