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Summary The propagation of low-amplitude air pressure transients within the drainage and vent systems of underground habitable structures may result in system failure due to trap seal loss and foul odour ingress into the occupied space. This paper develops the simulation of such transient response, and presents comparisons between predicted system air pressures and those monitored during the operation of the drainage and vent system in a large London Underground tube railway station. The use of the model to investigate system operating regimes ahead of installation as a direct design aid is demonstrated.

Drainage ventilation systems for underground structures II: Simulation of transient response

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List of symbols

- A Duct cross-sectional area (m^2)
- c Acoustic velocity or wave propagation speed in air ($m\ s^{-1}$)
- D Duct diameter (m)
- dp Pressure change accompanying air pressure transient (Pa)
- du Air velocity change accompanying air pressure transient ($m\ s^{-1}$)
- f Friction factor
- K_c Loss coefficient at closing vent entry due to tank surcharge
- K_{max} Loss coefficient taken to represent full closure of vent entry due to surcharge
- K_0 Loss coefficient at an opening vent due to tank emptying
- n Index controlling the loss coefficient variation during vent closure or opening
- p Air pressure ($N\ m^{-2}$)
- T_c Time to close vent entry due to surcharged tank (s)
- T_0 Time to open vent entry as tank empties (s)
- t Time (s)
- u Air flow velocity ($m\ s^{-1}$)
- x Distance from duct entry (m)
- Δp Pressure change during Joukowsky instantaneous flow stoppage ($N\ m^{-2}$)
- ΔT_c Time into vent closure operation in slow closure case (s)
- ΔT_0 Time into vent opening operation in slow closure case (s)
- Δt Time step governed by Courant Criterion (s)
- Δu_0 Velocity reduction during Joukowsky flow stoppage ($m\ s^{-1}$).
- ρ Air density ($kg\ m^{-3}$)

1 Drainage vent system design for below-ground structures

Habitable below-ground structures requiring drainage provision present a number of special challenges to the designer. If user satisfaction is to be maintained it follows that the prerequisite of above-ground building drainage networks that foul odours be prevented from entering the habitable space must be adhered to. In the more standard above-ground application this criterion of satisfaction can be met by providing a vent system that will limit the air pressure fluctuations caused by the passage of discharges down the building's vertical stacks, and the entrainment of an airflow that eventually discharges to the main sewer system. In below-ground structures this is insufficient due to the necessity to collect waste water and discharged solids at some low level within the structure, for subsequent pumping to the surface and discharge to the surface sewer system.

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Under these design constraints the options available are to provide low-level collection sumps for both waste water discharge and 'bilge' water ingress, and a separate and separately vented system of holding tanks for WC discharge collection. One solution to ensuring that no foul odour escapes into the habitable space from the sumps is a mechanical vent driven by a surface-mounted fan. Such a provision imposes a slight negative pressure on the overall network, thereby ensuring that foul air is removed from the sumps and the surrounding space. Figure 1 illustrates a typical below-ground structure drainage and vent system.

Above-ground drainage and vent systems are subjected to air pressure transient propagation as a result of the unsteady annular flows established in the vertical stacks by the random and user-dependent discharges of the various appliances connected to the system[1]. In the below-ground case transient propagation may occur for a range of reasons, including appliance discharge to holding tanks or sumps causing local air pressure fluctuations, changes in holding tank free air volume due to appliance discharge, changes in fan speed or fan shutdown due to power failure, or the generation of substantial pressure excursions if the holding tank or sump becomes surcharged. (The latter may occur due to excessive inflow to the sump or due to pump drive power or level sensing failure.)

2 Basis for a system simulation

The propagation of low-amplitude air pressure transients through a drainage and vent system within a below-ground structure may be modelled by the solution of the St Venant equations of continuity and momentum via the finite-difference based Method of Characteristics[2]. In order to provide a time-dependent solution it is necessary to identify suitable boundary conditions to represent the interface between drain and vent sections and a range of system fittings and equipment. The range of necessary boundary conditions was identified previously[3] to include the following categories:

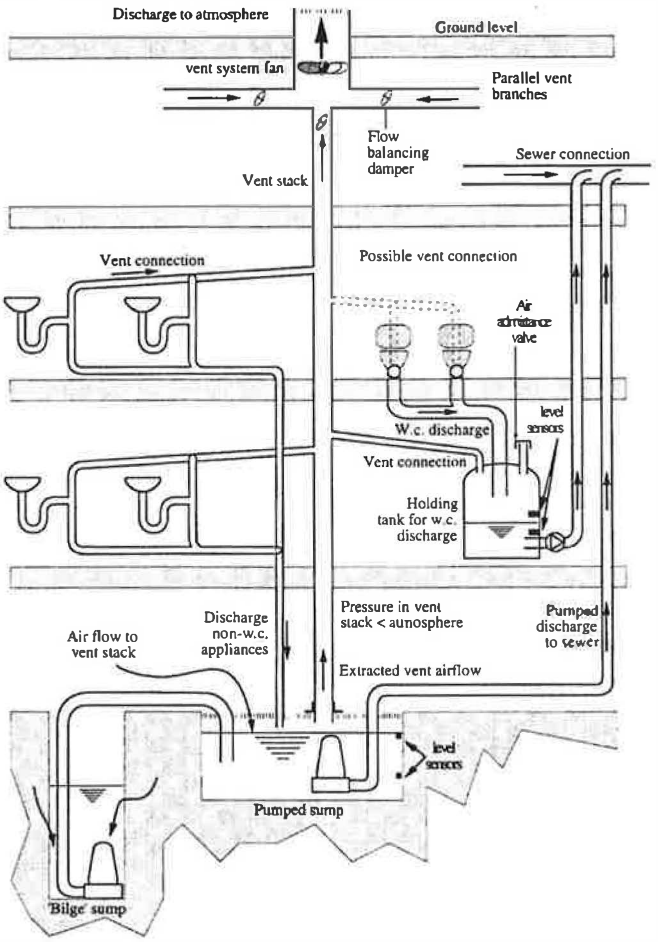

Figure 1 Schematic representation of the main components of a drainage and vent system appropriate for an underground structure

(*a*) Passive boundary conditions that arise as a result of the design of the system, including junctions, open and dead ends, constant pressure zones.

