

High Vacuum, High Airflow Blower Testing and Design for Soil Vapor Intrusion Mitigation in Commercial Buildings

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ABSTRACT

This paper describes the method used to design and optimize a soil vapor intrusion mitigation system for a medical center that was a former metal products manufacturing facility. The site is heavily impacted by benzene and chlorinated solvents. The New Jersey Non Residential Indoor Screening Level is 3 ug/m³. The pre-mitigation indoor air samples averaged 198 ug/m³ for Trichloroethene (TCE). The building has undergone significant renovation and requires a unique solution. In addition to construction features that inherently increase indoor to sub-slab pressure differentials, soil depressurization is complicated by a highly compacted fill layer and a second concrete slab that was installed eighteen inches above the original floor slab. The low permeability, compacted fill between the two concrete floors required high vacuum and high airflow. Multiple, more common, low airflow-high vacuum blowers would have typically been required. The use of high vacuum, high airflow radial blowers in this situation was an improvement over conventional engineered controls because it enabled two blowers to deliver high static vacuum to sixteen suction point locations. The blower bench testing method and the vacuum field extension method and analysis to determine the blower specifications is described. The authors demonstrate how the application of fewer high output radial blowers is an improvement over multiple conventional inline, centrifugal and regenerative blowers in providing vapor intrusion mitigation solutions for buildings constructed over low permeability soils.

1.0 INTRODUCTION

The authors used extensive sub-slab vacuum field extension diagnostics and bench testing results for common vacuum fans and a radial blower to determine what combination of blowers, suction holes and piping would provide the best choice for inducing a minimum sub-slab vacuum beneath the medical office building constructed in an older metal fabricating building¹. The results and evaluations of all the diagnostic measurements and the final system performance are documented in this paper.

2.0 BUILDING FEATURES

The structure is a 9000 ft² one story slab on grade building that is being used as a medical facility. The original structure is at least 80 years old and was used by a manufacturer and fabricator of metal products. The suspected source of contaminants were the solvents used in the metal products manufacturing. Concentrations of Trichlorethene (TCE) exceeded New Jersey's Non Residential Screening Standards for indoor air, sub slab soil gas and ground water.

The building appeared to be constructed over four separate original foundation areas. The building underwent extensive renovation in 1996 when a second 6" slab was constructed 18" above the original slab. The fill that was used between the slabs was a matrix of low permeability clay, stone, concrete and coal tar that was mechanically compacted. The roof is supported by a large wood truss system and interior support columns. Air flow within the office is influenced by three separate HVAC systems. The building is very poorly insulated; in fact there is no insulation above the drop sealing. There is obvious air leakage out of the top of the building through broken upper windows when the building is warmer than the outside or from wind passing over the two 12" turbine attic ventilators. This loss of air out the top of the building creates vacuum on the lower levels of the building that draws in soil gases.

3.0 DESIGN GOAL

ASTM has defined the amount of sub-slab depressurization for vapor intrusion mitigation (VIM) to be between -6.0 pascals (-0.024" water column (W.C.)) and -9.0 pascals (-0.036" W.C.). The authors past experience has demonstrated that a minimum sub-slab vacuum of -2.0 pascals (-0.008" W.C.) to -4.0 pascals (0.016" W.C.) successfully stopped soil vapor intrusion. The design goal for this project was a sub-slab vacuum field between -2.0 pascals (-0.008" W.C.) and -4.0 pascals (-0.016" W.C) with a minimum cold weather performance of -1.0 pascals (-0.004" W.C.) throughout the footprint of the building. ^{2, 3}

4.0 CALIBRATING TEST EQUIPMENT

WPB performed fan bench testing using a four inch inline multi-pitot flow grid that was designed for balancing heat recovery ventilators to determine airflow. The flow grid was calibrated by carefully making traverse pitot tube measurements as defined by ASHRAE 1989 Fundamentals ⁴ in ten feet of four inch piping that the flow grid was placed in and having a fan

draw varying amounts of air through the pipe₃ while resistance was applied to the pipe opening. A second calibration measurement was made by enlarging the inlet of the pipe and connecting it to a 24" box with an open end. The airflow into the box was measured with a new Alnor Lo-Flow 6200 balometer that has an accuracy to 3% plus or minus 5 SCFM. All calibration measurements and bench testing included atmospheric factors to determine the air density to determine the SCFM. Field measurements used a standard air density of 0.0765 lb.ft³. The static vacuum and pitot tube measurements were made with an Energy Conservatory DG-700 digital micro-manometer capable of readings to 0.1 pascal with accuracy variation less than 1%. An MA-Line 12835 was used for static measurements greater than 1200 Pascals. This instrument read within 2% of the DG-700.

An Alnor RVA+ and Extech AN100 vane anemometer were placed inside PVC fittings. The units were then connected to ten feet of 4" piping that had the flow grid installed to determine the actual airflow as a fan varied the airspeed. The actual SCFM was divided by the feet per minute air velocity to determine the multiplier used to determine airflow at different velocities. The multiplier was also determined for the velocity obtained from a pitot tube that was placed in the center of different diameter pipes using the flow grid to determine the actual airflow.

5.0 DIAGNOSTIC PROCEDURES

In order to determine blower, piping size and suction point locations for an Active Soil Depressurization (ASD) system, extensive sub slab vacuum field extension (VFE) tests were conducted in October 2009 to determine the necessary sub-slab airflow to meet the sub-slab vacuum goals.^{1, 7, 8, and 9}

Test suction holes were made by core cutting a 2 ½ inch (64 mm) hole through the floor slab and then auguring out soil to a depth of approximately 18 to 20 inches. In general the test suction holes were located where future suction points could be installed. Smaller 5/16" test holes were drilled at varying distances in an expanding radius from the suction hole.

Air was drawn out of the suction holes using a Ridgid 6.5 HP shop vacuum with a speed controller. The micro-manometers were used to measure ten second average pressure differentials at the test holes depicted by "T#" in the floor plan in Figure 2. The initial testing was done on a Saturday afternoon and week day evening when the building was not occupied to accommodate the owner's request. With the vacuum off, a digital micro-manometer was used to make multiple 10 second measurements at each test hole which were averaged to define the baseline pressure differential between the building and the underlying soil. The sub-slab pressure differentials ranged from -0.0002" W.C. to +0.0056" W.C., with an average of +0.0028. Compensating for the baseline pressure differential should be included in the design process. Night measurements were preferred to minimize external wind affecting the sub-slab vacuum measurements.



Figure 1:
Airflow Measurements

The vane anemometers were used to measure the airflow induced by the shop vacuum while the sub-slab vacuum 12” from the suction hole was measured. The vacuum measurement 12” from the test hole approximates the distance the suction hole will be dug out during the system installation.

The soil resistance data was combined with the piping resistance using three anticipated piping sizes and equivalent pipe length to provide a plot in a spreadsheet graph. The total resistance was compared to the performance of different blowers to determine the optimal vacuum fan and the optimal number of suction holes the blower should draw on.

Table 1 below lists the vacuum achieved from the suction hole when the maximum vacuum was applied to the 2.5” suction holes and the vacuum in the suction hole when a five inch suction hole that had approximately 1.0 ft³ (7.5 gallons, 28 liters) of sub-slab material was removed. Note the high soil vacuum readings of three of the fully excavated 5” suction holes and the airflow out of these same holes varied from 19 CFM to 71 CFM.

