# DESIGN CONSIDERATIONS FOR MASTER KITCHEN EXHAUST SYSTEMS

**G.M. Elovitz, P.E.**
*Associate Member ASHRAE*

## ABSTRACT

*Master kitchen exhaust systems, consisting of multiple exhaust hoods in multiple tenant spaces all connected to a single main exhaust duct and fan, are commonly used in food courts in modern shopping centers. Developers favor master kitchen exhaust systems over multiple single-tenant systems because of flexibility, low first cost, reduced shaft space through upper floors, and ease of maintenance.*

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## INTRODUCTION

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Due to the fire hazards associated with greasy exhaust, there are several special challenges to successful design of master kitchen exhaust systems, especially sizing the main ductwork, achieving proper air balance, and fire protection. Since master kitchen exhaust systems are not adequately covered by existing codes and standards, designers, code officials, and fire departments are often unfamiliar with these challenges. This paper will discuss the design problems presented by master kitchen exhaust systems and propose some solutions.

## WHY MASTER KITCHEN EXHAUST

Many shopping centers have food courts, consisting of a group of independent fast food operators located close together. Master kitchen exhaust systems are often favored where the food court is not located directly beneath a roof, where individual tenants cannot install their own exhaust systems without passing through other tenant areas, or where the design of the roof will not permit a large number of individual small exhaust fans. For each master system, the owner installs an exhaust fan, a main exhaust riser to the food court area, and main ductwork to each tenant space, under the base building contract. Tenants then install their own exhaust hoods and individual branch ductwork from the hoods to the main duct. The owner also usually provides a master unconditioned make-up air system for tenant use.

The master exhaust system must be flexible in both the total exhaust airflow and the distribution of the exhaust within the food court. The main exhaust duct from the food court to the fan is usually sized and installed early in the project, based on an overall estimate of tenant requirements, long before individual tenants have completed their designs and often even before the merchandising plan has been finalized. The sizing of the main exhaust duct riser is forgiving of moderate errors in these projections; a riser sized at 1,800 fpm at the projected total exhaust airflow will accommodate actual airflows up to 20% more or less than the projected design and still be within the range of exhaust air velocities allowed by most codes.

It is usually more cost-effective to install one master system than multiple small systems. One large fan is usually less expensive to purchase and install than many smaller fans. Larger ducts generally have a higher ratio of cross section to perimeter, so less metal is needed to make one large duct than many small ducts with the same total capacity. Where ducts must pass from floor to floor, less shaft space is required for one large duct than many small ducts because of the required clearance between the duct and the shaft. Therefore, the master exhaust system "steals" less floor space on upper floors than multiple individual systems. Finally, it is usually less expensive to maintain one large fan and duct system than many smaller systems. All of these reasons attract shopping center developers to master kitchen exhaust systems.

## DISADVANTAGES OF THE MASTER KITCHEN EXHAUST SYSTEM

The main disadvantage of the master kitchen exhaust system is that the base building portions of the system (fan, main riser, and main ductwork to the tenant spaces)

**Gary M. Elovitz** is a consulting engineer with Energy Economics, Inc., Natick, MA.

must usually be designed and installed before individual tenants have completed their own designs. The designer of the master exhaust system must "guess" how much exhaust the tenants will need and how it will be distributed. This can lead to base building designs that do not match tenant requirements, either in total capacity or in distribution of the capacity over the tenant area.

If actual total kitchen exhaust requirements do not match the projected exhaust, the fan and duct system may be undersized or oversized. An oversized system is particularly problematic because it results in low air velocity in the main duct riser; air velocity below 1,500 fpm, the minimum permitted by most codes (Gladstone 1985), will not keep particles of grease entrained in the airstream. Grease buildup in the ductwork is a fire hazard. An undersized system leads to high air velocities, which increase noise, pressure losses, and operating cost. High duct velocities also impair proper air balance, as will be discussed below.

Even if the projection of total exhaust for the system is correct, there can still be problems if the actual distribution of exhaust requirements among the food court tenants does not match the assumptions that the designer used to lay out the base building ductwork. Tenant layouts tend to be fluid as leasing agents negotiate their best deals with numerous prospective tenants. A space initially designated for a heavy-duty cooking tenant can be leased to an ice cream vendor, for example, which can cause velocities in the branch ductwork to go seriously awry.

A second disadvantage to master exhaust systems is that codes do not permit installation of any sort of balancing device within kitchen exhaust ductwork. A damper in the greasy exhaust airstream is liable to become coated with grease, creating a fire hazard. With all tenants providing their own hoods and ductwork from the hood to the master duct, how can the designer of the master exhaust system ensure that the air balance will be acceptable, without the use of balancing dampers?

A third disadvantage is that the master exhaust duct connecting all of the tenant hoods together can spread fire from one tenant space to the next. Although it is permissible for individual UL-listed hoods to have built-in fire dampers, codes do not permit installation of fire dampers within the exhaust duct. Once a fire spreads from one tenant's hood to the master exhaust duct, the fire can then spread throughout the master exhaust duct to any other tenant space. Duct walls that have become coated with grease due to low air velocity or poor maintenance can only serve to speed the spread of fire within the duct.

A fourth disadvantage is that the exhaust requirements of a food court are not always stable over time. The food court may open while only partially leased and may take many months to become fully occupied. Tenants may change the nature of their operations or go out of business. It is impossible to design a master exhaust system that can automatically adjust to changes in exhaust requirements and remain within code operating parameters.

Finally, while all of the advantages are both obvious and very attractive to developers, most of the disadvantages are neither obvious nor simple to explain and can easily be ignored by developers. While low air velocity in the ductwork can be a real fire hazard, the hazard is concealed, and the system can appear to work perfectly. Even with the best of intentions, developers usually have enough problems getting and keeping shopping centers running and do not have time to worry about hidden conditions that may pose a hazard at some time in the indefinite future.

## SUCCESSFUL MASTER KITCHEN EXHAUST SYSTEMS

Since owners will continue to favor master kitchen exhaust systems, engineers will continue to design them despite the disadvantages. There are, however, several strategies that can minimize the disadvantages and lead to successful master kitchen exhaust systems. The engineer should

- make informed projections of exhaust requirements,
- maximize flexibility of design,
- match the design to the tenant layout,
- keep the owner informed,
- coordinate fire protection,
- be involved in the tenant fit-up stage, and
- make final adjustments once the system is operating.

### Making Informed Projections of Exhaust Requirements

Most utility services are made available to tenants on the basis of standard allowances. For example, each tenant is permitted up to X watts per square foot of electrical system capacity or Y tons of air-conditioning capacity per square foot. This is not a practical solution for master kitchen exhaust systems, for several reasons. First, tenant exhaust requirements can vary over a very wide range, from 0 to more than 9 cfm/ft$^2$, depending on the type and amount of cooking equipment in the space. It is not possible to force a tenant to reduce the exhaust requirement without eliminating cooking equipment, which could seriously affect the tenant's profitability. Second, oversizing an exhaust system is not desirable. Unlike electricity and air conditioning, where excess capacity can be provided to handle higher than anticipated loads, exhaust systems cannot be allowed to operate at partial capacity. Third, it is very difficult to shift exhaust system capacity from one location to another. For these reasons, exhaust systems should be sized to meet actual tenant exhaust requirements.

In virtually all cases, however, the overall base building design will need to be completed long before the tenant mix has been finalized. At that time, the number of tenants and the type of food service of each tenant space may be no more than a first pass estimate by the owner's leasing agent. In many cases, the base building construction will be well under way before the final tenant layout has been reached and actual tenants are assigned to each space. The designer of the master kitchen exhaust system will therefore need to project tenant kitchen exhaust requirements based on information provided by the owner in order to design the base building system.

While kitchen exhaust requirements for any individual tenant will depend on that tenant's actual complement of cooking equipment, which cannot be predicted in advance, there is a general correlation among exhaust requirements, tenant size, and type of food service. Figures 1 and 2 present actual tenant exhaust data for a representative sampling of food court tenants in several recently devel-