(*b*) Active boundary conditions that represent equipment connected to the network, in this case including the fan and its speed control mechanism, flow control dampers, air admittance valves and appliance trap seals.

(*c*) Boundary conditions that only arise as a result of system operation and are not represented by any item of equipment at a fixed location.

The boundary equations required to model categories (*a*) and (*b*) were dealt with previously[3]. The most significant category (*c*) boundary is that formed by a holding tank or sump surcharge. Surcharge effectively blocks the vent entry and drives the system operating point up the fan characteristic. This results in the propagation of a 'surge' type transient whose magnitude, if the closure of the vent is rapid, may be in excess of the system trap seal depths and causes failure of the system to prevent the release of odour to the usable space. Similarly the subsequent reopening of the vent entry at a sump or holding tank may generate pressure transients of similar magnitude but reversed sign.

The St Venant equations of continuity and momentum describe transient propagation within vent systems. They may be solved numerically via the Method of Characteristics[2], yielding the following total differential equations in terms of flow velocity u and acoustic velocity c:

$$du/dt \pm [2/(\gamma - 1)]dc/dt + (4fu|u|/2D) = 0 \qquad (1)$$

where γ is the adiabatic index of air, f is the Darcy–Weisbach friction factor, D is the pipe diameter and c is defined, in terms of the adiabatic index γ, the air pressure p and the air density ρ as:

$$c = (\gamma p/\rho)^{1/2} \qquad (2)$$

Note that the absolute value of air velocity in equation 1 ensures that friction always opposes motion.

2.1 *Mechanical vent entry closure/opening boundary condition*

Figure 2 illustrates a typical holding tank arrangement where, as this storage tank accepts WC discharge, a dedicated vent is provided, terminated in an air admittance valve, together with a through air flow connection to the main vent system and the surface-mounted fan. Under normal operating conditions a flow of vent air passes through the holding tank if the pressure in the dedicated vent is low enough to lift the air admittance valve diaphragm.

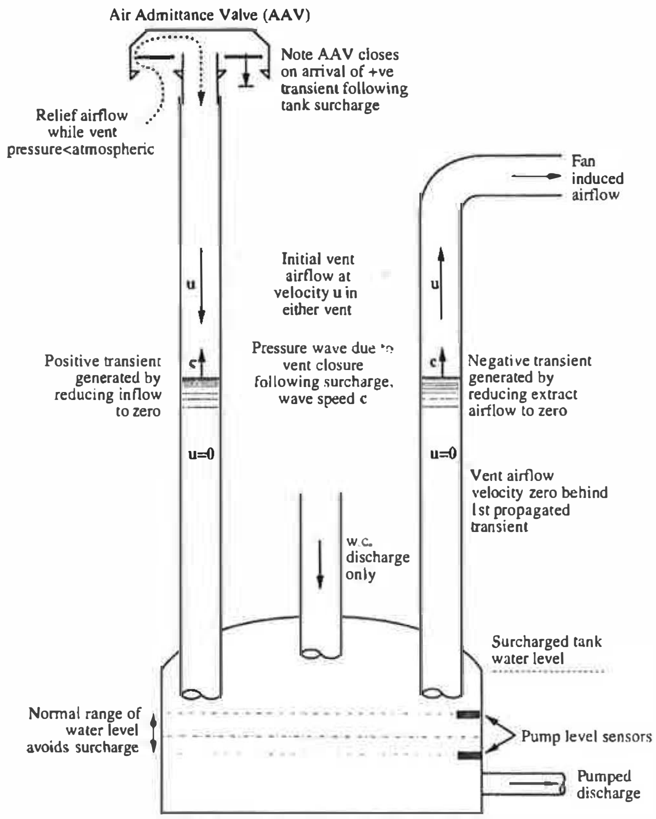

Figure 2 Holding tank boundary conditions, illustrating the propagation of pressure transients through the connected vents in the event of tank surcharge

If the holding tank becomes surcharged then this airflow path is closed and pressure transients resulting from a flow stoppage are propagated in both the dedicated vent and into the main vent system. In each case the boundary condition to be solved with the available characteristic once the vent closure is complete is

$$u_{\text{vent/tank interface}} = 0.0 \qquad (3)$$

Sump or holding tank surcharge may occur in the event of pump failure, sump level sensor failure or excessive inflow. This results in the propagation of large air-pressure transients through the system and, as a submerged vent entry acts as a dead end, a change in the system steady-state operating conditions. The air pressure transients generated, or the change in the system operating point, may well result in the depletion of appliance trap seals throughout the system.