Suction Hole	S1	S3	S4	S5	S7	S6	S9	S12	S16
Hole Size	5”	2.5”	2.5”	2.5”	5”	2.5”	5”	5”	5”
Maximum CFM	71	49	17	20	131	12	19	47	136
Vacuum 12” away	33”	5.3”	44”	43”	14”	6.7”	32”	37”	15”

Table 1: VFE Test Hole Results

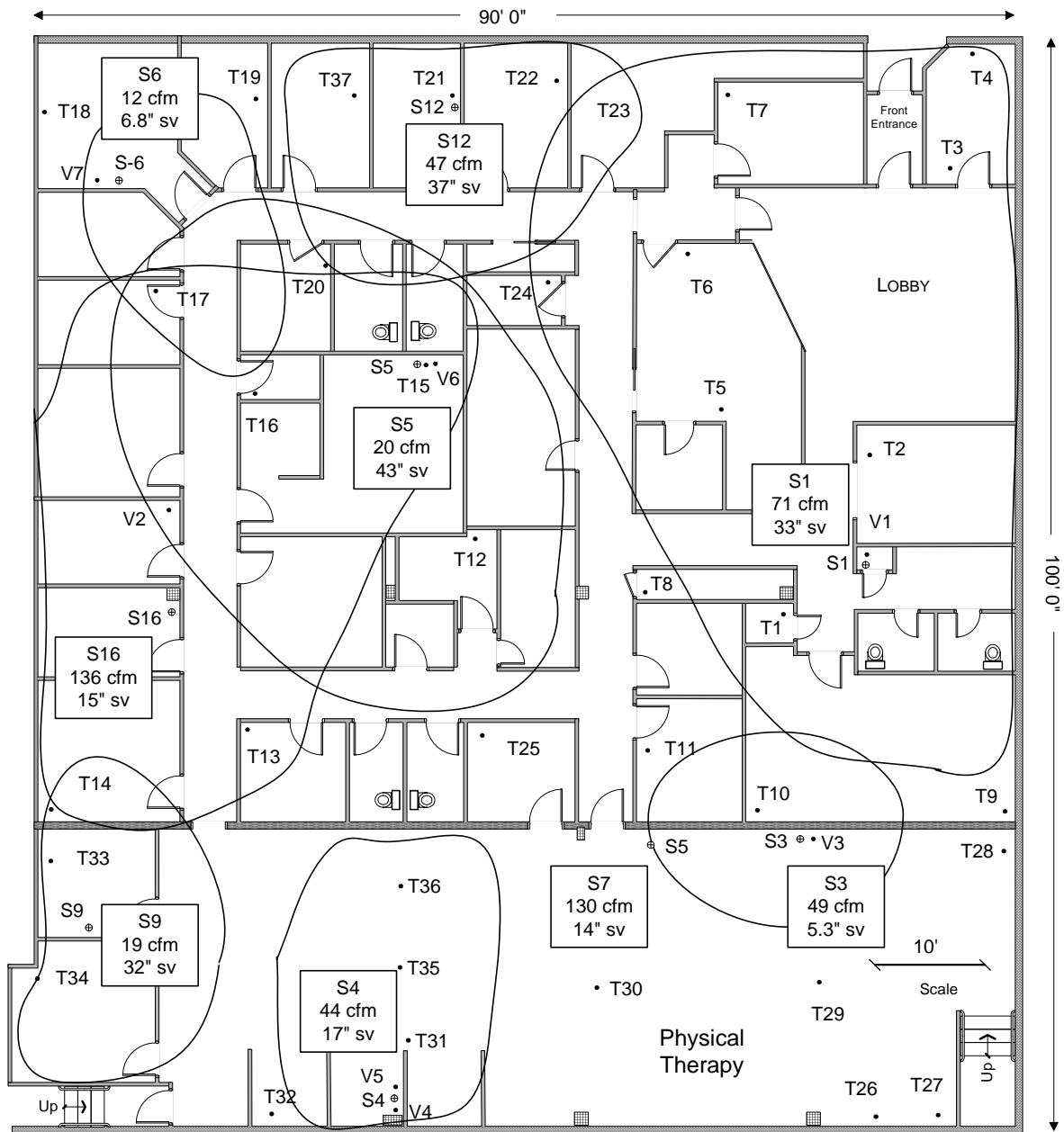


Figure 2: Suction Test Hole CFM & Static Vacuum (sv) & PFE to - 0.008"

The vacuum field extension (VFE) test had airflow that varied from 12 CFM to 136 CFM. At maximum shop vacuum speed the area each suction hole generated at least - 0.004" W.C. is depicted in Figure 2 above. The total airflow from all the suction test holes in Figure 2 above is 528 CFM being induced by high vacuum suction and even with this amount of airflow the entire footprint has not been depressurized to the design goal. This information changed our original plan of using typical inline fans and high vacuum-low flow centrifugal blowers to high vacuum blowers with high air capacity. The higher operating cost of high vacuum, high air flow blowers

versus multiple inline fans and high vacuum-low flow blowers needs to be considered in the system design. The comparison of equipment and operating costs is described in Section 12.0 below.

Chart 1 below is a comparison between bench test results for a shop vacuum versus a common inline fan and centrifugal high vacuum-low flow blowers. PFE testing is often done with the vacuum operated at full capacity. If the data collected from this measurement is directly applied, the shop vacuum operating at full speed can easily over-predict the final system sub-slab vacuum because a shop vacuum achieves higher air flow and vacuum than typical active soil depressurization (ASD) system fans and blowers.

6.0 BENCH TESTING FAN PERFORMANCE

A bench test was performed for all the fans, blowers and the shop vacuum used in this study. Chart 1 below depicts the performance of three fans considered for this project compared to a Ridgid 6.5 HP shop vacuum. The HS2000 and the HS5000 are manufactured by RadonAway. The Force fan is manufactured Festa Manufacturing Enterprises. Bench testing was done by placing the flow grid in ten feet of 4" piping. The airflow resistance is increased from open pipe to fully closed at the inlet side while the static vacuum and airflow SCFM is measured at twelve points (See Figure 2 and Figure 3 below). The complete testing method and results for all the common ASD fans are available at WPB's web site ³.

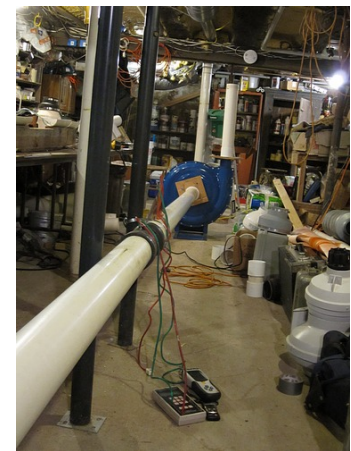
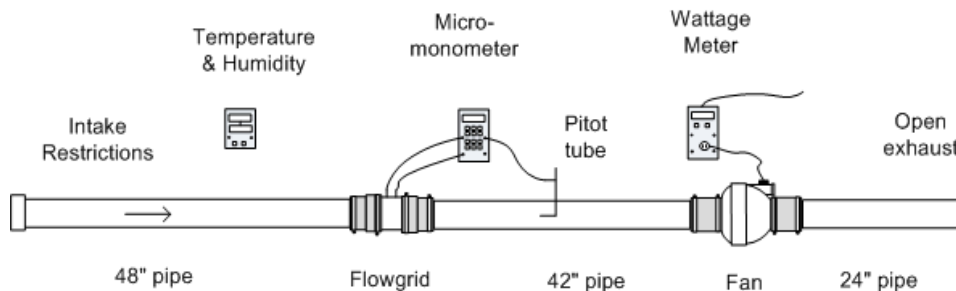


Figure 2:
Bench Testing Fan

Regenerative blowers are often chosen when greater than 10" of static vacuum is required. Although regenerative blowers can achieve static vacuum of 60 inches, their airflow output is low compared to radial blowers unless very large HP motors (greater than 3.0 HP) are used. Radial blowers can generate similar static vacuum as regenerative blowers if the RPM is high and the radial wheel is large enough. One of the smallest radial blowers from Flow Tech Fan (FTF) was chosen. The FTF QP 40/75 was purchased with a single phase 1.5 HP motor in order to bench test its performance. The results of this bench test are included in Chart 2 below. Although this fan can move 750 CFM with ten feet of open ended pipe, it will over amp the motor and cause premature motor failure. At the maximum 1.15



service factor of the recommended amperage of the motor, the fan will move 375 CFM and still produce over 16" of static vacuum.