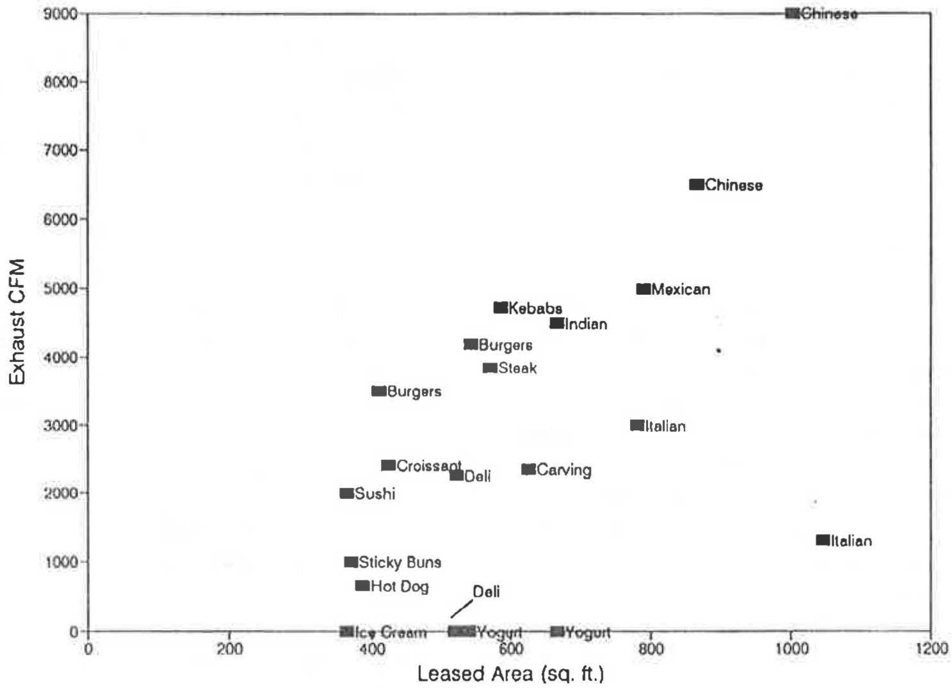


*Figure 1 Typical food court kitchen exhaust requirements—cfm vs. leased area.*

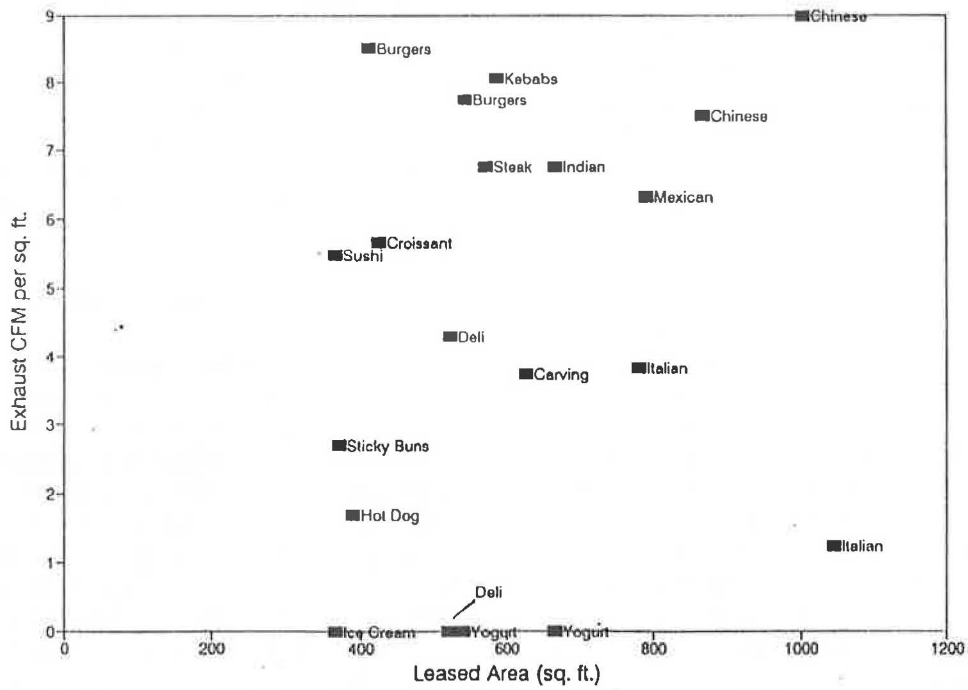


*Figure 2 Typical food court kitchen exhaust requirements—cfm per square foot leased area vs. leased area.*

oped shopping centers. Figure 1 shows total exhaust cfm, and Figure 2 shows exhaust cfm per square foot for each tenant. All data are based on design airflows for UL-listed hoods, which generally require less exhaust than nonlisted hoods. The overall average for all of these tenants is on the order of 4.5 cfm/ft$^2$. This factor is useful for sizing the main exhaust riser. As Figure 2 shows, exhaust requirements can range from 0 cfm/ft$^2$ for noncooking uses (ice cream, yogurt, some delis) to as high as 9 cfm/ft$^2$ for very heavy cooking uses (Chinese, burgers). These data can be used to estimate exhaust requirements on a space-by-space basis for the purposes of sizing the main ductwork. It must be stressed that these are general estimates only. By collecting and maintaining a data base of actual tenant exhaust requirements, designers will improve their ability to project exhaust requirements.

## Maximizing Flexibility of the Design

The master kitchen exhaust system must be flexible enough to accommodate differences between the designer's projected exhaust layout and actual tenant exhaust requirements. Making informed projections of exhaust requirements should minimize the differences between projections and reality, but some factors are unpredictable. In an extreme case, the owner may plan for a particular space to be a Chinese restaurant but may finally make a deal with an ice cream shop. Also, when one tenant goes out of business, the replacement tenant may have very different exhaust requirements.

Maximizing the flexibility of the design means maximizing the ability to accommodate differences between exhaust projections and actual requirements. The following "rules" maximize flexibility:

1. Don't use a master exhaust system to serve fewer than five tenant spaces. This will ensure that a large percentage difference between one tenant's projected and actual exhaust will be a small percentage difference in the overall exhaust for that branch. If there are only three tenants on a system, losing one tenant could mean losing 33% or more of the exhaust.
2. With five to eight tenants on a system, it is better to locate the exhaust riser at one end of the main duct run, rather than in the middle. With the riser in the middle, the system is effectively two subsystems, each serving fewer than five tenants, defeating the flexibility sought by rule 1. It is acceptable to have multiple branches, each serving one tenant space and all meeting at the main riser, however, since each branch would then be either "off" or "on," depending on whether that tenant had exhaust.
3. Size all of the ductwork at 1,800 fpm using the best available exhaust projections. Do not use common duct design practices such as equal friction or static regain. Most codes impose velocity limits on kitchen exhaust ductwork of 1,500 fpm minimum and 2,200 fpm maximum. Designing at 1,800 fpm leaves about 20% margin on either side. This rule implies that the main duct size should change at each tenant space where exhaust is expected and should not change where no exhaust connection is expected. If the duct needs to change shape to avoid obstructions, it should maintain the same cross-sectional area. Otherwise, the flexibility of the system is compromised.

   For example, a duct with a cross-sectional area of 3 ft$^2$ can accommodate from 4,500 cfm to 6,600 cfm, a range of 2,100 cfm. A duct with a cross-sectional area of 3.5 ft$^2$ can accommodate from 5,250 cfm to 7,700 cfm, a range of 2,450 cfm. A duct that changes from 3.5 ft$^2$ to 3 ft$^2$ with no new exhaust connection is limited to the higher of the two minimums, 5,250 cfm (3.5 ft$^2$ at 1,500 fpm), and to the lower of the two maximums, 6,600 cfm (3 ft$^2$ at 2,200 fpm), reducing the capacity range to 1,350 cfm.
4. Mark points of tenant connection on the base building ductwork and on lease outline drawing masters distributed to the tenants, but do not cut in stub taps for future tenant connection. If an existing stub tap is cut in, the tenant's contractor is likely to use it whether it is appropriately sized or not.
5. Avoid excessive pressure drop in the main ductwork and fittings by minimizing the number of elbows and transitions, using long radius elbows (centerline radius at least equal the duct width), and avoiding long duct runs between tenant spaces. As discussed in more detail in the appendix, this minimizes the variation in static pressure in the main duct from one point of tenant connection to another, making it possible to establish a narrow range of operating static pressure over all points of tenant connection. All tenants can thus be required to design their portions of the system based on the expectation of a specified static pressure at the point of connection to the landlord's duct. This only applies to the duct from the point of the first tenant connection to the end of the duct. The pressure drop between the first tenant connection and the fan is not critical.

## Matching the Design to the Tenant Layout

The master kitchen exhaust system cannot be successful unless the design process follows the tenant leasing process. Early in the base building design period, the food court may not have reached its final size and shape, yet the designer will be called upon to lay out the master system. Even when the design drawings are released for pricing, particular tenants may not have been assigned to each space. It is best to divide the exhaust system conceptually into two parts: (1) the main riser and fan and (2) the horizontal ductwork on the food court floor.

The location and general size of the main riser and fan will need to be set early so that the architect and engineering designers for the upper floors can proceed with the design of those floors. Usually by the time the drawings need to be released for pricing, the food court will have reached its final size, so the size of the riser can be fairly safely finalized on the basis of total food court square footage.

The final tenant mix is likely to be unsettled when the design is released for pricing, however, so laying out the main ductwork to each space at that time is likely to be less satisfactory. The following procedure, although somewhat unusual, can permit the final ductwork layout to be delayed until relatively late in the leasing process, when most actual tenants will have been identified.

Lay out the ductwork on the design drawings based on early tenant data for the purposes of coordination and pricing, but identify the ductwork in the food court area as "future." Instruct the mechanical contractor to price that duct layout, to provide a unit price per pound of sheet metal for changes to that duct layout, and to define how much time the contractor will need from the time the final layout is released to fabricate and install that ductwork. Follow the leasing process closely, asking the owner to provide updates of all leasing developments in the food court area. Finalize the ductwork layout based on the best available tenant data at the latest date that will leave the sheet metal contractor enough time to fabricate and install the ductwork.

### Keeping the Owner Informed

At a certain point, the final ductwork layout will have to be released for fabrication. Further changes to the ductwork layout are then impractical. This may not prevent the owner from making further changes to food court layout, however. At this point, the designer's responsibility is inform the owner of the consequences of any proposed changes to the food court layout. If the owner is considering shifting tenants around or moving the entire food court, the designer should inform the owner whether the proposed change is likely to impair system performance. It is helpful to develop a computerized model of the exhaust system, as described in the appendix, to evaluate these changes. Special attention should be paid to changes that are likely to push the velocity in the main duct beyond the acceptable velocity limits or that will increase the pressure drop in the main duct.

### Coordinating Fire Protection

Fire protection is always an important issue for kitchen exhaust systems. Fire protection is particularly important for master kitchen exhaust systems, because the main ductwork provides an open passageway from one tenant space to the next, with no fire dampers where the duct penetrates rated partitions. A fire that starts under one tenant's hood and spreads to the exhaust duct can thus spread to all of the other tenant spaces on that system.

Effective fire protection of the master exhaust system is enhanced by two elements: (1) fire dampers at every hood and (2) coordinating extinguishing systems.

1. Fire dampers in kitchen exhaust systems are only permitted where they provided by the hood manufacturer as part of a "listed hood and damper assembly." The fire damper is generally installed at the hood's duct collar. Exhaust hoods that have fire dampers fall into one of two categories: either "UL listed hood with damper" or "UL listed grease extractor" (generically called "water wash hood") (UL 1990). Hoods without dampers cannot be UL listed but can be "UL classified hood without damper."
2. Fire extinguishing systems are required by code in most kitchen exhaust applications. Until recently, however, very little information was available to guide designers of master kitchen exhaust systems. Recent editions of NFPA Standard 17, Dry Chemical Extinguishing Systems, and NFPA Standard 17A, Wet Chemical Extinguishing Systems, now provide detailed information on applying these types of extinguishing systems to multiple-hood systems to protect the common duct as well as the individual hoods (NFPA 1990a,b).

For master kitchen exhaust systems, it is most practical for each tenant to install and maintain his own fire extinguishing system to protect his cooking surface, hood, and ductwork connection to the main duct. The most practical solution for protecting the base building (common) duct is to provide a separate extinguishing system to protect that duct independent of the tenants' systems. The common duct can be protected by sprinklers or by a dry or wet chemical system ($CO_2$ systems are not normally used because of the expense).

If sprinklers are used, NFPA 13 (section 4-4.17) requires sprinkler heads for every 10 feet of horizontal duct and at the top of each vertical rise (NFPA 1989). NFPA 96 (section 8-2.1.2) requires sprinkler heads to be replaced at least annually (NFPA 1991). Provisions should be made for the ductwork to drain after a sprinkler discharge. All branch duct connections should be made on the top or sides of the main duct, leaving at least a one-inch lip at the bottom of the main duct so that any water from the sprinklers will not drain back to a tenant's space. The sprinklers in the base building duct should be on a separate zone, and the flow switch for that zone should be wired to shut off fuel to all cooking equipment in that zone.