The pressure transient generated as a result of instantaneous closure may be developed by reference to the pressure change across the transient. Application of the momentum equation across the wave front (Figure 2) in the absence of viscous forces yields

$$(p + dp)A - pA = cA\rho[-(c - du) - (-c)] \tag{4}$$

where the wave front has been brought to rest by the superposition of an equal and opposite velocity. This expression reduces to

$$dp = \rho c du \tag{5}$$

where dp is the pressure change accompanying a pressure wave that changes local flow velocity by du. This expression is perhaps better known as the Joukowsky relationship for the pressure change accompanying an instantaneous flow stoppage[4,5]:

$$\Delta p = -\rho c \Delta u_0 \tag{6}$$

where Δp is the pressure change due to the air flow stoppage, ρ is the air density, c is the acoustic velocity in air and Δu_0 is the local air velocity destroyed by the air flow stoppage.

(Note that if the acoustic velocity is 325 m s^{-1} and the air density is taken as 1.3 kg m^{-3}, then the pressure transient generated by reducing an airflow velocity of 1 m s^{-1} to rest instantaneously is approximately 40 mm of water gauge.)

It should be noted that the sign of the pressure change accompanying flow stoppage will depend on the flow initial direction. Hence in the dedicated vent the pressure transient will be positive, as the initial air flow was into the holding tank and Δu_0 was negative while the pressure transient propagated into the main vent network will be negative, as the vent closure prevented a continuous supply of air into the vent from the holding tank, or sump. These transients can deplete any upstream appliance trap seals.

In addition it is necessary to include the frictional regain in pressure following the eventual damping of the transient associated with vent closure. In the case of the connection from a holding tank or sump to the main vent network, the pressure at the closure site will tend to the manifold pressure at the fan connection, unless the transient completely depletes a trap seal within the network, in which case the evacuated trap will act as an air inlet to the system. In any event the pressure at the closure site will be below atmospheric. In the case of the dedicated holding tank vent with an air admittance valve termination it is possible for there to be 'trapped' positive pressure in the vent since the air admittance valve will close in response to the positive transient.

While the development above has concentrated upon instantaneous flow stoppage, it is unlikely that this will occur. However it is also necessary to understand that the full Joukowsky pressure transient will be generated if the closure is completed before any reflected transient can arrive back at the closure site to modify the process[6]. It is assumed that transients travel at the acoustic velocity in air at the local pressure. A major simplification in the study of low amplitude air pressure transients is that, unlike waterhammer pressure surge events, the pressure excursions are too small to affect the duct walls and hence changes in wave speed due to wall elasticity may be ignored. The 'round trip' time for a transient to arrive back at the closure site from an upstream reflector, in this case the closing air admittance valve, the fan or major duct junctions in the network, is referred to as the pipe period and closures completed in less than one pipe period are defined as rapid.

In many cases the geometry of the vent-to-sump connection will result in a less than rapid closure of the vent entry. The model represents slower vent closures by introducing a variable closing loss coefficient K_c to represent the increasing loss between the sump, or collection tank, and the vent pipe. Hence the vent pressure at the sump/vent interface may be defined as

$$p_{\text{vent/sump interface}} = p_{\text{sump}} - 0.5\rho K_c(u\,|u\,|)_{\text{vent/sump interface}} \tag{8}$$

and may be solved with the C^+ characteristic[3]. The loss coefficient during closure may be expressed as

$$K_c = K_{max}\Delta T_c^{\,n}/T_c^{\,n} \tag{9}$$

where K_{max} is the maximum loss coefficient achieved during closure (0 for a fully open entry and 60 000 for a fully closed entry, i.e. the latter figure is large enough to ensure a sensibly zero throughflow), T_c is the total time for vent closure, ΔT_c is the time into the closure and n is an index to represent the mode of closure.

When the sump water level drops, a submerged vent entry will reopen, allowing the re-establishment of the air flow from the sump. This scenario, which will again result in the propagation of large air pressure transients and a change in the system operating point, may lead to the depletion of appliance trap seals.

The model treats an instantaneous reopening of a vent entry in an identical manner to an instantaneous closure. The only difference is that as the air flow is re-established the pressure at the vent entry will increase, hence the negative sign in equation 6 is negated.

A slow opening of the vent entry is also treated in the same manner as a slow closure. The variable opening loss coefficient K_0 is given by:

$$K_0 = K_{max}[1 - (\Delta T_0/T_0)^n] \tag{9}$$

where K_{max} is the maximum loss coefficient achieved prior to the reopening of the vent entry (0 for a fully open entry, 60 000 for a fully closed entry), T_0 is the total time for vent reopening, ΔT_0 is the time into the reopening and n is an index to represent the mode of opening.

The choice of closing index n in equations 8 and 9 may be related to the rate of closure, or re-opening of the vent. If n is set to unity the vent loss coefficient changes in a linear fashion — probably an unlikely condition. In the closure mode the loss coefficient will initially change little; however the rate of increase in K_c will increase rapidly towards the end of the closure. This would be a realistic condition and would imply an index greater than unity. In the reopening case the loss coefficient will drop substantially as soon as some of the vent entry is exposed, and will then change slowly towards its fully open value. This would be represented by an index value less than unity.

3 Validation of below-ground structure drainage vent system simulation

In order to validate the simulation developed, it was necessary to monitor the transient pressure response of an installed below-ground structure drainage and vent system and to compare the measured transients with those predicted by the model for a range of operating conditions. It was decided that the drainage systems installed in London Underground stations represented a complexity of network suitable for this purpose.

3.1 Characteristics of London Underground station drainage and vent systems

The drainage networks installed within London Underground Ltd (LUL) stations are characterised by a relatively small number of appliances which, as a result of variations in staff levels/requirements, are used sporadically. In addition LUL station drainage networks receive large pump discharges from low-level collection sumps, the frequencies and magnitudes of which are dependent primarily upon the prevailing weather conditions. As a result of the layout of LUL station drainage networks, where normally aspirated vents are often ineffective, mechanical fans are employed to provide venting for low-level appliances, collection sumps and holding tanks. These characteristics clearly indicate that the empirical design methods developed for above-ground buildings are unsuitable for the design of LUL station drainage and vent systems. Therefore the efficient and economic design of such networks should be undertaken via a time-dependent numerical simulation based on the relevant governing equation of unsteady flow, capable of handling the complex interactions between the fan-generated air flows, the water-entrained air flows and the large air volumes contained within the collection sumps and holding tanks.