Figure 3: Fan Bench Test Setup

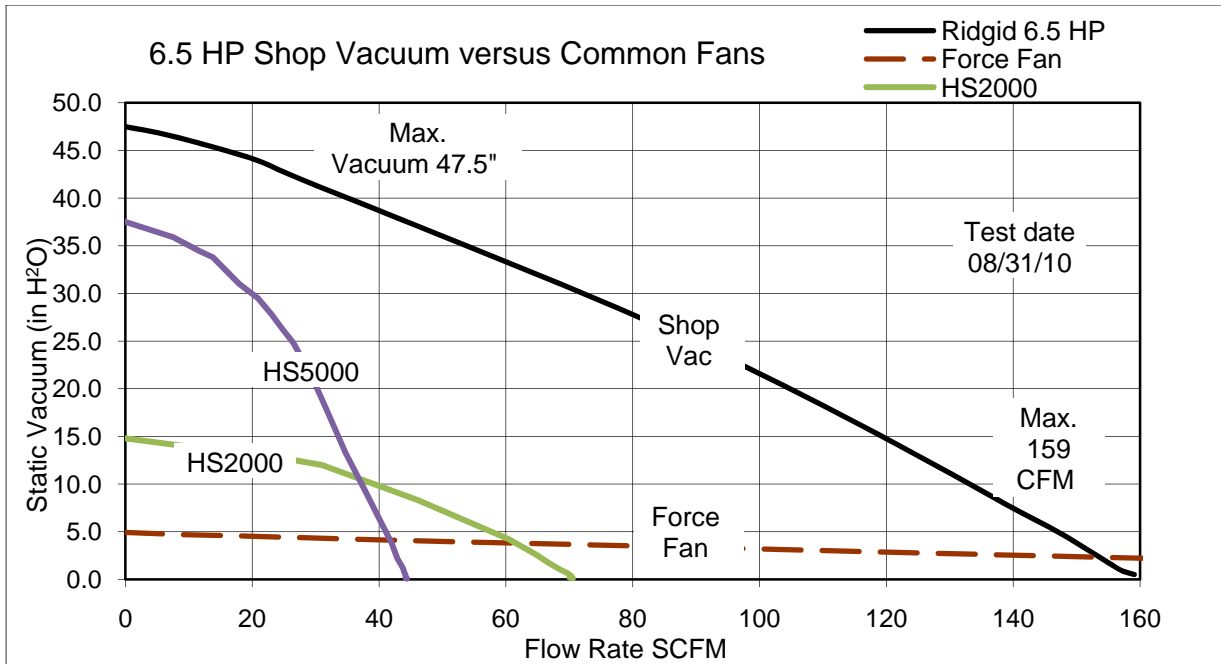


Chart 1: WPB Bench Test of Shop Vacuum versus typical High Vacuum Fans

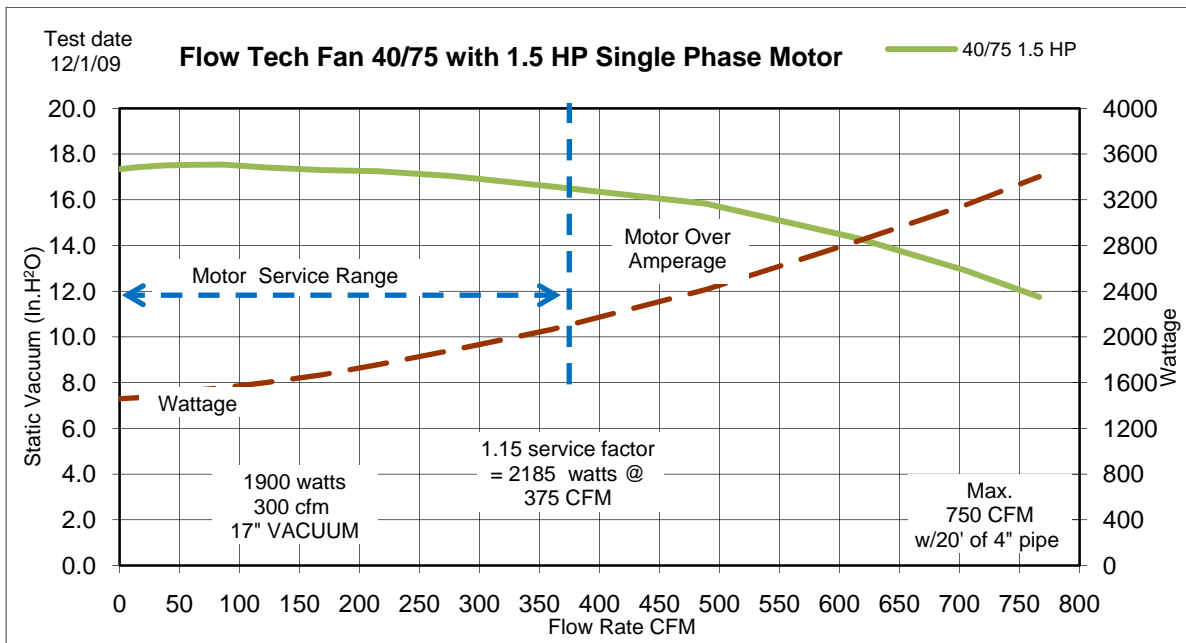


Chart 2: WPB Bench Test of FTF 40/75

7.0 PLOTTING TOTAL RESISTANCE VERSUS FAN PERFORMANCE

Plotting the sub-slab, piping and pipe fitting resistance versus different fan performance significantly improves determining which fan or blower would be optimal for different system layouts. If just bench testing performance is used to determine the optimal blower, Chart 1 would indicate the HS5000 for airflows below 37 CFM, the HS2000 for airflows from 37 CFM to 60 CFM and the Force fan for airflows from 60 CFM to 160 CFM. This method however does not take into consideration the sub-slab and piping resistance that determine the airflow versus vacuum relationship at the suction hole.

A spreadsheet was developed that plotted the total resistance of the pipe, pipe fittings and the soil resistance at varying airflows using the data obtained during the diagnostic testing. In Chart 3 through Chart 9, the combination dashed and solid lines rising from the left bottom to the right top, represent the total resistance of sub-slab, piping and fittings that depict the increasing airflow as vacuum is increased. The solid line is determined from the shop vacuum airflow and static vacuum that is generated in the soil twelve inches from the suction hole. Because the fans being considered often had performance curves outside of the shop vacuum data, the dashed lines in the chart are the projected resistance at airflows greater or lesser than the diagnostic vacuum performance. Airflow resistance was measured for typical PVC pipe and fittings by WPB₃. The airflow resistance used in the spreadsheet charts was the sum of the resistance of three different pipe sizes and pipe lengths to approximate their contribution to the total resistance. By adjusting the pipe size and lengths, the influence of the piping resistance at different airflows could be depicted allowing a determination of optimal pipe sizing for each section of the piping from the fan to the suction hole. The suction hole total airflow resistance was plotted against fan performance curves that were determined from the fan bench testing performed by WPB. The fan producing the highest air flow against the total resistance produces the largest PFE for that suction hole.

When depressurizing low permeability soils, multiple suction holes are typically connected to a single fan because airflow yields from suction holes with low permeability soils are often 12 to 25 CFM and commonly used fans have higher airflows. Determining fan performance for multiple suction holes is complicated by the piping variation and airflow variation that occurs. The airflow resistance of multiple identical suction holes was plotted that included the airflow resistance of three different configurations of piping size, length and airflow to determine the total resistance of the multi-suction system. The airflow of each fan as it intersected the multiple hole resistance equaled the fan's total airflow against that resistance. Dividing that total airflow by the number of suction holes determined the average airflow that could be extracted from each identical suction hole of a multiple suction point system. This allowed comparison of fan performance if it was piped to a single suction hole or multiple holes that had similar sub-slab resistance. The expected sub-slab vacuum at the test holes using the plotted blower performance could be approximated by using the predicted airflow to determine the sub-slab vacuum at the suction hole and adjusting the test hole predicted vacuum proportionally the same amount as the difference in the suction hole vacuum. Previous other projects had confirmed that the sub-slab vacuum in test holes with tight slabs tended to change linearly.

Chart 3 and Chart 4 below depicts two tests performed at test hole S1. S1 was a 2.5" suction hole excavated down 18" to 24" and had a maximum airflow of 101 CFM at -11.8" W.C. After S1 was enlarged to a 5" hole through the slab and five gallons of sub-slab soil was removed it had a lower airflow of 71 CFM at a higher static vacuum of 33" W.C.. It was speculated that the lower airflow with the full excavated pit may have been due to the larger pit being dug horizontally while the smaller pit was dug primarily vertically down below the fill that was brought in to raise the slab during the last building renovation or the close proximity of S1 to a floor drain system that was back filled as part of the renovation. If the maximum airflow of 71 CFM measured from the shop vacuum is used and compared to the fan bench test results from Chart 1, the best fan would be a Force fan. Using the total resistance compared to blower performance comparison in Chart 3 and Chart 4, the HS series blowers will draw about two times more air from the S1 suction hole than the Force fan (42 CFM vs 14 CFM and 35 CFM vs 19). The HS2000 would be the better choice even if two identical S1 suction holes were connected to a single fan.