If a separate dry or wet chemical system is used for the common duct, it is usually possible to install one or more chemical nozzles in one location at the end of the duct remote from the fan to protect the entire duct up to the fan. The exhaust airflow carries the chemical throughout the main duct. It may be necessary to install several heat sensors throughout the common duct. In the event of a fire in the common duct, the fuel to all cooking equipment served by that common duct should be shut off.

If individual tenant extinguishing systems are used in combination to protect the common duct, all of the chemical agents used by the tenants must be compatible.

## Involvement in the Tenant Fit-Up Stage

In order to ensure that a master kitchen exhaust system will be successful, the designer must remain involved in the project as tenants design and install their portions of the system. The designer should prepare a document outlining tenant design criteria to ensure proper operation of the system. If possible, this document should be made part of the standard tenant lease or tenant mechanical, electrical, and plumbing criteria document.

Perhaps the most important criterion is the static pressure that the tenants can expect at the point of connection to the landlord's exhaust duct. As long as the pressure drop within the main exhaust duct is kept low over the portion of the duct where tenants make their connections, it should be possible to establish a relatively constant negative pressure at all points in that duct. The design criteria should promise a narrow range of pressures rather than one exact pressure, since the pressure drop in the main duct will not be zero. About 0.3 in. wg pressure drop in the main exhaust duct is reasonable. For example, "Landlord will maintain pressure in the main kitchen exhaust duct at the point of Tenant connection at −1.55 in. wg, ±0.15 in. The pressure drop through Tenant's exhaust hood and ductwork must be within this range."

Requiring all hoods to use the same type of grease extraction will help steer tenants toward compliance with the design pressure criteria. Kitchen hood grease extraction generally falls into three categories: mesh filters (not permitted in most jurisdictions) have a pressure drop of about 0.4 in. wg; panel filters have a pressure drop of about 0.7 to 0.9 in. wg; and high-velocity baffles, such as water wash hoods and dry hoods with similar baffle design, generally have a pressure drop in the range of 1.2 to 1.4 in. wg.

High-velocity baffle-type hoods are preferable because of their higher pressure drop, which minimizes the impact of any small variation in pressure in the main duct. The appendix to this paper describes a computer model that predicts "actual" airflow from all hoods on a master kitchen exhaust system, based on the geometry of the common exhaust duct and the "design" airflows and static pressures of each individual hood. For the simple example system in the appendix, using all high-velocity baffle hoods results in actual airflows at each hood of ±11% of the design. Using all panel-filter-type hoods, the actual airflow at each hood is between +10% and −21% of the design. Mixing baffle- and panel-type hoods results in a range of actual airflows of +10% to −23% of the design.

A further benefit of high-velocity baffle-type grease extraction is that, according to manufacturers' literature, most high-velocity baffles (both water wash and dry hoods) have grease-extraction efficiencies of 90% and higher, which is much greater than that of panel filters (about 70% maximum). This reduces the inherent risk of fire due to grease in the exhaust duct.

The designer should also review individual tenant designs to ensure that the tenants comply with the design criteria. The designer should check hood manufacturers' shop drawings to ensure that the manufacturers have the proper design data. If system capacity is very tight, the designer may be able to convince a tenant's designer to reduce the design exhaust quantity by calculating the required exhaust for the particular complement of cooking equipment and by discouraging "short-cycle" make-up air when its use increases the exhaust requirements (Black 1989).

With a computer model of the exhaust system, the designer can project conditions for each individual tenant during the design review process and can suggest modifications to the tenant's design to improve performance. For example, if the computer model predicts that the available pressure at a particular tenant's connection will be lower than the design, the designer can recommend that the tenant compensate by modifying his portion of the system to decrease the pressure requirements, perhaps by increasing duct size, using smooth radius elbows, or decreasing the pressure requirement of the hood. Most hood manufacturers are able to modify hood designs to increase or decrease the pressure requirement at the hood collar.

## Final Adjustments

Even with the best of intentions and attention, it is possible that the final operating parameters of the exhaust system will fall outside the parameters of the design. Hopefully, there will not be any problems with excessively high velocities that prevent the system from operating properly, because the designer will have alerted the owner to these possibilities and developed solutions during the leasing stage. But there may be areas where the main duct velocity will be below the code minimum, especially if there are any unoccupied tenant spaces.

The best solution to these conditions is direct bypass of make-up air from the common unconditioned make-up air system to the exhaust duct, to inject enough unconditioned make-up air directly into the kitchen exhaust duct to maintain the minimum velocity at all points of the system. Figure 3 shows such a bypass connection.

The make-up air bypass crossover duct should be constructed of the same material as the exhaust duct. There should be a fire damper in the crossover duct at the

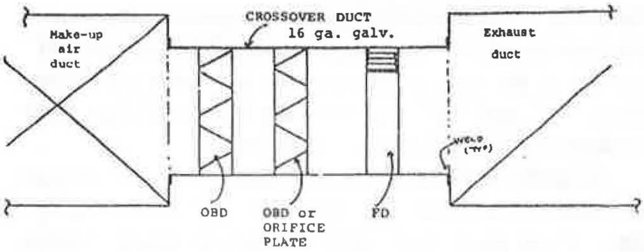


*Figure 3* *Make-up air to kitchen exhaust bypass connection detail.*

exhaust duct connection. Because of the large pressure difference between the exhaust and make-up air ducts, two volume dampers (or one damper and one orifice plate) should be installed in the crossover duct between the make-up air duct and the fire damper to adjust the amount of bypass. The first damper or the orifice plate absorbs most of the pressure difference, and the second damper is used for fine adjustment. The volume damper(s) should be secured in position to prevent accidental adjustment.

This bypass arrangement does not increase the fire hazard of the kitchen exhaust system. No greasy exhaust passes by either the volume damper or the fire damper; all air in the bypass is clean make-up air flowing from the make-up air duct to the exhaust duct, so there is very little risk of grease accumulating on the dampers.

## SUMMARY

The successful design and implementation of a master kitchen exhaust system is a challenge, but it can be achieved. Perhaps the most important challenge is to convince the owner and the owner's leasing staff of the need for special care and cooperation. The designer must design the base building ductwork using the best data available, delaying final duct layout until the tenant mix is finalized if possible, and making informed projections of exhaust requirements where necessary. The designer should remain involved through the tenant leasing process and startup, ensuring that the individual tenant designs are compatible with the overall system. A computer simulation model will help to predict the operating characteristics of the system and to identify problem areas.

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## APPENDIX

This appendix describes a computer spreadsheet-based simulation model for a master kitchen exhaust system and presents an example case where the computer simulation is used to evaluate different design alternatives.

### Simulation Model

A computer simulation model is particularly useful for master kitchen exhaust systems. Most types of supply and exhaust systems can be balanced to obtain the design airflows at all points by means of balancing dampers; as long as the fan develops sufficient pressure to handle the worst-case outlet (or inlet), the flow for all other outlets can be adjusted to the design by balancing dampers.

Since balancing dampers are not permitted in kitchen exhaust ductwork, the distribution of airflow in the system cannot be adjusted after construction. It is, therefore, helpful to be able to predict the airflow distribution for any proposed configuration of hoods and ductwork and to be able to test the effects of possible modifications to the system.

The spreadsheet model calculates the airflow for each hood using the three steps described below. All pressure drop calculations follow the methodology described in chapter 32 of *ASHRAE Fundamentals* (ASHRAE 1989).

1. For each individual hood and duct combination, calculate the pressure drop from the hood inlet to the point of connection to the main duct for the design airflow. This establishes a reference for each hood and branch duct combination. The airflow for any other pressure drop can then be calculated by applying the fan laws, as follows:

$$Q = Q_o (P/P_o)^{1/2} \qquad (1)$$

*where*

$Q$ = airflow for pressure drop of interest,
$P$ = pressure drop of interest,

$Q_o$ = reference airflow,
$P_o$ = reference pressure drop.

For example, the pressure drop for a hood and duct combination is calculated as 1.5 in. wg at a design airflow of 5,000 cfm. That is, if the pressure at the point of connection to the main duct is −1.5 in. wg, the airflow will be 5,000 cfm. If the pressure in the main duct is decreased to −1.2 in. wg, the airflow will be: 5000 $(1.2/1.5)^{1/2}$ = 4,472 cfm.

2. Calculate the available pressure at each branch duct connection using the design airflows. Select a common reference point in the main duct and a pressure to be maintained at this location (in practice, this pressure can be achieved by adjusting the fan speed). Calculate the pressure drop from the common reference point to each branch duct connection point. The available pressure at each branch duct connection point will be the pressure at the common reference point less the pressure drop from the common reference point to the branch duct connection point. This establishes a "design" available static pressure for each hood and branch duct combination.

3. The actual operating pressure drop from any common point in the main duct to the inlet of each tenant's hood must be the same because the air entering each tenant's hood is at the same (atmospheric) pressure. Therefore, if the available pressure at the connection to the main duct does not equal the design pressure drop for the hood and branch duct combination, the actual airflow will be different from the design flow.

An iterative calculation is then used to find the airflow for each hood that makes the pressure drop through the hood and duct combination equal to the available pressure at the branch duct connection. The process works as follows:

- Using Equation 1 and the design data for each hood and duct combination, calculate the airflow for each hood that corresponds to the available pressure calculated in step 2.
- Substitute the new airflows into the main duct pressure drop calculation (step 2) to find new available static pressure at each connection point.
- Repeat the process until the calculation converges to a unique solution. The calculation may require as many as 20 to 30 iterations to arrive at a unique solution, but fortunately modern PCs can accomplish this task in seconds.

Table 1 is the output of a typical calculation after the final iteration. Each line in the spreadsheet represents a feature of the system. Airflow is generally up the page.

Note the lowercase letters to the far left of several rows of Table 1 and on the schematic duct layout of Figure 4: Row "a" represents a length of straight duct downstream of the connection for Hood 1, and row "e" represents a length of straight duct between connections for Hood 1 and Hood 2. Row "b" represents the branch connections of the tee fittings for Hood 1, and row "d" represents the main connection for that same tee fitting. Rows "c" and "g" represent the hood and duct combination for Hoods 1 and 2, respectively.

Note the uppercase letters at the top of each column of Table 1; columns "A" through "F" are the design data for the main duct, including length, width, height, and equivalent (hydraulic) diameter for each section of straight duct, and descriptive information for each fitting, including the table number where pressure drop data are found (ASHRAE 1989, chapter 32), and the "C" factor for each fitting, as described in the ASHRAE Handbook. This part of the table is similar to ordinary pressure drop calculations in the ASHRAE Handbook.

Columns "G" and "H" are the design airflow and resulting pressure drop ("dP") for each hood. Columns "I" and "J" are the main duct airflow and velocity for the initial case, based on the design airflow to each hood.