In order to validate the model, site tests were carried out on the mechanical vent system installed at Green Park Underground Station. This system, a schematic of which is shown in Figure 3, was selected as it was considered representative of LUL station vent systems in general and because the exposed vent pipes within the escalator machine rooms or staff-only areas greatly simplified the installation of the monitoring equipment.

3.2 System instrumentation and test methods

Fifteen pressure transducers, located regularly throughout the system, were used in conjunction with a PC-based data acquisition system to record air pressures within the vent pipes. In addition, a pitot static tube/pressure transducer arrangement was used in a number of different locations to monitor the air flow rates within the system.

To enable a wide variety of both steady and unsteady conditions to be monitored in a short time, events were initiated artificially.

4 Comparison of measured and predicted conditions

The air pressures and air flow rates predicted by the mathematical model compared very well with those measured on site, as shown in the following examples.

Figure 4 shows the measured and predicted variation in air flow rate, through the mechanical vent pipe connected to pump sump number 1, as the fan attains maximum speed via a series of intermediate speed settings. As can be seen, the predicted variation in flow rate compares very well with that measured. In both the measured and predicted cases, the relatively slow fan transition time (approximately 20 seconds) results in smooth changes in the air flow rate.

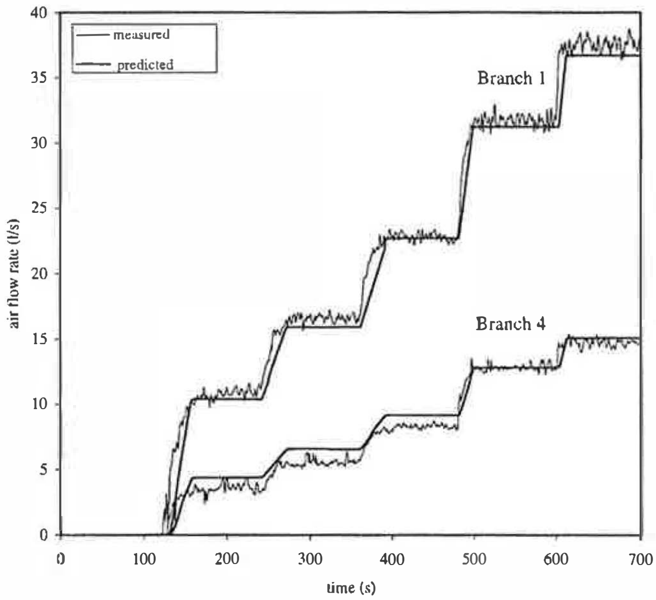

Figure 4 Measured and predicted variation in air flow rate through branches 1 and 4 (linking sumps 1 and 4 respectively to fan, Figure 3) with variable fan speed up to 2590 rpm

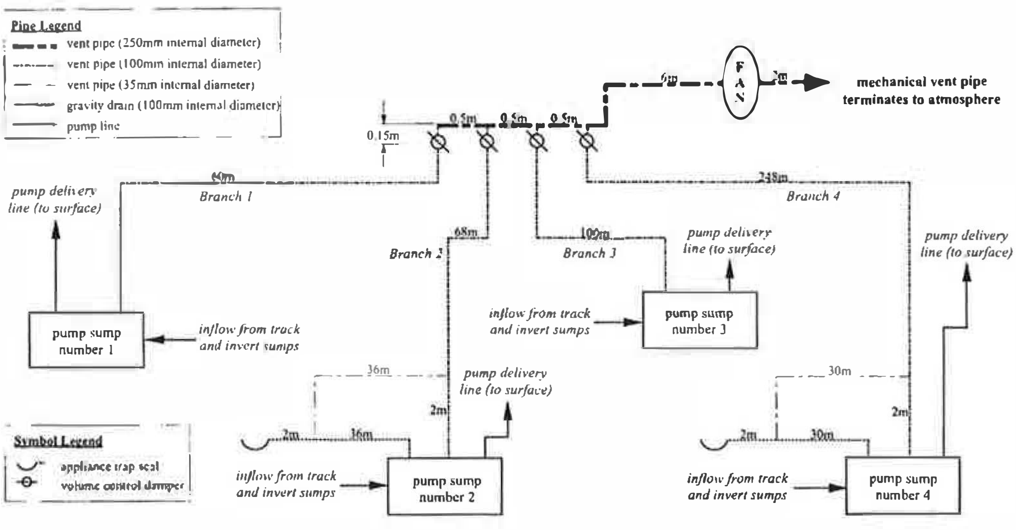

Figure 3 Mechanical drainage system vent network selected for monitoring purposes as installed at Green Park Underground Station, London

Similarly Figures 5 and 6 illustrate the pressure comparisons along branch 4, leading from pump sump number 4, as the fan starts up and attains full speed, including the effect at an appliance trap, and the pressure in branch 1, leading from pump sump number 1, as the fan slows to rest from full speed.

It should be noted that in these predictions the fan speed has been assumed to decrease linearly with time; however the fan inertia will tend to prolong rotation, explaining the slight variations between the measured and predicted pressure–time curves illustrated.