In Chart 5 below, the shop vacuum test at S3 had a maximum airflow of 49 CFM at 5.3" W.C.. If two identical suction holes were piped to an HS2000, the average airflow of each suction hole would be only 31 CFM. If a Force fan was used two identical holes would only average 35 CFM and four identical holes would average 30 CFM per hole. The FTF 40/75 is predicted to induce an average of 83 CFM from each of four identical S3 suction holes. This would be the maximum airflow of this fan. Two Force fans could be used but one FTF 40/75 would induce 2.7 times more airflow. The PFE from S3, illustrated in Figure 2 above, was very limited at 49 CFM. The FTF 40/75 would be the better choice in this case because it could induce greater airflow than the shop vacuum produced. The electrical cost for the single FTF 40/75 would, however, be three times more than two Force fans.

Chart 6 below depicts the performance of S7 suction hole which is located adjacent to S3. The maximum shop vacuum airflow was 131 CFM at 14" W.C., which is approaching the shop vacuum's maximum flow capacity. Both S3 and S7 were located near an interior block wall that was likely allowing this greater airflow and reduced PFE because of sub-slab air leakage into the block wall that was exposed below the slab. A Force fan drawing from two identical S7 suction holes would produce an average of 41 CFM from each hole and an average of 30 CFM if it was drawing from four S7 holes. If an FTF 40/75 was used, it could produce an average flow of 57 CFM from 7 identical suction holes. Two Force fans could be used but 30 CFM is almost half the airflow of the 57 CFM average of the FTF 40/75.

Chart 7 below depicts the total resistance of S9. The shop vacuum maximum airflow was 19 CFM while generating 32" W.C. of vacuum in the soil. In this case, the Force fan moves the least airflow. The Chart indicates the HS5000 can remove an average of 15 CFM of airflow for two S9 suction holes. The HS2000 or the HS5000 could move an average of only 10 CFM for four S9 suction holes. The FTF 40/75 could produce 13 CFM average for 7 suction holes or only slightly less 12.5 CFM average for 20 identical S9 suction holes. This small difference is because the performance curve for the FTF 40/75 is almost straight up until it reaches 200 CFM of airflow. This is verified by reviewing the performance in Chart 2. If there were eight suction holes identical to S9, then four HS5000's would outperform one FTF 40/75. In this case the FTF

40/75 would use about 1075 watts while four HS5000 fans would use about 1750 watts, cost twice the purchase price, and have higher installation cost and less performance.

Chart 8 below depicts the total resistance of S12. The shop vacuum maximum was 47 CFM at 37" of vacuum. The Force fan would move an average of only 11 CFM from two suction holes. An HS2000 can move an average of 19 CFM from two identical S12 suction holes. The FTF 40/75 however could move 28 CFM from eight suction holes or at its maximum operating flow, 23 CFM from 15 identical S12 suction holes. In this case the FTF 40/75 connected to eight suction holes would use more power at about 1200 watts while four HS2000 fans would use about 700 watts, but cost twice the purchase price and have higher installation cost and less performance.

Chart 9 below depicts the fan performance at S16. The maximum shop vacuum airflow was 136 CFM at 15" of vacuum. The HS series fans would not be appropriate for this easy flow. The Force fan would pull about 45 CFM from each of two identical S16 suction holes and 35 CFM average if it was drawing from four identical suction holes. The FTF 40/75 could move an average of 50 CFM from seven identical S16 suction holes. In this case four Force fans using a total of 920 watts could be used to provide comparable performance to one FTF 40/75 fan using 1470 watts.