Columns "K" through "S" calculate the available static pressure at each point of connection (in column "R" on the row with the hood data) and the airflow for each hood for that available static pressure (column "K"). Column "L" shows the calculated airflow as a percentage of the design airflow for each hood. Column "M" shows the calculated airflow in each section of the main duct. Columns "N" and "O" calculate average velocity (based on actual area) and equal friction velocity (based on hydraulic diameter) for each section in the main duct. Column "Q" estimates the friction loss in the main duct, using the following formula:*

$$dP/100' = 2.56(1/D_h)^{1.18}(V_{EF}/1000)^{1.8} \qquad (2)$$

*where*

$dP/100'$ = friction loss (in wg) per 100 ft of duct,
$D_h$ = hydraulic diameter,
$V_{EF}$ = equal friction velocity.

The results of Equation 2 are generally in good agreement with a "ductulator."

Column "R" is the calculated pressure loss for the element described on each row. Column "S" is the cumulative pressure loss to each point in the system relative to the common reference point. In Table 1, the pressure loss from the common reference point to the branch connection for Hood 3 is 0.07 in. wg (column "S"). Positing an available pressure at the common reference point of 1.65 in. wg (top of column "S"), the available pressure at the branch connection for Hood 3 is, therefore, 1.58 in. wg (column "R").

Example

The example calculates airflows to each hood of a sample system under six scenarios, representing different

*Pat Brooks, United McGill Corp., personal communication, 1986.

configurations of hood types and modifications to the main ductwork. This simple system consists of a single main exhaust duct with six hood connections. There is one 90° elbow in the main duct between Hoods 3 and 4. The layout of the system is shown schematically in Figure 4.

Two exhibits accompany each case: a complete printout table of the computer model output, including design data for the hoods and main duct system as well as the calculations described above, and a schematic diagram of the system, showing the design and the calculated (predicted) airflows for each hood and the calculated airflow as a percentage of the design.

The first four cases represent systems in which the designer is able to size the main duct for the actual design airflows for each hood. In other words, the main duct is optimally sized for all hoods. In the final two cases, the same main ductwork layout is used, but the design airflow for one hood is increased by about 60%. This represents a more realistic scenario in which the designer laid out the main ductwork on the basis of projected airflows for each tenant space, but these projected airflows are not entirely accurate.

Case I—All high-pressure-drop hoods, low-pressure-drop main (Table 1 and Figure 4): This is the recommended case. Hoods and branch ductwork are designed for a relatively high pressure drop of 1.4 in. wg, representing hoods with high-velocity baffle construction. The main ductwork is designed for nearly constant velocity of 1,800 fpm. Pressure drop in the main duct is kept as low as possible by limiting the distance between hood connections and by using a smooth inside radius elbow. The calculated airflows range from 89% to 111% of the design.

Case II—All high-pressure-drop hoods, high-pressure-drop main (Table 2 and Figure 5): This is the same as Case I, except that an additional pressure drop is imposed on the main duct between Hoods 3 and 4 by increasing the length of straight duct from 20 ft to 70 ft and by substituting a mitered elbow for the smooth radius elbow. The changes have very little effect on Hoods 1 through 4, but the additional pressure drop has decreased the airflow for the last hood on the line to 82% of the design. Although the design velocity in the main duct is always more than 1,700 fpm, the calculations in Table 2 show that the actual velocity in the last two sections of main ductwork will be below the code minimum of 1,500 fpm.

Case III—Mixed-pressure-drop hoods, low-pressure-drop main (Table 3 and Figure 6): In this case, half of the tenant hood and duct combinations are high pressure drop, as in the first cases, and half have a relatively low pressure drop of 0.85 in. wg, typical for panel-type filters. The pressure drop in the main is kept low, as in Case I. With the pressure in the main duct adjusted so that no hood is at more than 110% of design airflow, the airflow for the low-pressure-drop hoods is as low as 67% of the design.

Case IV—All low-pressure-drop hoods, low-pressure-drop main (Table 4 and Figure 7): In this case, all of the tenant hood and duct combinations are low pressure drop, designed for 0.85 in. wg at design airflow. The pressure drop in the main is kept low, as in Case I. With the pressure in the main duct adjusted to keep the upper limit of airflow for the hoods closest to the riser at 110% of the design airflow, the airflow for the hoods at the end of the run is as low as 79% of the design. Again, the last two sections of main duct are below the code minimum of 1,500 fpm.

Case V—All-high-pressure-drop hoods, deviation from the design (Table 5 and Figure 8): The main ductwork design for this case is identical to that of Case I. The only change is that the design airflow for Hood 4 is increased from 4,000 cfm to 6,500 cfm. All tenant hood and duct combinations are designed for high pressure drop. In this case, the calculated airflows for the individual hoods range from 86% to 111% of the design, and all calculated velocities for the main duct remain within code limits.

Case VI—All low-pressure-drop hoods, deviation from the design (Table 6 and Figure 9): This case is identical to Case V, except that the tenant hood and duct combinations are designed for low pressure drop. The calculated airflows for the individual hoods range from 75% to 110% of the design, a significantly higher range than for Case V. The calculated main duct velocities for the main duct beyond the connection for Hood 4 are below the code minimum velocity.

## TABLE 1
## Computer Simulation of Airflows in Master Kitchen Exhaust System, Case I

CASE I: All high pressure drop hoods, low pressure drop main

| | A | B | C | D | E | F | G | H | I | J | K | L | M | N | O | P | Q | R | S |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | DUCT | | | | | Design Data | | | | | | Calculations | | | | | | |
| | Item | Length or 'Fitting'* | W(in) | H(in) | Hydral. D(in) | 'C' | HOOD CFM | dP in wg | Main CFM | Main Vel ft/min | Hood CFM | % of design | Main CFM | Avg Vel ft/min | Eq. Fric Vel ft/min | VP in wg | P/100' in wg | dP in wg | Cum dP in wg |
| | Riser | | 54 | 32 | 45.1 | | | | | | | | 21927 | 1827 | | Avail. Pressure at Riser: | | | 1.65 |
| | Elbow | 3-5 | r/W=1 | | | 0.19 | | | | | | | | | | 0.244 | | 0.05 | 0.05 |
| a | Straight | 5 | 54 | 32 | 45.1 | | | | | | | | 21927 | 1827 | 1980 | 0.244 | 0.098 | 0.00 | 0.05 |
| b | T-B | 5-9 | Qb/Qc | 0.23 | | -0.5 | | | | | | | | | | 0.244 | | -0.12 | -0.07 |
| c | Hood 1 | | | | | | 4500 | 1.4 | 21700 | 1808 | 4989 | 111% | | | | | | 1.72 | |
| d | T-M | 5-3 | Qb/Qc | 0.23 | | 0.27 | | | | | | | | | | 0.244 | | 0.07 | 0.10 |
| e | Straight | 20 | 54 | 26 | 40.3 | | | | | | | | 16938 | 1737 | 1912 | 0.228 | 0.105 | 0.02 | 0.12 |
| f | T-B | 5-9 | Qb/Qc | 0.24 | | -0.48 | | | | | | | | | | 0.228 | | -0.11 | 0.01 |
| g | Hood 2 | | | | | | 3800 | 1.4 | 17200 | 1764 | 4110 | 108% | | | | | | 1.64 | |
| | T-M | 5-3 | Qb/Qc | 0.24 | | 0.28 | | | | | | | | | | 0.228 | | 0.06 | 0.17 |
| | Straight | 20 | 42 | 26 | 35.9 | | | | | | | | 12828 | 1692 | 1828 | 0.208 | 0.111 | 0.02 | 0.19 |
| | T-B | 5-9 | Qb/Qc | 0.21 | | -0.55 | | | | | | | | | | 0.208 | | -0.11 | 0.07 |
| | Hood 3 | | | | | | 2500 | 1.4 | 13400 | 1767 | 2653 | 106% | | | | | | 1.58 | |
| | T-M | 5-3 | Qb/Qc | 0.21 | | 0.26 | | | | | | | | | | 0.208 | | 0.05 | 0.24 |
| | Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 10175 | 1661 | 1823 | 0.207 | 0.126 | 0.01 | 0.25 |
| | Elbow | 3-5 | r/W=1.5, 90 deg | | | 0.15 | | | | | | | | | | 0.207 | | 0.03 | 0.28 |
| | Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 10175 | 1661 | 1823 | 0.207 | 0.126 | 0.01 | 0.30 |
| | T-B | 5-9 | Qb/Qc | 0.39 | | 0.17 | | | | | | | | | | 0.207 | | 0.04 | 0.29 |
| | Hood 4 | | | | | | 4000 | 1.4 | 10900 | 1780 | 3944 | 99% | | | | | | 1.36 | |
| | T-M | 5-3 | Qb/Qc | 0.39 | | 0.44 | | | | | | | | | | 0.207 | | 0.09 | 0.36 |
| | Straight | 20 | 32 | 18 | 26.0 | | | | | | | | 6232 | 1558 | 1694 | 0.179 | 0.142 | 0.03 | 0.39 |
| | T-B | 5-9 | Qb/Qc | 0.40 | | 0.34 | | | | | | | | | | 0.179 | | 0.06 | 0.45 |
| | Hood 5 | | | | | | 2700 | 1.4 | 6900 | 1127 | 2500 | 93% | | | | | | 1.20 | |
| | T-M | 5-3 | Qb/Qc | 0.40 | | 0.46 | | | | | | | | | | 0.179 | | 0.08 | 0.47 |
| | Straight | 20 | 20 | 17 | 20.1 | | | | | | | | 3732 | 1580 | 1687 | 0.177 | 0.190 | 0.04 | 0.51 |
| | Elbow | 3-5 | r/W=1, 90 deg | | | 0.21 | | | | | | | | | | 0.177 | | 0.04 | 0.54 |
| | Hood 6 | | | | | | 4200 | 1.4 | 4200 | 1779 | 3732 | 89% | | | | | | 1.11 | |

* - Duct length in feet, or ASHRAE table for fitting
T-B = Tee, branch
T-M = Tee, main

