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EMPTYING SEWAGE FROM THE VALVE-PIT SUMP THROUGH SERVICE LATERAL TO VACUUM MAIN

A simple hydraulic model of liquid and gas flow from a collecting sump via vacuum service lateral to a vacuum main has been presented. The model was formulated and validated on experimental data and CFD simulations. The standard gas (air) to liquid (sewage) volumetric ratio is roughly equal to the ratio of subsequent admittance times of these two phases, provided that the liquid plug is not fully sucked from the service lateral to the vacuum main during the valve open time. A longer air suction time is needed only when the service lateral is too short to provide enough air to transport the sewage past the nearest downstream lift on the vacuum main. Sizing properly the active sump volume and valve open time for a given service lateral length, one can provide the required air to liquid ratios along the vacuum main, thus minimizing the energy consumption by the vacuum pumps.

1. INTRODUCTION

The vacuum sewerage system is relatively young compared to the classical gravity sewerage system, already known in ancient times. It belongs to so-called alternative sewer systems, located inside buildings and vehicles and/or outside buildings. Its main advantages are a closed, controlled system with a central vacuum station, shallow and flexible piping, aeration of wastewater, no leakages, and no need for terrain slope in direction of the wastewater treatment plant. Apart from the main driving force – the pressure differential – a significant role still plays gravity, particularly in outdoor sewerage systems. They have also some limitations: in flat areas, the lines can only reach up to 3–4 km, operation and maintenance are rather complex, and the electrical energy, needed for sewage transport, is relatively high.

Some problems in designing and operating the system result from the lack of exact hydraulic calculation methods. The main reason is the complexity of the three-phase

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(air, water, and solids), highly unsteady, flow. The air in pipelines should be treated as a compressible fluid. When rapid transients are under consideration, it is assumed that the pressure changes occur instantaneously, allowing no time for heat transfer between the gas in the pipe and the surroundings. In this case, an adiabatic approach is adopted. Despite the rapid transients, in vacuum sewer design an assumption that the process is isothermal is common. Simplifications are possible due to relatively low air density under vacuum conditions, which can be treated as an ideal gas. Relatively low gas velocity (<30 m/s), limited pressure differences (<80 kPa), and a small fraction of solid particles (<2%) make the complex flows easier to simulate.

The driving pressure differential is created by the admittance of atmospheric air through vacuum service laterals to branches and mains. One of the pioneers of vacuum sewerage – S.A.J. Liljendahl – defined the sewage conveying principle in his famous patent description [1] as follows: *accomplishing a sudden and unobstructed introduction of a coherent vehicle plug, consisting of material to be conveyed and a small volume of liquid, into the conduit under the pressure of and, while moving through the conduit and still in a coherent vehicle plug shape, followed by a materially larger volume of air under pressure higher than the said sub-atmospheric pressure*. He postulated that for a small amount of liquid (1–2 dm^3) the initial plug length in the conduit should be at least several times the conduit inside diameter, which should range from approximately 3/4″ to 21/2″ (19.1–63.5 mm).

The amount of admitted air is crucial for reliable sewage transport and operation costs. Skillman [2] found that the energy used for sewage transport is proportional to the air-to-liquid ratios (*ALR*) required to support this process. For design purposes, water consumption in the USA is taken as 285 dm^3/cap×0183 d and 3.5 persons per house connection (AIRVAC 2018). Electrical energy consumption in typical AIRVAC vacuum stations ranged from 200 to 400 kWh/year per connection (WEF 2008), i.e., approximately 0.8–1.6 kWh per 1 m^3 of sewage. Li and Zhou [3] estimated that for indoor vacuum appliances the range is even broader, namely 0.7–2.6 kWh/m^3. Such high values observed in flat terrains mean that the energy efficiency of sewage transport is relatively low (<10%). At the same time, there must be a huge potential to diminish the energy consumption via a proper operation of the systems.

The European standard EN 16932:2018 [4] recommends higher ALR_{vs} for longer vacuum mains and lower specific population density (Table 1). However, the problem seems to be more complex. Additional factors affecting ALR_p (in pipes) are the active volume of the sump in a valve pit, interface valve open time, and length and diameter of service lateral. Relationships between the above factors are rather poorly elucidated in the literature.