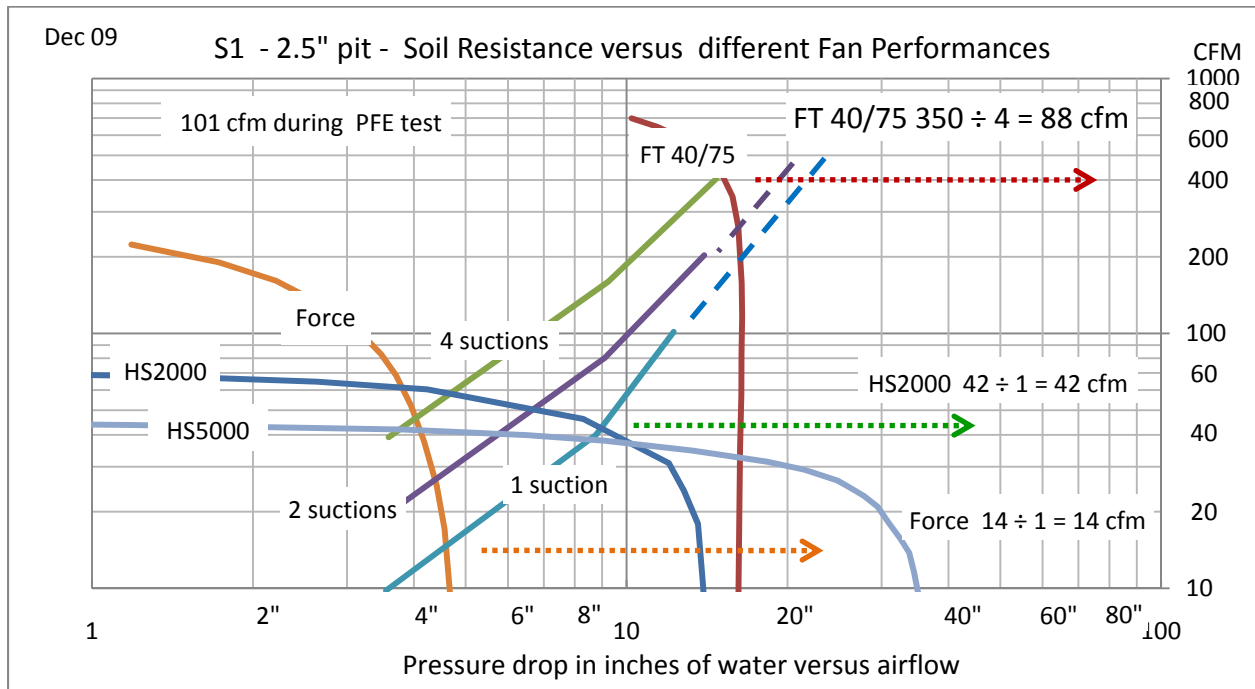


Chart 3: S1 with 2.5" suction hole

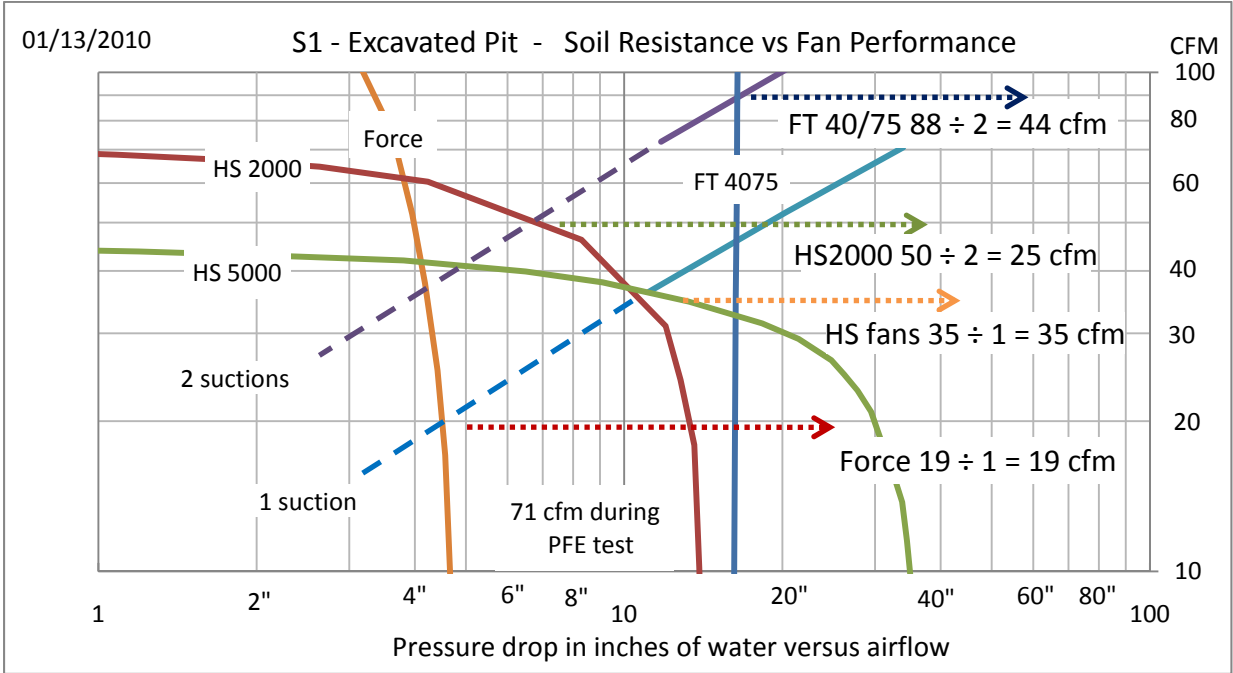


Chart 4: S1 with 5" full excavated suction hole

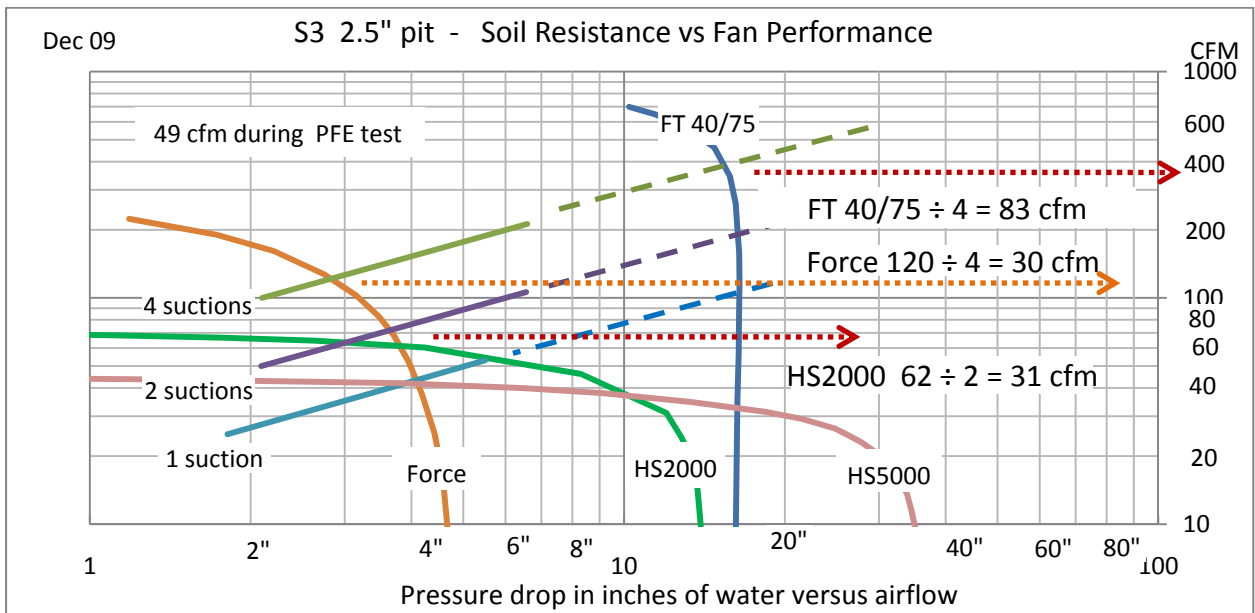


Chart 5: S3 with 2.5" suction hole

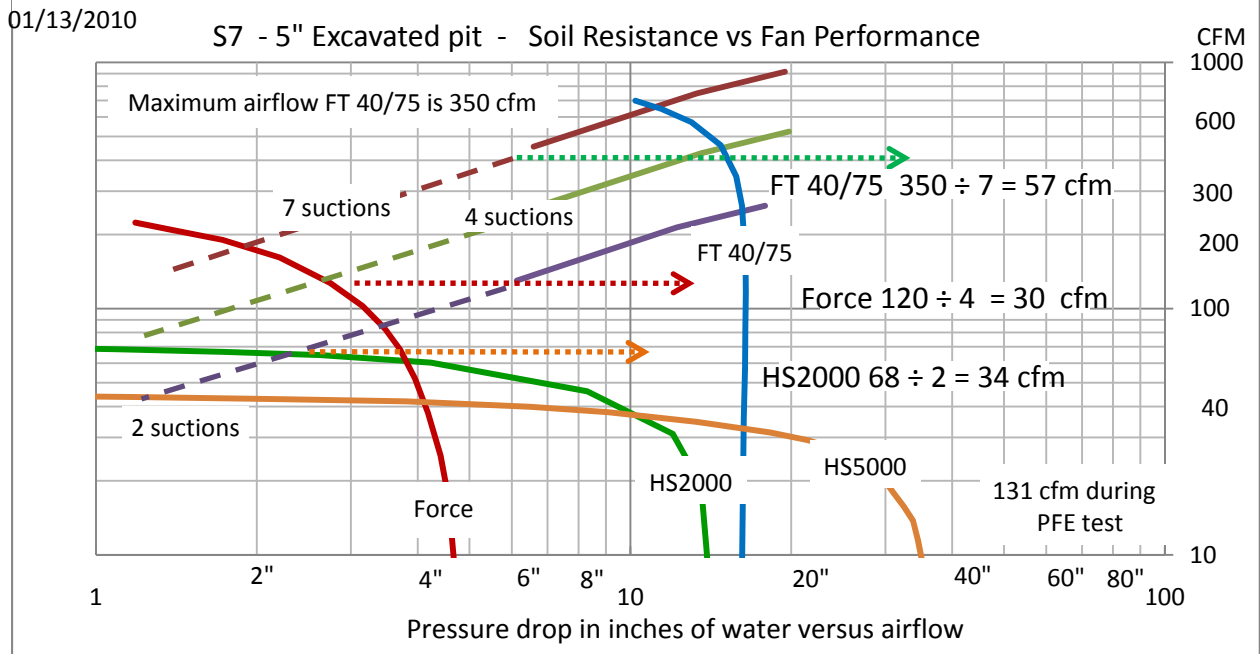


Chart 6: S7 with 5" full excavated suction hole

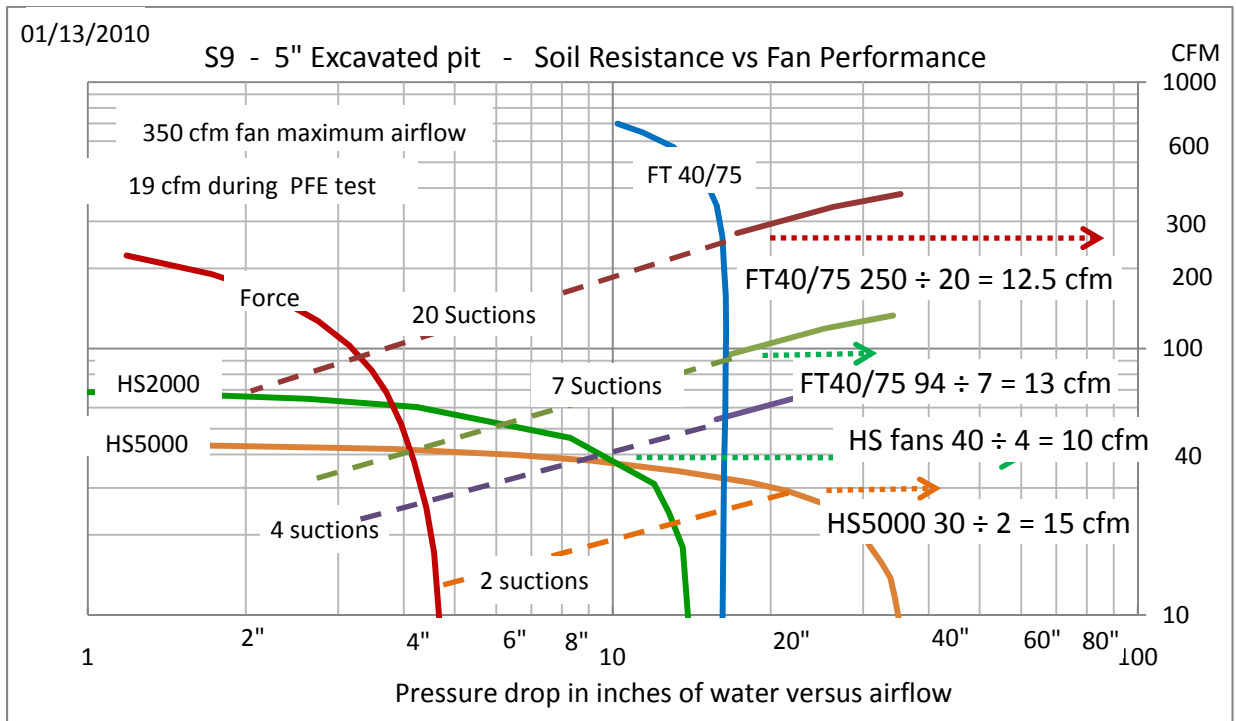


Chart 7: S9 with .5" full excavated suction hole

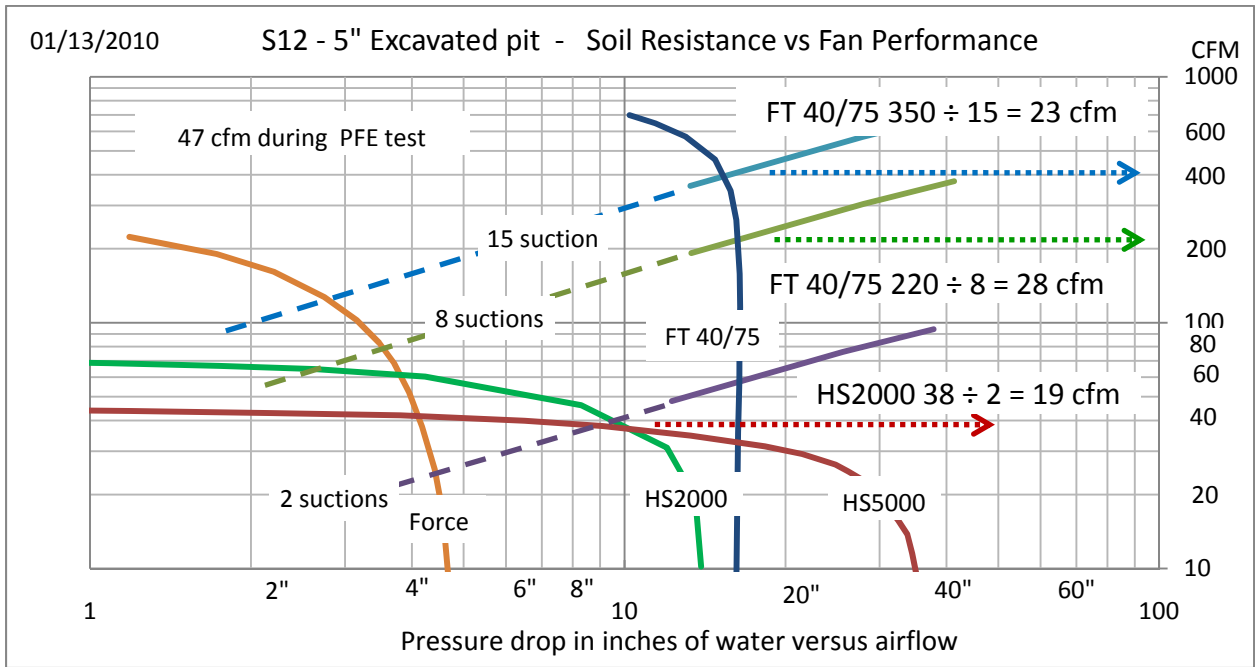


Chart 8: S12 with 5" full excavated suction hole

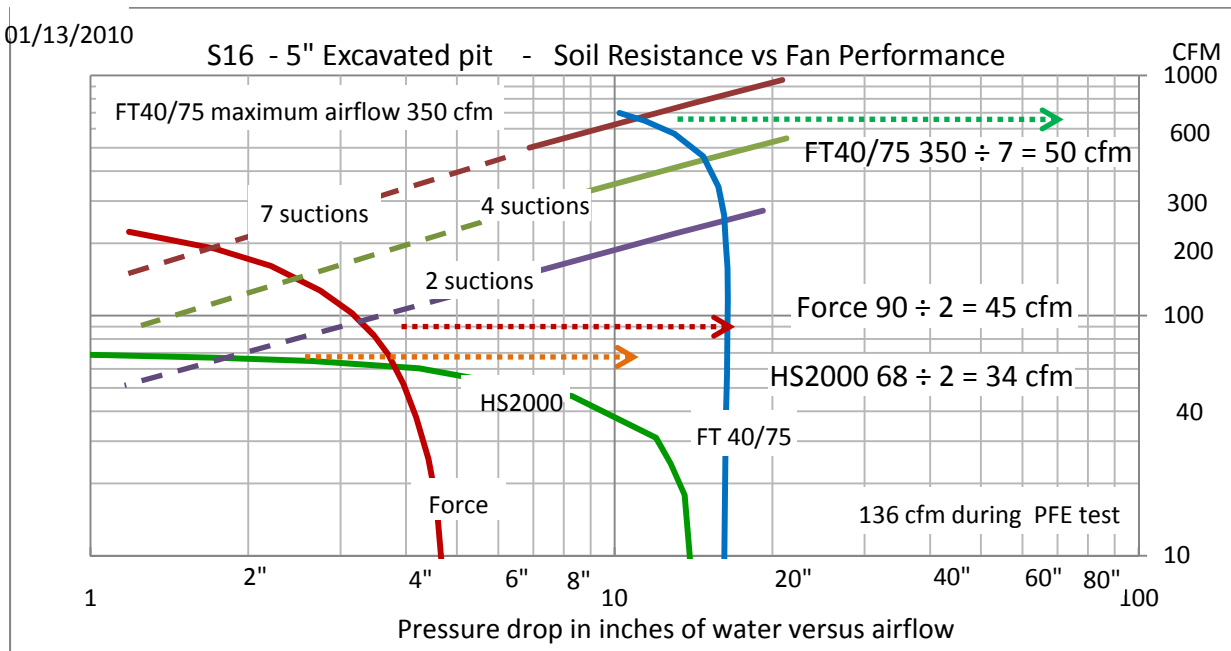


Chart 9: S16 with 5" full excavated suction hole

8.0 BLOWER SELECTION

Table 2 below lists the performance comparisons of all the blowers considered. The Regenerative Gast R5 moves 110 CFM at 30" of vacuum but requires a 2.5 HP motor. The FTF 48/75 can move more than twice the air (250 CFM) at almost the same vacuum (27") and yet uses a smaller motor (2.0 HP). The FTF 40/75 can move ten times the airflow of the HS5000 (300 CFM versus 30 CFM) and almost the same vacuum (18" versus 20").

If all the airflows are added up from each of the test suction holes in Figure 2, it totals an airflow of 528 CFM. The suction areas that achieved the design goal however did not cover all of the building footprint. In order to cover the entire footprint it was apparent that more than 600 CFM needed to be extracted from the sub-slab space. During the diagnostics, there were no obvious areas that could be sealed in order to reduce subslab to building airflow. It was unknown during the design phase if any slab leakage could be discovered and sealed to reduce the total necessary airflow. If only HS2000 and HS5000 series fans were used with an average airflow of 40 CFM per hole it would require 12 to 15 fans to obtain 500 to 600 CFM of airflow. To reduce the number of blowers required to obtain the necessary high airflow removal from the soil it was decided to use two 1.5 HP radial blowers rather than multiple HS and Force fans. Regenerative blowers were considered because they can obtain high static vacuum but their airflow using the same HP motors is considerably less. The Gast R5 regenerative blower in Table 2 moves 112 CFM at -30" W.C. of vacuum with a 2.5 HP motor. The FTF4875 moves twice as much air at 250 CFM at -27" W.C. with a smaller 2.0 HP motor. To provide the responsible party with the lowest operating power requirement, 1.5 HP blowers were chosen. The 1.5 HP blower FTF 40/75 was chosen because it had the highest CFM rating (300 CFM) at greater than -16" W.C.