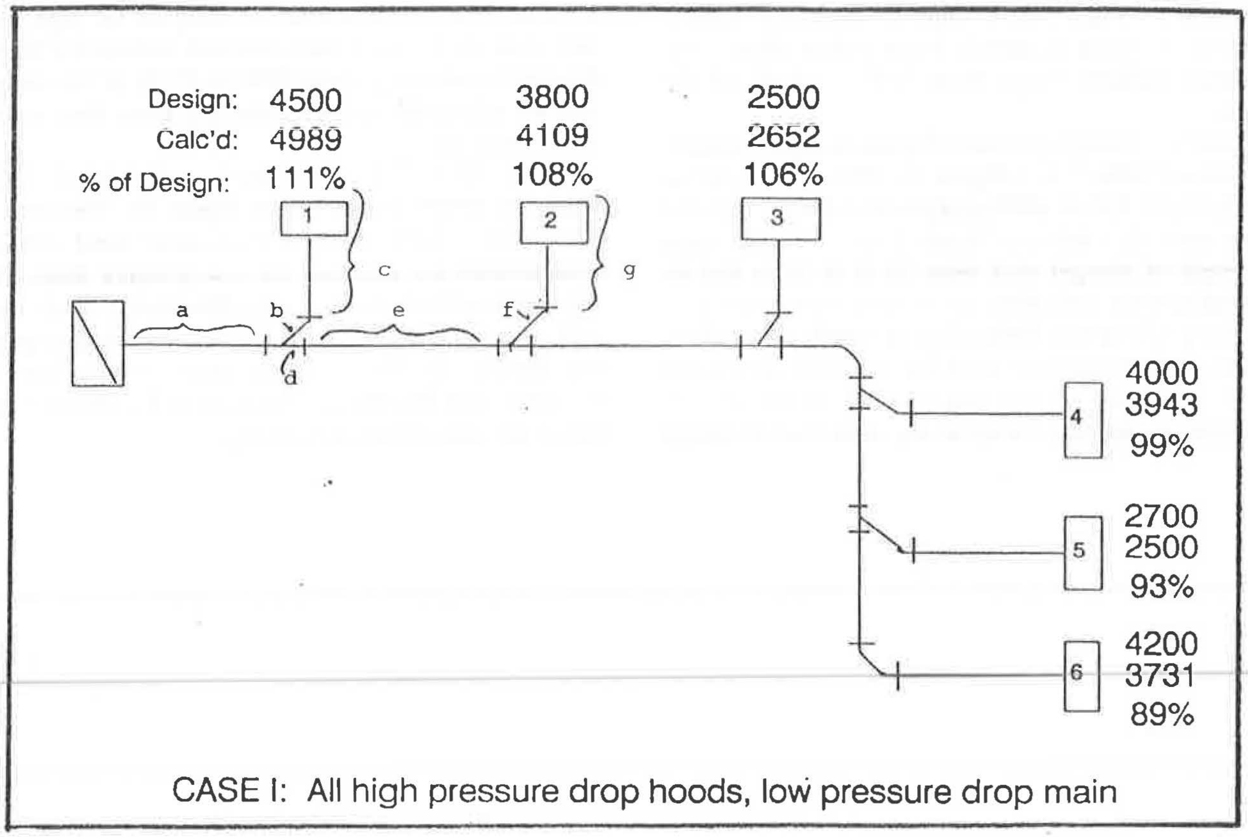


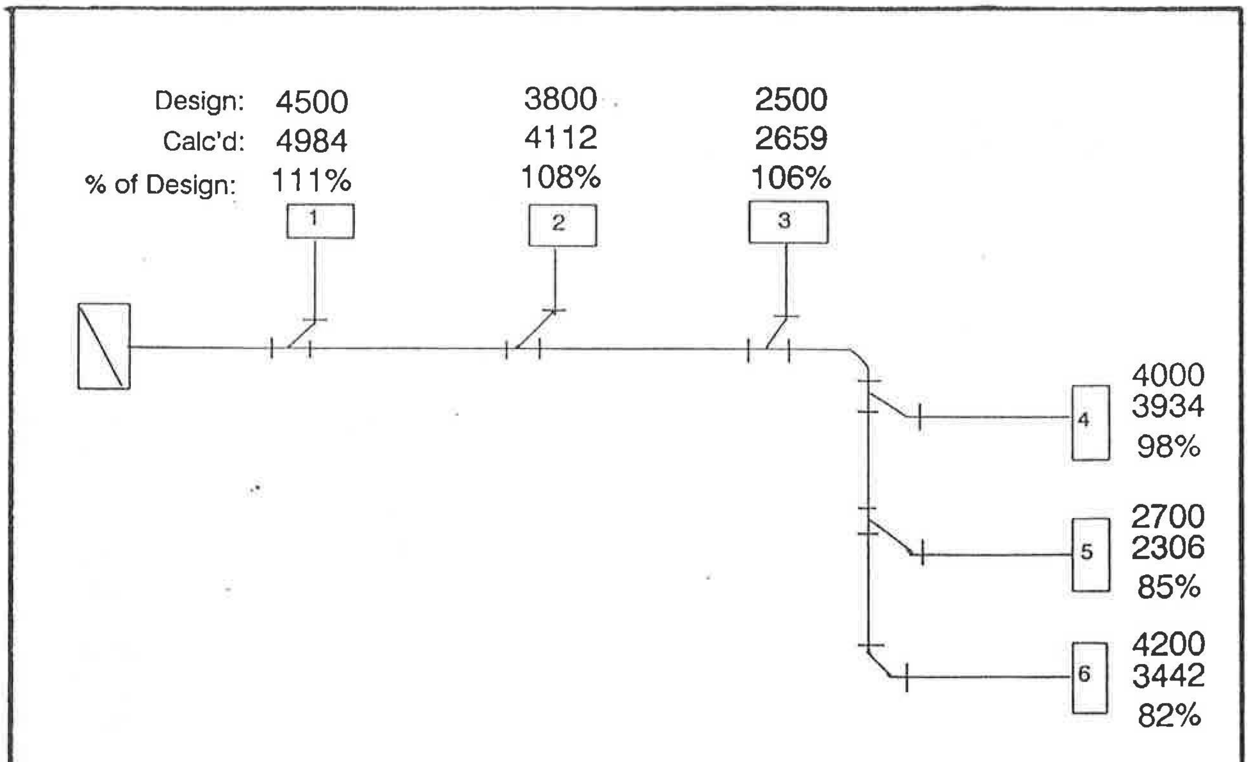

*Figure 4 Master kitchen exhaust system schematic duct layout with design and calculated airflows for each hood, case I.*

## TABLE 2
## Computer Simulation of Airflows in Master Kitchen Exhaust System, Case II

*CASE II: All high pressure drop hoods, high pressure drop main

| | DUCT | | | | | Design Data | | | | | | Calculations | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | Length or 'Fitting'* | W(in) | H(in) | Hydral. D(in) | 'C' | HOOD CFM | dP in wg | Main CFM | Main Vel ft/min | Hood CFM | % of design | Main CFM | Avg Vel ft/min | Eq. Fric Vel ft/min | VP in wg | P/100' in wg | dP in wg | Cum dP in wg |
| Riser | | 54 | 32 | 45.1 | | | | | | | | 21440 | 1787 | | Avail. Pressure at Riser: | | | 1.65 |
| Elbow | 3-5 | r/W=1 | | | 0.19 | | | | | | | | | | 0.234 | | 0.04 | 0.04 |
| Straight | 5 | 54 | 32 | 45.1 | | | | | | | | 21440 | 1787 | 1936 | 0.234 | 0.094 | 0.00 | 0.05 |
| T-B | 5-9 | Qb/Qc | 0.23 | | -0.5 | | | | | | | | | | 0.234 | | -0.12 | -0.07 |
| Hood 1 | | | | | | 4500 | 1.4 | 21700 | 1808 | 4985 | 111% | | | | | | 1.72 | |
| T-M | 5-3 | Qb/Qc | 0.23 | | 0.27 | | | | | | | | | | 0.234 | | 0.06 | 0.09 |
| Straight | 20 | 54 | 26 | 40.3 | | | | | | | | 16455 | 1688 | 1858 | 0.215 | 0.100 | 0.02 | 0.11 |
| T-B | 5-9 | Qb/Qc | 0.25 | | -0.48 | | | | | | | | | | 0.215 | | -0.10 | 0.01 |
| Hood 2 | | | | | | 3800 | 1.4 | 17200 | 1764 | 4112 | 108% | | | | | | 1.64 | |
| T-M | 5-3 | Qb/Qc | 0.25 | | 0.28 | | | | | | | | | | 0.215 | | 0.06 | 0.15 |
| Straight | 20 | 42 | 26 | 35.9 | | | | | | | | 12343 | 1628 | 1759 | 0.193 | 0.104 | 0.02 | 0.17 |
| T-B | 5-9 | Qb/Qc | 0.22 | | -0.55 | | | | | | | | | | 0.193 | | -0.11 | 0.07 |
| Hood 3 | | | | | | 2500 | 1.4 | 13400 | 1767 | 2659 | 106% | | | | | | 1.58 | |
| T-M | 5-3 | Qb/Qc | 0.22 | | 0.26 | | | | | | | | | | 0.193 | | 0.05 | 0.22 |
| Straight | 40 | 42 | 21 | 32.0 | | | | | | | | 9684 | 1581 | 1735 | 0.188 | 0.116 | 0.05 | 0.26 |
| Elbow | 3-5 | r/W=0.5, 90 deg | | | 1.1 | | | | | | | | | | 0.188 | | 0.21 | 0.47 |
| Straight | 30 | 42 | 21 | 32.0 | | | | | | | | 9684 | 1581 | 1735 | 0.188 | 0.116 | 0.03 | 0.50 |
| T-B | 5-9 | Qb/Qc | 0.41 | | 0.17 | | | | | | | | | | 0.188 | | 0.03 | 0.30 |
| Hood 4 | | | | | | 4000 | 1.4 | 10900 | 1780 | 3935 | 98% | | | | | | 1.35 | |
| T-M | 5-3 | Qb/Qc | 0.41 | | 0.44 | | | | | | | | | | 0.188 | | 0.08 | 0.55 |
| Straight | 20 | 32 | 18 | 26.0 | | | | | | | | 5749 | 1437 | 1563 | 0.152 | 0.123 | 0.02 | 0.58 |
| T-B | 5-9 | Qb/Qc | 0.40 | | 0.34 | | | | | | | | | | 0.152 | | 0.05 | 0.63 |
| Hood 5 | | | | | | 2700 | 1.4 | 6900 | 1127 | 2307 | 85% | | | | | | 1.02 | |
| T-M | 5-3 | Qb/Qc | 0.40 | | 0.46 | | | | | | | | | | 0.152 | | 0.07 | 0.64 |
| Straight | 20 | 20 | 17 | 20.1 | | | | | | | | 3442 | 1458 | 1556 | 0.151 | 0.164 | 0.03 | 0.68 |
| Elbow | 3-5 | r/W=1, 90 deg | | | 0.21 | | | | | | | | | | 0.151 | | 0.03 | 0.71 |
| Hood 6 | | | | | | 4200 | 1.4 | 4200 | 1779 | 3442 | 82% | | | | | | 0.94 | |

* - Duct length in feet, or ASHRAE table for fitting
T-B = Tee, branch
T-M = Tee, main