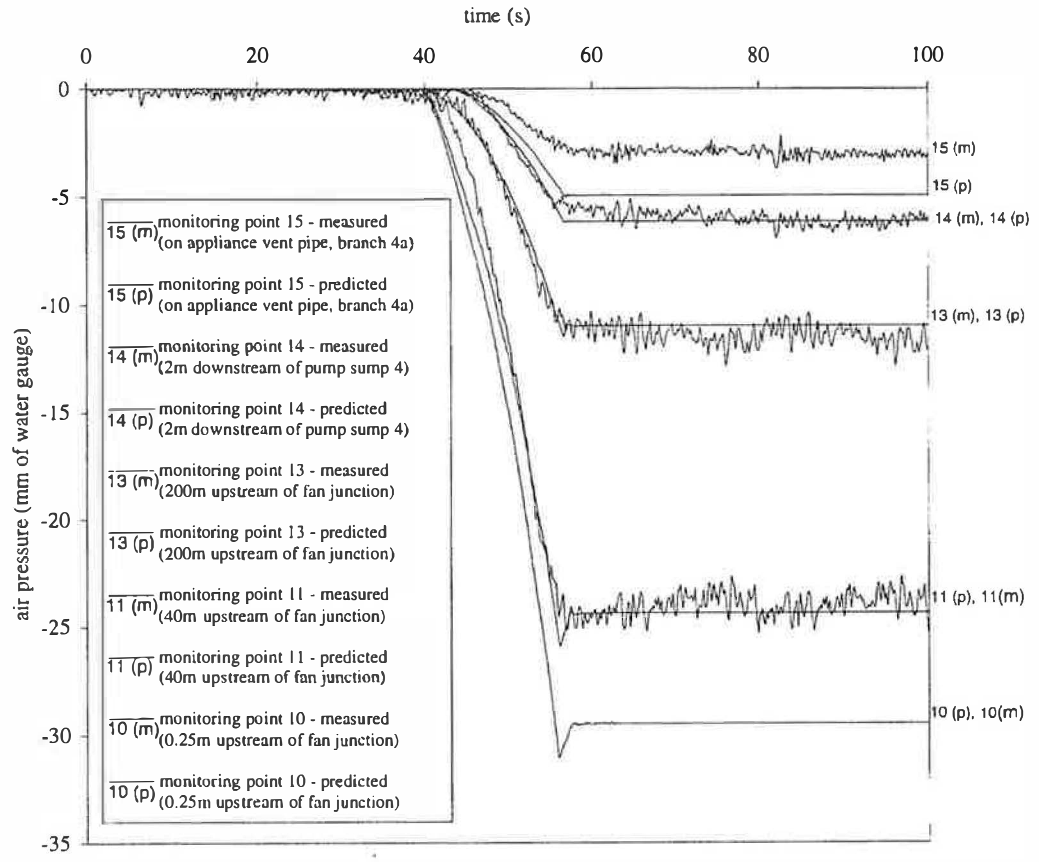

Figure 5 Measured and predicted variation in air pressure along branch 4 (linking sump 4 to fan, Figure 3) as fan accelerates to full speed set at 2590 rpm

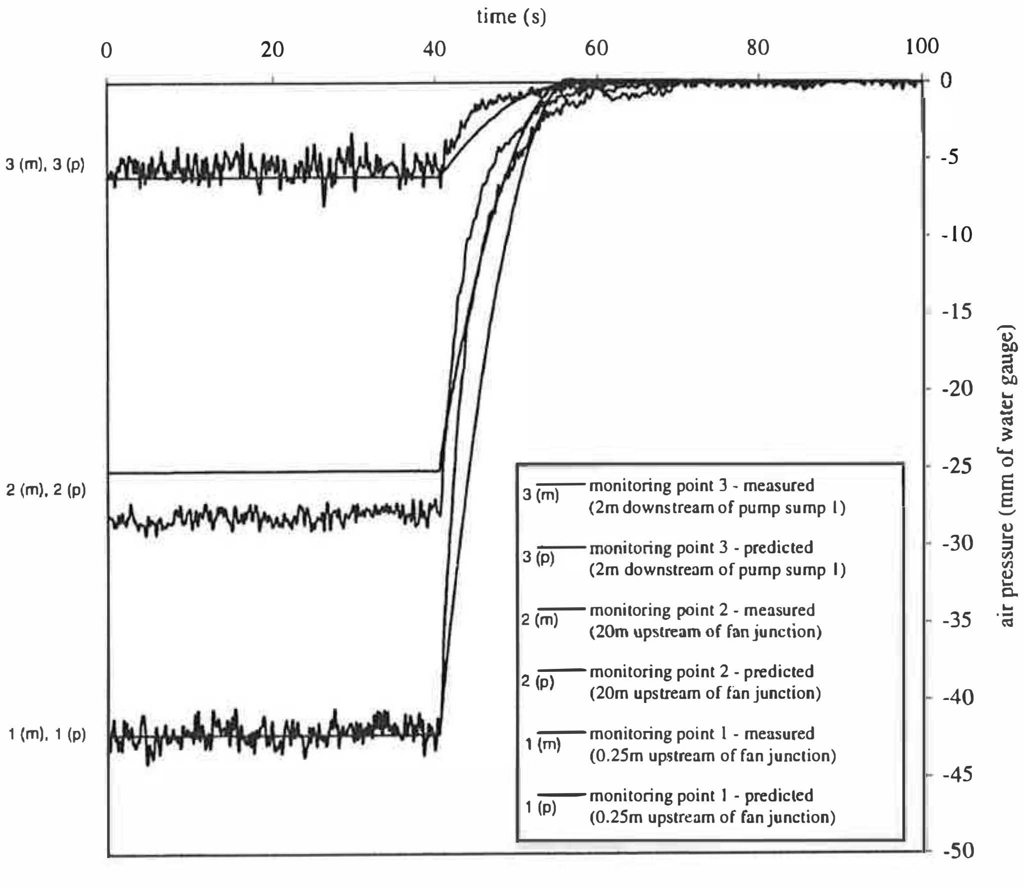

Figure 6 Measured and predicted variation in air pressure along branch 1 (linking sump 1 to fan, Figure 3) as fan decelerates from 2590 rpm to rest