Type	Fan Model	HP	Wattage	Orifice	Operating CFM	in H ² O
Inline	Force	0.37	274	4"	100	3"
Vacuum motor	HS2000	0.23	170	2"	40	10"
Vacuum motor	HS5000	0.5	370	2"	30	20"
Radial	FTF 40/75	1.5	1110	3"	300	18"
Radial	FTF 40/100	2.0	1480	4"	400	18"
Radial	FTF 48/75	2.0	1480	3"	250	27"
Radial	NYB 1404A	1.5	1110	3"	290	16"
Radial	NYB 1704A	2.0	1480	4"	320	22"
Radial	Atlantic ABC-400	2.0	1480	4"	300	10"
Regenerative	Gast R5	2.5	1850	2"	112	30"

Table 2: High Vacuum Fan Comparisons

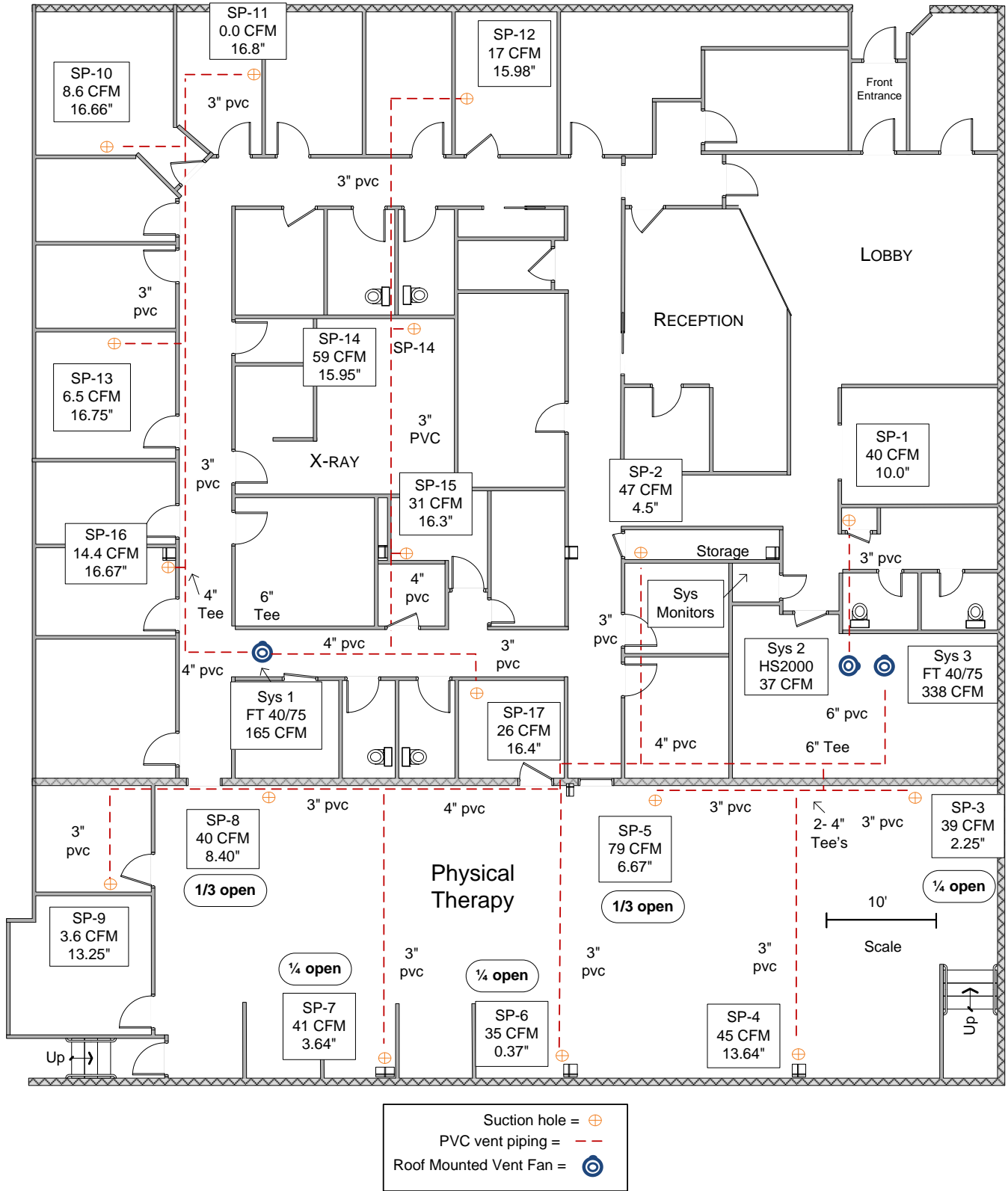


Figure 4: Final Mitigation Suction, Piping and Fan Locations

9.0 SYSTEM LAYOUT

The final system piping runs, fan locations and suction hole locations are depicted in Figure 4 above. Two FTF 40/75 fans with 1.5 HP three phase motors were used. See photos of the fans mounted on the roof in Figure 5 below. Each of these fans was connected to eight suction holes. Note that suction holes S1 and S2 could have been connected to blower system 1 which was running at half its airflow capacity. There was a bearing wall in the attic that divided S1 and S2 from system 1, making the piping run difficult. To gain additional capacity for the high airflow suction holes in the Physical Therapy Room, an HS2000 fan was used for suction hole S1. Six inch piping was routed from each of the blowers to a six inch tee with four suction holes on each side of the tee. Four inch piping could have been used at the fan but at the highest blower airflow, 50 equivalent feet of piping was calculated to reduce the static vacuum two additional inches more than if six inch pipe was used. Four inch piping was routed from the 6" tee towards the suction points with three inch piping routed down to the suction holes.



Figure 5: FTF 40/75 Installed on the Roof

10.0 SYSTEM ACTIVATION

Each of the suction point riser pipes were sealed air tight in the slab with urethane sealants that were allowed to cure before system start up. Each of the suction holes had a sliding damper to allow airflow adjustments after the system was operating. See Figure 9 below. Adjusting airflow with high vacuum systems requires very small movements of any damper. A radial blower without sufficient resistance can allow a motor to operate at higher amperage than its maximum service factor. In Chart 2 above, the FTF 40/75 could move 750 SCFM but the amperage is 150% of its rated service factor. Each of the blowers was started while measuring the amperage draw. If the amperage draw was above the service factor, the motor could over heat and shorten its life. The piping dampers were progressively closed until the amperage was within the recommended range. The airflow in each riser was measured using a pitot tube and a digital micromanometer. See Figure 6 below. Each time the dampers were adjusted, the

amperage needed to be checked to ensure the system was operating within the rated maximum service factor.



Figure 6: Pitot
Airflow Measurement



Figure 7: System Vacuum Gauges

Three magnehelic vacuum gauges were installed adjacent other building monitoring equipment in a storage room. This allows the vacuum induced by the motors to be easily checked. See Figure 7 above.

11.0 SYSTEM BALANCING

During start up, high airflows in some of the PVC riser pipes and whistling noise from behind the sheetrock indicated significant slab leakage was taking place. Large leakage was found and sealed around the columns that are close to suction points 15 and 16. The sheetrock surrounding the columns was removed, revealing a plywood form around each of the vertical columns with an unsealed 2” gap that was leaking through the openings in the plywood forms. A suspended backer rod and urethane seal was installed to seal the gap. Two additional columns required the same sealing procedure.

Sections of carpet that abutted the pilaster and exterior floor wall joint in the Therapy Room was pulled back and small amounts of chemical smoke was puffed along the joint to determine if room air was leaking into the sub-slab. See Figure 8 below. If the slab was leaking, the carpet baseboard and sheet rock was removed and the perimeter sealed. Upon completion of sealing, the sheetrock was reinstalled and refinished with matching paint.

After sealing was complete, all the pipe valves for each of the suction holes for system 1 could be left open because the total airflow was only 165 CFM, which is well below the motor service factor. The suction holes in the Therapy Room that adjoined the interior block wall, S3, S5 and S8 were moving excessive air. The pipe valves for these suction holes were adjusted to maximize the weakest sub-slab vacuum measurements. See Figure 9 below. After each adjustment of the valve, a pitot tube and the digital micromanometer were used to determine

what the new piping airflow was and the vacuum in the weakest vacuum test holes was re-measured. The valves were critical to maintain a balance between the maximum airflow the FTF 40/75 could move without having the amperage be greater than its service factor and obtaining the optimal vacuum in the weakest vacuum test holes. An example is the test hole T-10 in the conference room did not meet the design goal. The gate valve damper at suction point S-5 was opened and the vacuum at T-10 increased to $-0.005''$ W.C (-1.25 pascals). Other valves were then dampened down to restrict airflow so that the blower motor service factor was not exceeded.