CASE II: All high pressure drop hoods, high pressure drop main

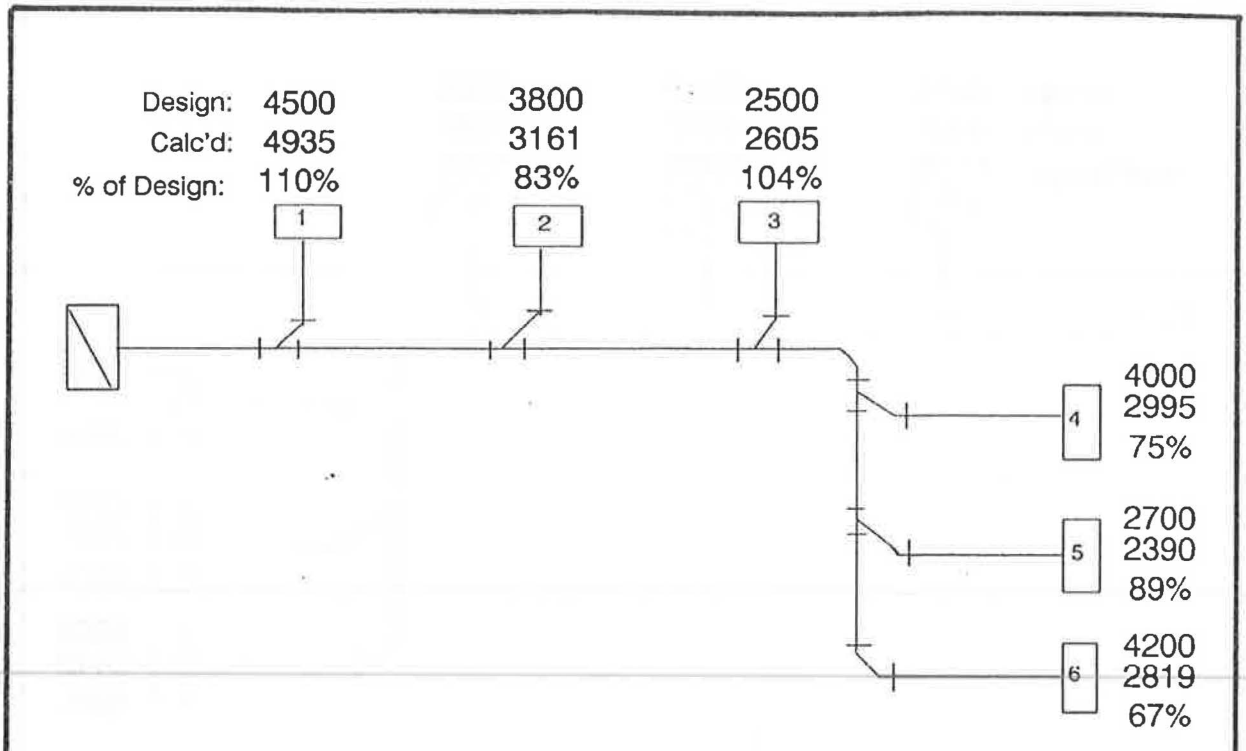
*Figure 5 Master kitchen exhaust system schematic duct layout with design and calculated airflows for each hood, case II.*

## TABLE 3
## Computer Simulation of Airflows in Master Kitchen Exhaust System, Case III

CASE III: Mixed pressure drop hoods, low pressure drop main

| | DUCT | | | | | Design Data | | | | | | Calculations | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | Length or 'Fitting'* | W(in) | H(in) | Hydral. D(in) | "C" | HOOD CFM | dP in wg | Main CFM | Main Vel ft/min | Hood CFM | % of design | Main CFM | Avg Vel ft/min | Eq. Fric Vel ft/min | VP in wg | P/100' In wg | dP in wg | Cum dP in wg |
| Riser | | 54 | 32 | 45.1 | | | | | | | | 18909 | 1576 | | Avail. Pressure at Riser: | | | 0.97 |
| Elbow | 3-5 | r/W=1 | | | 0.19 | | | | | | | | | | 0.182 | | 0.03 | 0.03 |
| Straight | 5 | 54 | 32 | 45.1 | | | | | | | | 18909 | 1576 | 1708 | 0.182 | 0.075 | 0.00 | 0.04 |
| T-B | 5-9 | Qb/Qc | 0.26 | | -0.5 | | | | | | | | | | 0.182 | | -0.09 | -0.05 |
| Hood 1 | | | | | | 4500 | 0.85 | 21700 | 1808 | 4936 | 110% | | | | | | 1.02 | |
| T-M | 5-3 | Qb/Qc | 0.26 | | 0.27 | | | | | | | | | | 0.182 | | 0.05 | 0.06 |
| Straight | 20 | 54 | 26 | 40.3 | | | | | | | | 13973 | 1433 | 1578 | 0.155 | 0.074 | 0.01 | 0.08 |
| T-B | 5-9 | Qb/Qc | 0.23 | | -0.48 | | | | | | | | | | 0.155 | | -0.07 | 0.00 |
| Hood 2 | | | | | | 3800 | 1.4 | 17200 | 1764 | 3161 | 83% | | | | | | 0.97 | |
| T-M | 5-3 | Qb/Qc | 0.23 | | 0.28 | | | | | | | | | | 0.155 | | 0.04 | 0.11 |
| Straight | 20 | 42 | 26 | 35.9 | | | | | | | | 10812 | 1426 | 1541 | 0.148 | 0.082 | 0.02 | 0.13 |
| T-B | 5-9 | Qb/Qc | 0.24 | | -0.55 | | | | | | | | | | 0.148 | | -0.08 | 0.05 |
| Hood 3 | | | | | | 2500 | 0.85 | 13400 | 1767 | 2605 | 104% | | | | | | 0.92 | |
| T-M | 5-3 | Qb/Qc | 0.24 | | 0.26 | | | | | | | | | | 0.148 | | 0.04 | 0.15 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 8206 | 1340 | 1470 | 0.135 | 0.086 | 0.01 | 0.16 |
| Elbow | 3-5 | r/W=1.5, 90 deg | | | 0.15 | | | | | | | | | | 0.135 | | 0.02 | 0.18 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 8206 | 1340 | 1470 | 0.135 | 0.086 | 0.01 | 0.19 |
| T-B | 5-9 | Qb/Qc | 0.37 | | 0.17 | | | | | | | | | | 0.135 | | 0.02 | 0.18 |
| Hood 4 | | | | | | 4000 | 1.4 | 10900 | 1780 | 2995 | 75% | | | | | | 0.79 | |
| T-M | 5-3 | Qb/Qc | 0.37 | | 0.44 | | | | | | | | | | 0.135 | | 0.06 | 0.24 |
| Straight | 20 | 32 | 18 | 26.0 | | | | | | | | 5211 | 1303 * | 1417 | 0.125 | 0.103 | 0.02 | 0.26 |
| T-B | 5-9 | Qb/Qc | 0.46 | | 0.34 | | | | | | | | | | 0.125 | | 0.04 | 0.30 |
| Hood 5 | | | | | | 2700 | 0.85 | 6900 | 1127 | 2391 | 89% | | | | | | 0.67 | |
| T-M | 5-3 | Qb/Qc | 0.46 | | 0.46 | | | | | | | | | | 0.125 | | 0.06 | 0.29 |
| Straight | 20 | 20 | 17 | 20.1 | | | | | | | | 2820 | 1194 | 1275 | 0.101 | 0.115 | 0.02 | 0.32 |
| Elbow | 3-5 | r/W=1, 90 deg | | | 0.21 | | | | | | | | | | 0.101 | | 0.02 | 0.34 |
| Hood 6 | | | | | | 4200 | 1.4 | 4200 | 1779 | 2820 | 67% | | | | | | 0.63 | |

* - Duct length in feet, or ASHRAE table for fitting
T-B = Tee, branch
T-M = Tee, main




CASE III: Mixed pressure drop hoods, low pressure drop main

*Figure 6 Master kitchen exhaust system schematic duct layout with design and calculated airflows for each hood, case III.*

## TABLE 4
## Computer Simulation of Airflows in Master Kitchen Exhaust System, Case IV

CASE IV: All low pressure drop hoods, low pressure drop main

| | DUCT | | | | | Design Data | | | | | | Calculations | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | Length or 'Fitting'* | W(in) | H(in) | Hydral. D(in) | "C" | HOOD CFM | dP in wg | Main CFM | Main Vel ft/min | Hood CFM | % of design | Main CFM | Avg Vel ft/min | Eq. Fric Vel ft/min | VP in wg | P/100' in wg | dP in wg | Cum dP in wg |
| Riser | | 54 | 32 | 45.1 | | | | | | | | 20896 | 1741 | | Avail. Pressure at Riser: | | | 0.97 |
| Elbow | 3-5 | r/W=1 | | | 0.19 | | | | | | | | | | 0.222 | | 0.04 | 0.04 |
| Straight | 5 | 54 | 32 | 45.1 | | | | | | | | 20896 | 1741 | 1887 | 0.222 | 0.090 | 0.00 | 0.05 |
| T-B | 5-9 | Qb/Qc | 0.24 | | -0.5 | | | | | | | | | | 0.222 | | -0.11 | -0.06 |
| Hood 1 | | | | | | 4500 | 0.85 | 21700 | 1808 | 4964 | 110% | | | | | | 1.03 | |
| T-M | 5-3 | Qb/Qc | 0.24 | | 0.27 | | | | | | | | | | 0.222 | | 0.06 | 0.09 |
| Straight | 20 | 54 | 26 | 40.3 | | | | | | | | 15932 | 1634 | 1799 | 0.202 | 0.094 | 0.02 | 0.11 |
| T-B | 5-9 | Qb/Qc | 0.25 | | -0.48 | | | | | | | | | | 0.202 | | -0.10 | 0.01 |
| Hood 2 | | | | | | 3800 | 0.85 | 17200 | 1764 | 4042 | 106% | | | | | | 0.96 | |
| T-M | 5-3 | Qb/Qc | 0.25 | | 0.28 | | | | | | | | | | 0.202 | | 0.06 | 0.14 |
| Straight | 20 | 42 | 26 | 35.9 | | | | | | | | 11890 | 1568 | 1694 | 0.179 | 0.097 | 0.02 | 0.16 |
| T-B | 5-9 | Qb/Qc | 0.22 | | -0.55 | | | | | | | | | | 0.179 | | -0.10 | 0.06 |
| Hood 3 | | | | | | 2500 | 0.85 | 13400 | 1767 | 2587 | 103% | | | | | | 0.91 | |
| T-M | 5-3 | Qb/Qc | 0.22 | | 0.26 | | | | | | | | | | 0.179 | | 0.05 | 0.20 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 9303 | 1519 | 1667 | 0.173 | 0.108 | 0.01 | 0.21 |
| Elbow | 3-5 | r/W=1.5, 90 deg | | | 0.15 | | | | | | | | | | 0.173 | | 0.03 | 0.24 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 9303 | 1519 | 1667 | 0.173 | 0.108 | 0.01 | 0.25 |
| T-B | 5-9 | Qb/Qc | 0.40 | | 0.17 | | | | | | | | | | 0.173 | | 0.03 | 0.24 |
| Hood 4 | | | | | | 4000 | 0.85 | 10900 | 1780 | 3709 | 93% | | | | | | 0.73 | |
| T-M | 5-3 | Qb/Qc | 0.40 | | 0.44 | | | | | | | | | | 0.173 | | 0.08 | 0.29 |
| Straight | 20 | 32 | 18 | 26.0 | | | | | | | | 5594 | 1398 | 1521 | 0.144 | 0.117 | 0.02 | 0.32 |
| T-B | 5-9 | Qb/Qc | 0.41 | | 0.34 | | | | | | | | | | 0.144 | | 0.05 | 0.37 |
| Hood 5 | | | | | | 2700 | 0.85 | 6900 | 1127 | 2276 | 84% | | | | | | 0.60 | |
| T-M | 5-3 | Qb/Qc | 0.41 | | 0.46 | | | | | | | | | | 0.144 | | 0.07 | 0.38 |
| Straight | 20 | 20 | 17 | 20.1 | | | | | | | | 3318 | 1405 | 1500 | 0.140 | 0.154 | 0.03 | 0.41 |
| Elbow | 3-5 | r/W=1, 90 deg | | | 0.21 | | | | | | | | | | 0.140 | | 0.03 | 0.44 |
| Hood 6 | | | | | | 4200 | 0.85 | 4200 | 1779 | 3318 | 79% | | | | | | 0.53 | |