Figure 7 illustrates the measured and predicted variation in air pressure, immediately downstream of pump sump number 1, as a result of the assumed instantaneous closure of the mechanical vent entry connected to pump sump number 1. The two fan speeds are utilised, 1370 and 880 rpm, corresponded to air flow velocities of 2.04 and 1.4 m s^{-1} respectively in the vent connected to sump 1. The maximum negative transient will depend upon this initial air flow velocity and the air density and acoustic velocity, equation 6, pressures of −80.7 and −55.4 mm water column respectively. As the fan is, following vent closure, effectively operating against a closed inlet, the pressure at the closed sump will fall to the fan maximum suction at that speed, measured as −14.2 and −5.8 mm water column respectively at the fan speeds tested. The minimum pressure recorded at the sump should therefore be 94.9 and 61.2 mm of water gauge at the two fan speeds tested. As shown, the model accurately predicts the steady-state air pressures occurring before and after the closure event. Figure 7 also shows that the measured and predicted peak positive and negative pressures are of the same order of magnitude, the differences being due to the inability to replicate an instantaneous vent closure on site. This led to partial, relieving reflections arriving at the sump before the closure action was complete. Referring to Figure 3, the branch 1 length of 68 m would imply a pipe period of 0.42 seconds, illustrating the difficulty of achieving an instantaneous closure, or even a closure in less than one pipe period which would generate the same peak transient. As closure was initiated by submerging the vent entry in the sump by immersing it in a container of water drawn up over the vent entry, rather than by surcharging the sump, it is likely that closure was not achieved in less than one pipe period.

5 System simulation

The site testing at Green Park underground station summarised above was successful enough to allow the developed low-amplitude air-pressure transient model to be used to simulate system operation, so that other mechanisms and design decisions could be analysed. A number of particular design considerations were investigated, including the influence of the free air volume within a holding tank, the validation of the continuity of the mass flow equation for a holding tank, and the transient pressure–time history in the dedicated vent from a holding tank with and without an air admittance valve termination. Figure 8 illustrates the assumed layout for the simulations presented, and Table 1 presents the relative lengths and diameters of each pipe in the system.

Figure 8 illustrates the air pressure fluctuations predicted at the entrance to the main vent network, pipe 6, during fan start-up and acceleration to full rotational speed for two free air volumes. As expected, the increased air volume in the holding tank acts as a damper to the incoming transients and limits the oscillations at the interface to the vent system.

Figure 9 illustrates the air flow rates predicted in the three pipes connected to the holding tank illustrated. This figure confirms that the air admittance valve opens under the influence of the fan to allow a throughflow of air into the vent system, and that the simulation maintains the continuity of mass flow.

Figure 10 illustrates the pressure oscillations predicted in the dedicated vent, pipe 7, connected to the holding tank and terminated either by an air admittance valve, which would be