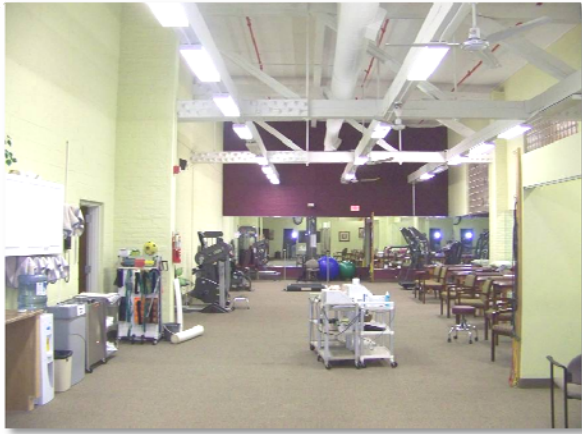


Figure 8: Physical Therapy Room



Figure 9: Slide Gate Damper

All of the slab test holes exceeded the sub-slab minimum vacuum goal of $-0.004''$ W.C.. The sub-slab vacuum readings are depicted in Figure 10 below.

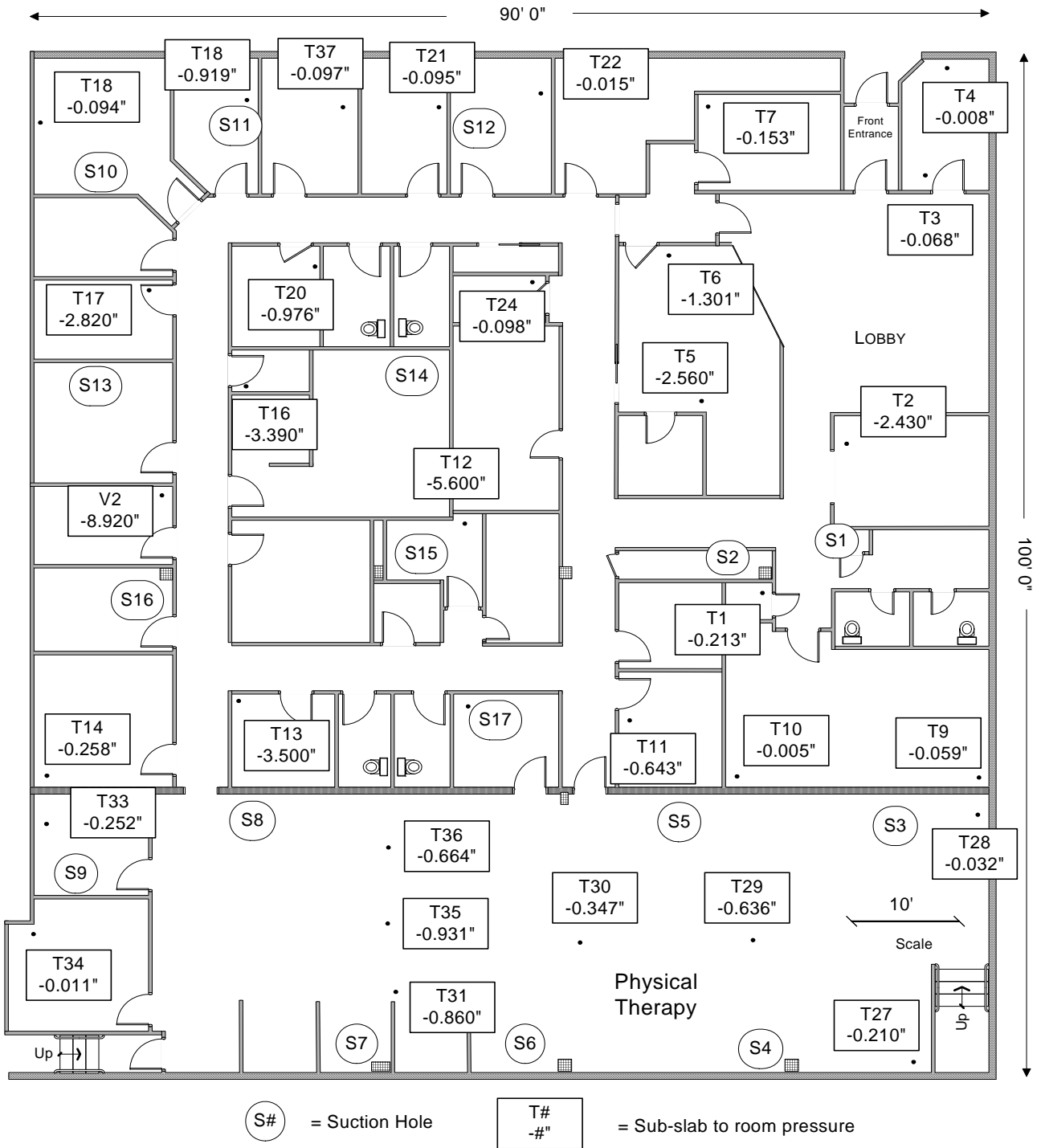


Figure 10: Final Sub-Slab Vacuum in Inches of Water Column

12.0 ENERGY USAGE

Typical ASD fans and blowers use single phase, 115 volt, fractional HP fans. Radial and regenerative vacuum blowers require much larger multi-horsepower motors. Three phase power, if available, provides significant energy savings over single phase. Comparison of wattage usage between the single phase and three phase 1.5 HP motor indicated the 3 phase motor used 67% of the energy of the single phase motor. Most large commercial buildings have three phase available. Using three phase also allows smaller wire size which often has to be routed long distances between the electrical source and the fans.

The fan costs and electrical consumption differences are listed below in Table 3 below. Note that the wattage usage of the two FTF 40/75 blowers was dependent upon their airflow.

	Fan	CFM	amps	Amps	amp	volts	watts	\$/KwHr	Cost/Yr
Sys 2	HS2000	40	1.82			115	209.3	0.18	\$330
Sys 1	FTF4075	160	3.26	3.11	2.99	115	1076.4	0.18	\$1,697
Sys 3	FTF4075	352	4.23	4.01	4.51	115	1466.25	0.18	\$2,312
Total CFM		552	Fan cost	\$5800		Yearly		Elec \$	\$4,339
Possible Fan Variation								Cost per month =	\$362
Fan	CFM	amp	Volts	watts	\$/KwHr	Cost/Yr	# Fans	Cost/Yr	
HS5000	30	3.8	115	437	0.18	\$689	4	\$2,756.25	
HS2000	40	1.8	115	207	0.18	\$326	3	\$979.19	
Force	100	2	115	230	0.18	\$363	3	\$1,087.99	
Total CFM		540	Fan cost	\$9600		Yearly		Elec \$	\$4,823
								Cost per month =	\$402

Table 3: Operating Cost Comparison

13.0 Summary

Designing an optimal active soil depressurization system for this buildings was complicated by the compacted low permeability sub-slab fill and inaccessible slab and bearing wall and column leakage that necessitated high airflow and high vacuum fans. 1.5 HP to 2.0 HP radial blowers with airflows up to 400 CFM can maintain static vacuum greater than 16" W.C. as illustrated in Chart 2. One radial blower can replace from three to ten inline or centrifugal blowers and provide comparable or higher static vacuums. Using fewer blowers reduces installation cost, provides maintenance savings and can in some cases use less power than comparable high vacuum centrifugal blowers. The higher electrical power requirements of radial blowers and the availability of three phase power however need to be considered during the design phase. Using spreadsheet graphs of multi-hole total system resistance versus fan performance allows direct

comparison of radial blowers versus more typical blowers and allows determination of the effect of different piping configurations on system performance. Measuring the fan motor amperage to keep the motor operating in its proper amperage range while measuring the sub-slab vacuum and adjusting the suction piping dampers is critical to obtaining optimal system performance. The individual responsible for the mitigation design must understand how to apply the raw PFE and airflow data to the selection of the blowers, number of suction points, suction point locations and pipe sizes. The authors found it helpful to test suction hole performance prior to fan and piping installation in order to verify that the previously selected fan and piping are still the optimal design. If suction point air flows varied significantly from the design projections, modifications could still be incorporated in the system design. Sealing slab openings is a critical component of increasing the sub-slab vacuum and reducing the necessary airflow so that the system is optimized and the fan motors are not operated higher than their service capacity. Radial blowers were an effective alternative to using multiple standard inline fans, low airflow-high vacuum fans or regenerative blowers for this building.

14.0 REFERENCES

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KEY WORDS

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