* - Duct length in feet, or ASHRAE table for fitting
T-B = Tee, branch
T-M = Tee, main

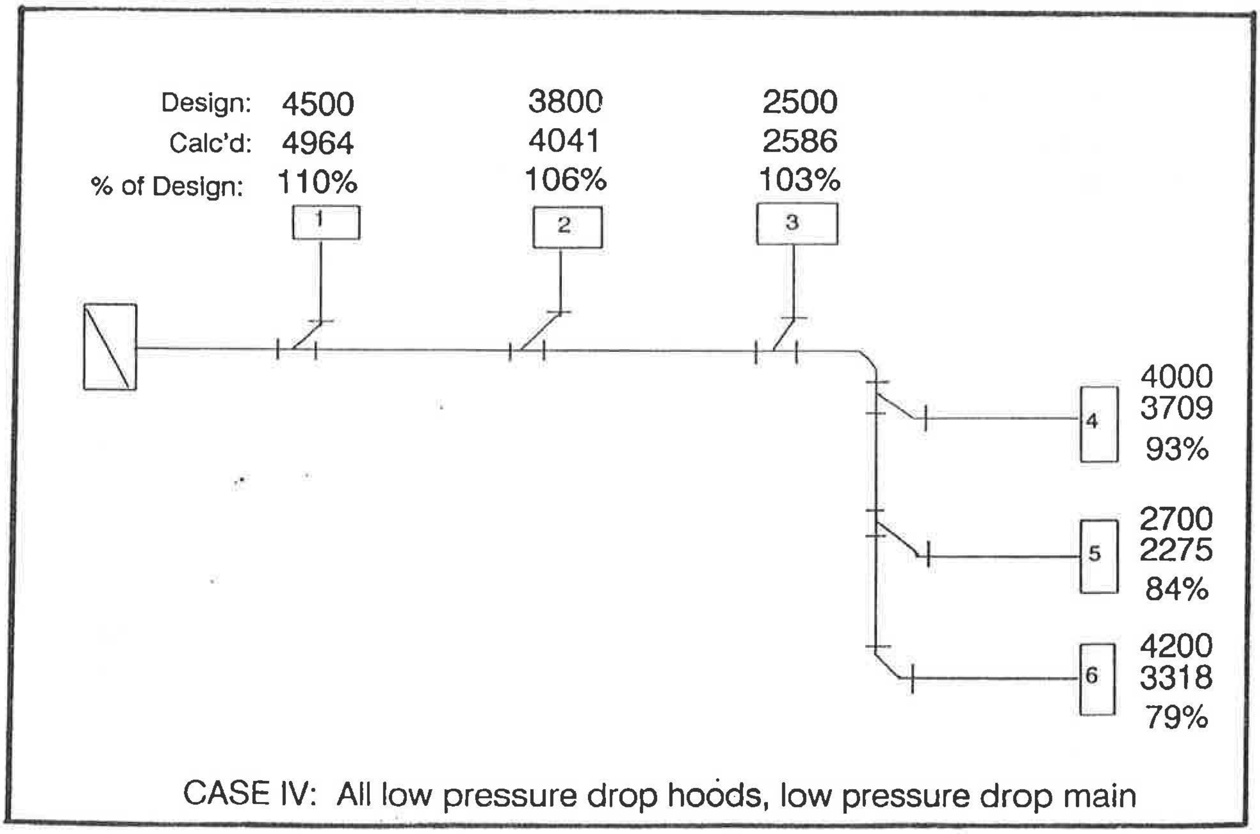



*Figure 7 Master kitchen exhaust system schematic duct layout with design and calculated airflows for each hood, case IV.*

## TABLE 5
## Computer Simulation of Airflows in Master Kitchen Exhaust System, Case V

CASE V: All high pressure drop hoods, deviation from projected design
Hood 4 projected 4000 CFM, actual 6500 CFM

| | DUCT | | | | | Design Data | | | | | | Calculations | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | Length or 'Fitting'* | W(in) | H(in) | Hydral. D(in) | 'C' | HOOD CFM | dP in wg | Main CFM | Main Vel ft/min | Hood CFM | % of design | Main CFM | Avg Vel ft/min | Eq. Fric Vel ft/min | VP in wg | P/100' in wg | dP in wg | Cum dP in wg |
| Riser | | 54 | 32 | 45.1 | | | | | | | | 23884 | 1990 | | Avail. Pressure at Riser: | | | 1.65 |
| Elbow | 3-5 | r/W=1 | | | 0.19 | | | | | | | | | | 0.290 | | 0.06 | 0.06 |
| Straight | 5 | 54 | 32 | 45.1 | | | | | | | | 23884 | 1990 | 2157 | 0.290 | 0.114 | 0.01 | 0.06 |
| T-B | 5-9 | Qb/Qc | 0.21 | | -0.5 | | | | | | | | | | 0.290 | | -0.15 | -0.08 |
| Hood 1 | | | | | | 4500 | 1.4 | 24200 | 2017 | 5000 | 111% | | | | | | 1.73 | |
| T-M | 5-3 | Qb/Qc | 0.21 | | 0.27 | | | | | | | | | | 0.290 | | 0.08 | 0.13 |
| Straight | 20 | 54 | 26 | 40.3 | | | | | | | | 18876 | 1936 | 2131 | 0.283 | 0.128 | 0.03 | 0.16 |
| T-B | 5-9 | Qb/Qc | 0.22 | | -0.48 | | | | | | | | | | 0.283 | | -0.14 | 0.02 |
| Hood 2 | | | | | | 3800 | 1.4 | 19700 | 2021 | 4098 | 108% | | | | | | 1.63 | |
| T-M | 5-3 | Qb/Qc | 0.22 | | 0.28 | | | | | | | | | | 0.283 | | 0.08 | 0.23 |
| Straight | 20 | 42 | 26 | 35.9 | | | | | | | | 14778 | 1949 | 2106 | 0.277 | 0.143 | 0.03 | 0.26 |
| T-B | 5-9 | Qb/Qc | 0.18 | | -0.55 | | | | | | | | | | 0.277 | | -0.15 | 0.11 |
| Hood 3 | | | | | | 2500 | 1.4 | 15900 | 2097 | 2625 | 105% | | | | | | 1.54 | |
| T-M | 5-3 | Qb/Qc | 0.18 | | 0.26 | | | | | | | | | | 0.277 | | 0.07 | 0.35 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 12153 | 1984 | 2177 | 0.296 | 0.174 | 0.02 | 0.37 |
| Elbow | 3-5 | r/W=1.5, 90 deg | | | 0.15 | | | | | | | | | | 0.296 | | 0.04 | 0.41 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 12153 | 1984 | 2177 | 0.296 | 0.174 | 0.02 | 0.43 |
| T-B | 5-9 | Qb/Qc | 0.50 | | 0.17 | | | | | | | | | | 0.296 | | 0.05 | 0.42 |
| Hood 4 | | | | | | 6500 | 1.4 | 13400 | 2188 | 6099 | 94% | | | | | | 1.23 | |
| T-M | 5-3 | Qb/Qc | 0.50 | | 0.44 | | | | | | | | | | 0.296 | | 0.13 | 0.43 |
| Straight | 20 | 32 | 18 | 26.0 | | | | | | | | 6055 | 1514 | *1646 | 0.169 | 0.135 | 0.03 | 0.46 |
| T-B | 5-9 | Qb/Qc | 0.40 | | 0.34 | | | | | | | | | | 0.169 | | 0.06 | 0.52 |
| Hood 5 | | | | | | 2700 | 1.4 | 6900 | 1127 | 2429 | 90% | | | | | | 1.13 | |
| T-M | 5-3 | Qb/Qc | 0.40 | | 0.46 | | | | | | | | | | 0.169 | | 0.08 | 0.54 |
| Straight | 20 | 20 | 17 | 20.1 | | | | | | | | 3626 | 1536 | 1639 | 0.167 | 0.180 | 0.04 | 0.57 |
| Elbow | 3-5 | r/W=1, 90 deg | | | 0.21 | | | | | | | | | | 0.167 | | 0.04 | 0.61 |
| Hood 6 | | | | | | 4200 | 1.4 | 4200 | 1779 | 3626 | 86% | | | | | | 1.04 | |

* - Duct length in feet, or ASHRAE table for fitting
T-B = Tee, branch
T-M = Tee, main