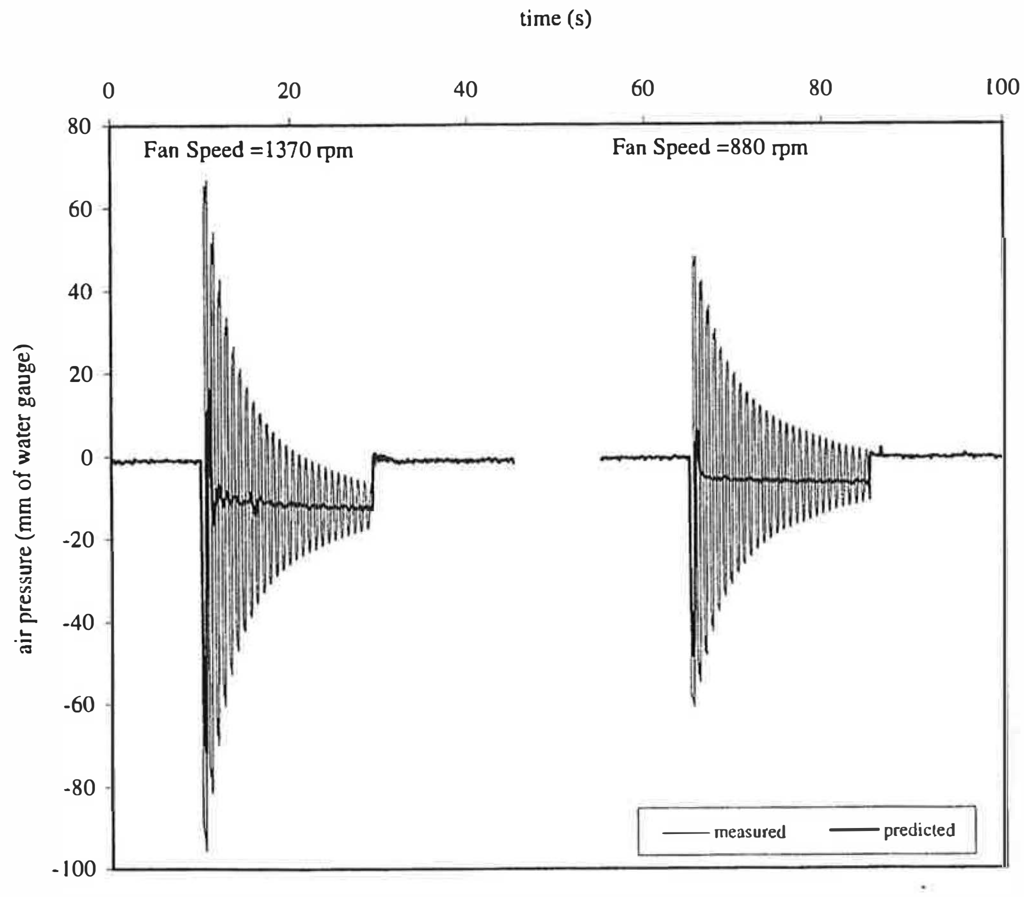

Figure 7 Measured and predicted variation in air pressure at the base of branch 1 with fan speed and the instantaneous closure and reopening of the mechanical vent entry connected to pump sump number 1 (fan speed 2590 rpm)

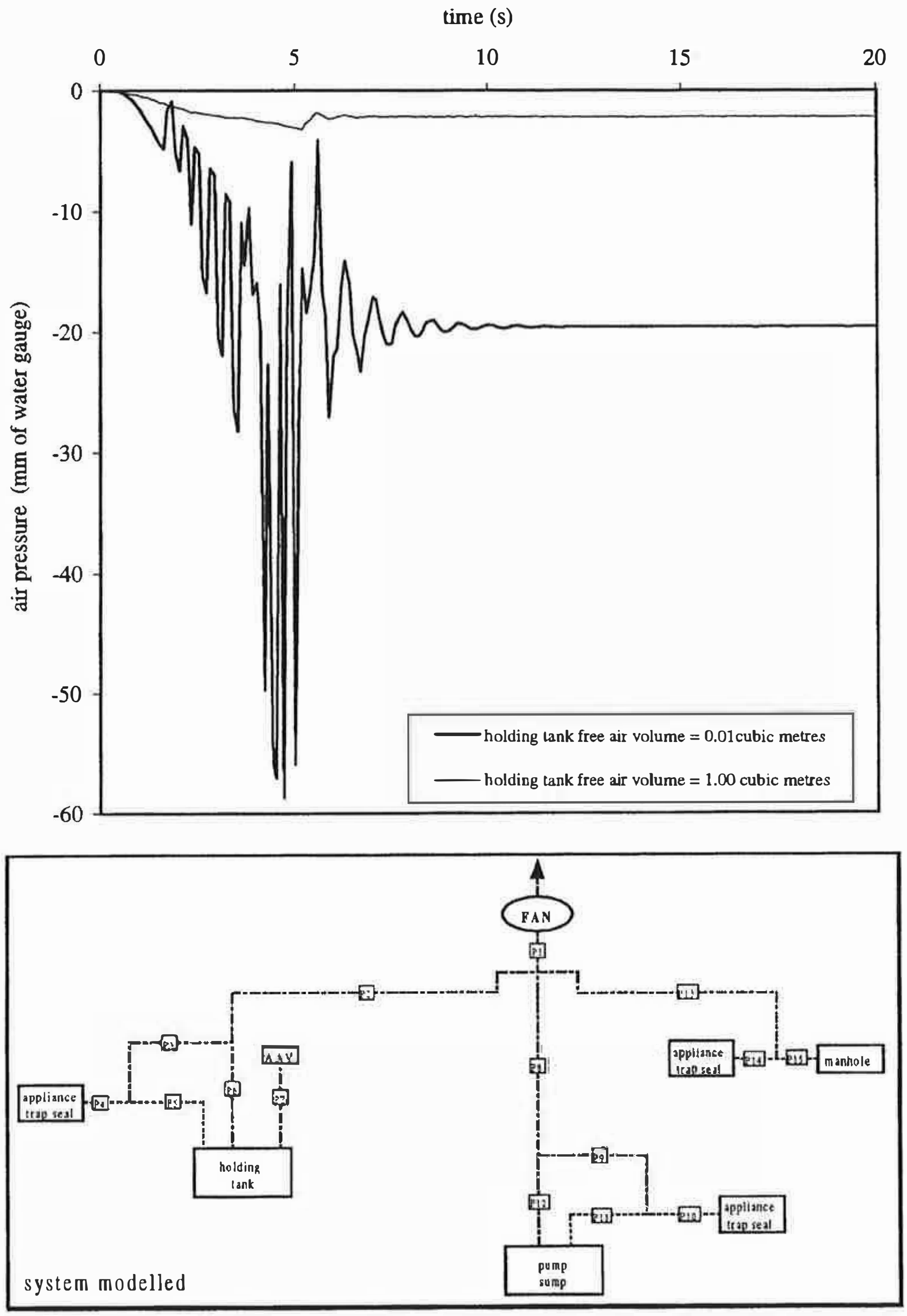

Figure 8 Predicted variation in air pressure immediately downstream of the holding tank (at entry to pipe 6) with fan start-up and tank free air volume

preferable to avoid ingress of foul air into the surrounding space, or terminated as an open end, which would be acceptable if the vent reached to ground level. The effect of fan start-up within the first 10 seconds is apparent. If an air admittance valve is fitted transients are observed in pipe 7 due to the valve's intermittent open/close motion before a steady suction pressure is established. Once the holding tank is surcharged at 20 seconds, two quite different responses may be seen, depending upon the upper termination of pipe 7. If the pipe is open to atmosphere the positive transient generated by vent closure at the holding tank interface is relieved by a series of wave reflections between a +1 reflection coefficient at the closed vent entry and a –1 reflection coefficient at the open upper termination. The oscillation dies away to atmospheric pressure. If an air admittance valve is in place then the positive transient generated by vent closure is reinforced by a positive reflection from the closed air admittance valve, and the oscillation dies away to a trapped positive pressure which can only be relieved by reopening the vent-to-holding-tank connection due to pumping waste out of the holding tank, thereby lowering the water level in the tank.

In general this simulation and similar studies inspired enough confidence in the methodology to allow the proposed simulation to be recommended as the basis for the design of below-ground drainage and vent systems. In all cases the predictions and simulations conformed to the expectations of system response based upon accepted pressure transient theory.

Table 1 Drainage and vent system data for simulated network shown in Figure 8

Pipe number	Length (m)	Diameter (mm)	Number of 45°bends	Number of 90°bends
1	10	250	0	0
2	50	100	2	1
3	20	35	1	0
4	10	100	0	0
5	20	100	1	0
6	20	100	1	0
7	20	100	0	0
8	200	100	10	5
9	20	35	1	0
10	10	100	0	0
11	20	100	0	0
12	20	100	0	0
13	120	100	6	3
14	10	100	0	0
15	10	100	0	0

A fan speed of 2590 rpm is assumed, giving a peak suction at zero flow of 500 N m^{-2} and a maximum flow of 350 l s^{-1} against a zero resistance; see Figure 6[3]. Fan manifold damper settings corresponded to loss coefficients of 123 on pipe 2, 18 on pipe 8 and 49 on pipe 13. Trap seals on pipes 10 and 14 were 50 mm, and 75 mm on pipe 4.

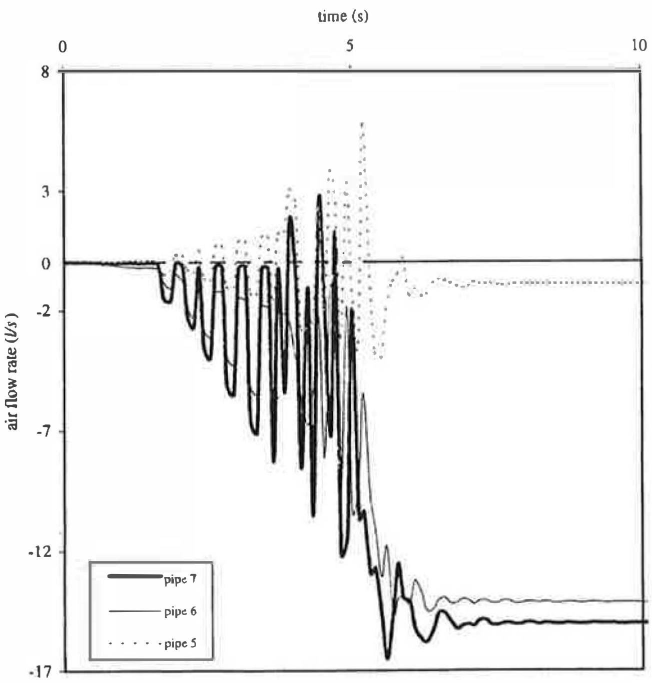

Figure 9 Predicted variation in the air flow rates through the pipes connected to holding tank, see Figure 8

6 Conclusions

Previous work at Heriot-Watt University has led to the development of computer-based numerical models capable of simulating the operation of multistorey building drainage networks and their associated vent systems. This paper has described the development of the additional boundary conditions and model subroutines required to accurately simulate the operation of the drainage vent systems installed within London Underground stations, taken as representative below-ground structures.

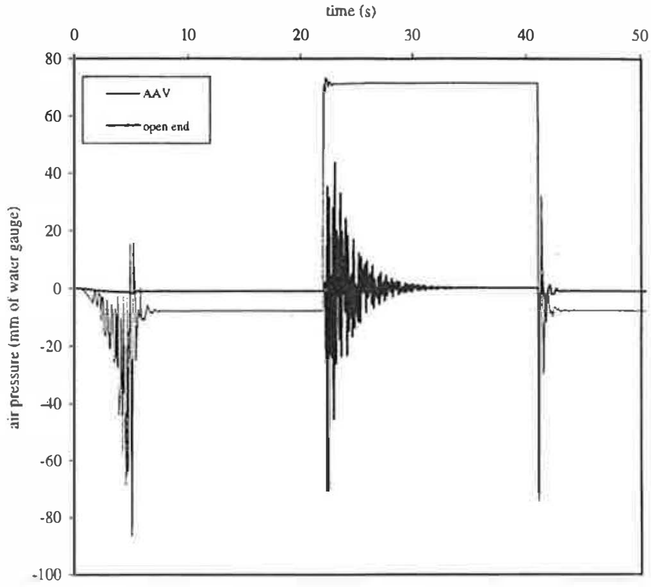

Figure 10 Predicted variation in air pressure at the mid-point of pipe 7, Figure 8, with instantaneous closure and reopening of the holding tank vent entry, with and without an air admittance valve installed

Boundary conditions have been developed to allow accurate representation of mechanical fans, which drive London Underground station vent systems, pump sumps, holding tanks and discharges to pump sumps, holding tanks and manholes. In addition the equations necessary to model the fan manifold flow balancing have been incorporated into the vent simulation to enable the effect of flow regulating valves to be simulated. The model developed, having been validated against data obtained from site tests, is now suitable for use as an aid to the design of drainage network mechanical vent systems for large underground structures. The use of the simulation to investigate system operating regimes ahead of installation as a design aid has been demonstrated.

Acknowledgements

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