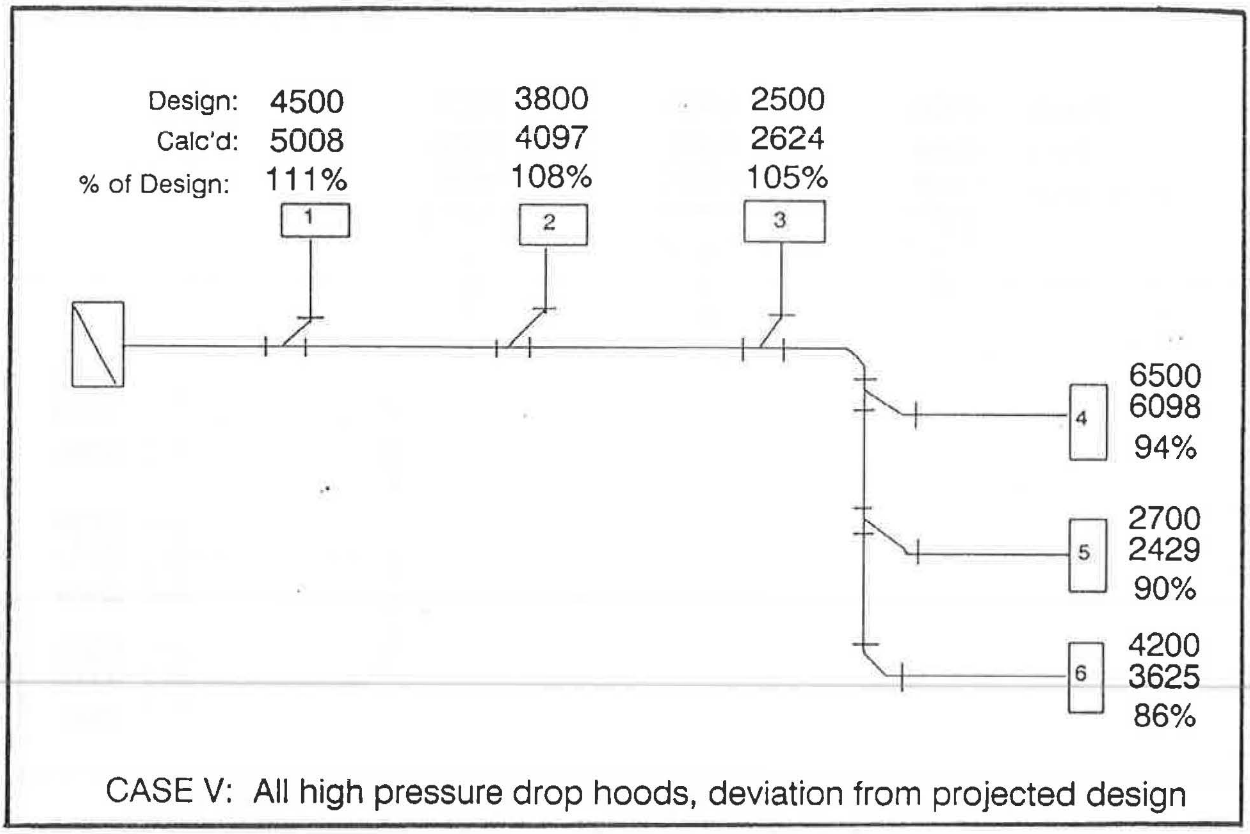


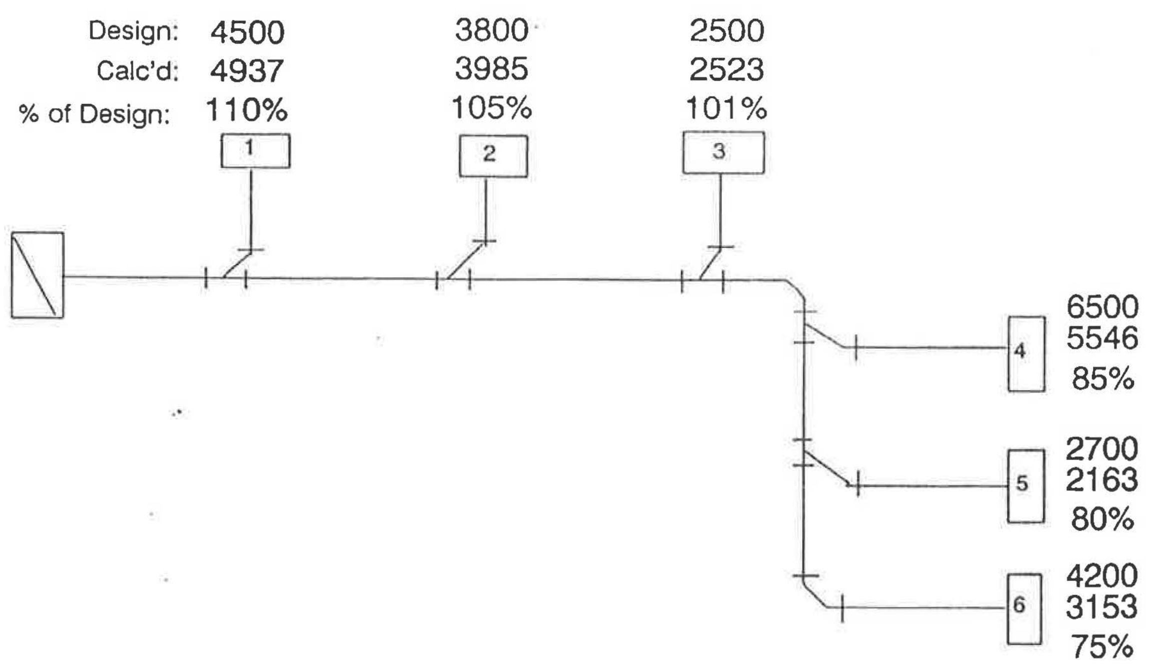

*Figure 8 Master kitchen exhaust system schematic duct layout with design and calculated airflows for each hood, case V.*

## TABLE 6
## Computer Simulation of Airflows in Master Kitchen Exhaust System, Case VI

CASE VI: All low pressure drop hoods, deviation from projected design
Hood 4 projected 4000 CFM, actual 6500 CFM

| Item | DUCT: Length or 'Fitting'* | DUCT: W(in) | DUCT: H(in) | DUCT: Hydral. D(in) | 'C' | Design Data: HOOD CFM | Design Data: dP in wg | Design Data: Main CFM | Design Data: Main Vel ft/min | Calculations: Hood CFM | Calculations: % of design | Calculations: Main CFM | Calculations: Avg Vel ft/min | Calculations: Eq. Fric Vel ft/min | Calculations: VP in wg | Calculations: P/100' in wg | Calculations: dP in wg | Calculations: Cum dP in wg |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Riser | | 54 | 32 | 45.1 | | | | | | | | 22311 | 1859 | | Avail. Pressure at Riser: | | | 0.95 |
| Elbow | 3-5 | r/W=1 | | | 0.19 | | | | | | | | | | 0.253 | | 0.05 | 0.05 |
| Straight | 5 | 54 | 32 | 45.1 | | | | | | | | 22311 | 1859 | 2015 | 0.253 | 0.101 | 0.01 | 0.05 |
| T-B | 5-9 | Qb/Qc | 0.22 | | -0.5 | | | | | | | | | | 0.253 | | -0.13 | -0.07 |
| Hood 1 | | | | | | 4500 | 0.85 | 24200 | 2017 | 4938 | 110% | | | | | | 1.02 | |
| T-M | 5-3 | Qb/Qc | 0.22 | | 0.27 | | | | | | | | | | 0.253 | | 0.07 | 0.11 |
| Straight | 20 | 54 | 26 | 40.3 | | | | | | | | 17373 | 1782 | 1962 | 0.240 | 0.110 | 0.02 | 0.13 |
| T-B | 5-9 | Qb/Qc | 0.23 | | -0.48 | | | | | | | | | | 0.240 | | -0.12 | 0.02 |
| Hood 2 | | | | | | 3800 | 0.85 | 19700 | 2021 | 3985 | 105% | | | | | | 0.93 | |
| T-M | 5-3 | Qb/Qc | 0.23 | | 0.28 | | | | | | | | | | 0.240 | | 0.07 | 0.18 |
| Straight | 20 | 42 | 26 | 35.9 | | | | | | | | 13388 | 1765 | 1908 | 0.227 | 0.120 | 0.02 | 0.21 |
| T-B | 5-9 | Qb/Qc | 0.19 | | -0.55 | | | | | | | | | | 0.227 | | -0.12 | 0.08 |
| Hood 3 | | | | | | 2500 | 0.85 | 15900 | 2097 | 2524 | 101% | | | | | | 0.87 | |
| T-M | 5-3 | Qb/Qc | 0.19 | | 0.26 | | | | | | | | | | 0.227 | | 0.06 | 0.28 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 10864 | 1774 | 1946 | 0.236 | 0.142 | 0.01 | 0.29 |
| Elbow | 3-5 | r/W=1.5, 90 deg | | | 0.15 | | | | | | | | | | 0.236 | | 0.04 | 0.33 |
| Straight | 10 | 42 | 21 | 32.0 | | | | | | | | 10864 | 1774 | 1946 | 0.236 | 0.142 | 0.01 | 0.34 |
| T-B | 5-9 | Qb/Qc | 0.51 | | 0.17 | | | | | | | | | | 0.236 | | 0.04 | 0.33 |
| Hood 4 | | | | | | 6500 | 0.85 | 13400 | 2188 | 5547 | 85% | | | | | | 0.62 | |
| T-M | 5-3 | Qb/Qc | 0.51 | | 0.44 | | | | | | | | | | 0.236 | | 0.10 | 0.34 |
| Straight | 20 | 32 | 18 | 26.0 | | | | | | | | 5317 | 1329 | 1446 | 0.130 | 0.106 | 0.02 | 0.36 |
| T-B | 5-9 | Qb/Qc | 0.41 | | 0.34 | | | | | | | | | | 0.130 | | 0.04 | 0.40 |
| Hood 5 | | | | | | 2700 | 0.85 | 6900 | 1127 | 2164 | 80% | | | | | | 0.55 | |
| T-M | 5-3 | Qb/Qc | 0.41 | | 0.46 | | | | | | | | | | 0.130 | | 0.06 | 0.42 |
| Straight | 20 | 20 | 17 | 20.1 | | | | | | | | 3154 | 1336 | 1425 | 0.127 | 0.140 | 0.03 | 0.44 |
| Elbow | 3-5 | r/W=1, 90 deg | | | 0.21 | | | | | | | | | | 0.127 | | 0.03 | 0.47 |
| Hood 6 | | | | | | 4200 | 0.85 | 4200 | 1779 | 3154 | 75% | | | | | | 0.48 | |

* - Duct length in feet, or ASHRAE table for fitting
T-B = Tee, branch
T-M = Tee, main




CASE VI: All low pressure drop hoods, deviation from projected design

*Figure 9 Master kitchen exhaust system schematic duct layout with design and calculated airflows for each hood, case VI.*