Gray et al. [5] performed lab experiments on a test facility for vacuum emptying water closets and urinals. The appliances were connected with a vacuum collection tank by a double L-shaped horizontal pipeline of diameter 50.4 mm, 56 m long (Fig. 1). They investigated flowrates and pressure drops at single and intermittent (cycle periods: 5, 10, 20, and 60 s, Table 2) usage of the sanitary appliances equipped with 50 mm interface valves.

Table 1

Ranges of air to liquid ratio recommended by EN 16932 [4]

Length of vacuum sewer, L_m [m]	Length specific population density, N_{inh}/L_m, p.e./m*			
	0.05	0.1	0.2	0.5
	Mean air to water volume ratio, ALR_{vs}, at the vacuum station			
500	3.5–7	3–6	2.5–5	2–5
1000	4–8	3.5–7	3–6	2.5–5
1500	5–9	4–8	3,5–7	3–6
2000	6–10	5–9	4–8	3.5–7
3000	7–12	6–10	5–9	4–8
4000	8–15	7–12	6–10	(5–9)**

*p.e. – population equivalent equal to 150 dm^3/d and $Q_p = 0.001$ dm^3/p.e. s.
**Only recommended for exceptional cases.

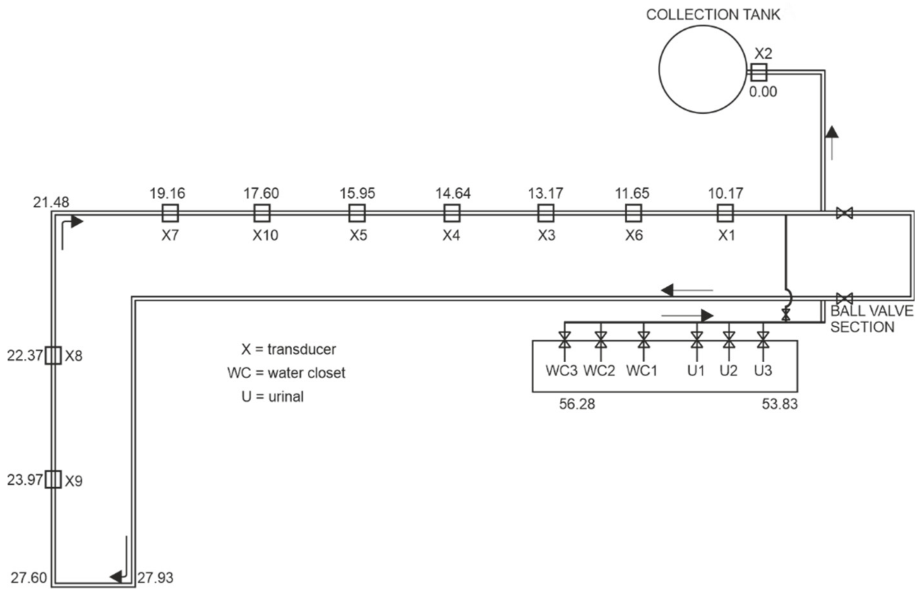

Fig. 1. Plan view of test facility showing flow directions (adapted from [5]).
Distances from the collection tank in m

The applied standard air volume to water ratio (*ALR*) ranged from 29:1 up to 235:1 dm^3/dm^3, far higher than in residential vacuum sewer practice. For calculation of frictional losses in pipelines, the AIRVAC design guidance [6] recommends applying the air to liquid ratio in pipe $ALR_p = 2:1$, possibly resulting from practical experience. The problem is that the ratio $ALR_p = ALR_{pa/p}$ depends on the absolute pressure in the vacuum main p ($30 \leq p \leq 100$ kPa) and atmospheric pressure p_a. During emptying sewage from a holding sump by its service lateral, the ratio $ALR_p \approx ALR$, as the pressure of the admitted air is slightly lower than the atmospheric one ($90 \leq p \leq 100$ kPa).

Table 2

Average cycle periods and flowrates in experiments by Gray et al. [5]

Run No.	Average valve open time per cycle, t_o [s]	Cycle period [s]	Average water fill volume per cycle, V_L, [dm^3]	ALR [Sdm^3/dm^3]
9	2.0	60	1.0	76
13	2.6	5	0.9	38
14	2.6	10	1.8	29
15	2.6	5	0.9	37
16	2.8	10	1.8	33
17	2.7	10	0.5	146
18	2.8	10	0.9	74
19	3.1	10	1.7	38
20	3.6	20	0.5	235
21	3.7	20	0.9	124
22	4.6	20	3.3	32

Large vacuum drops were found to be related to the transitory of plugs bridging the pipe bore. Profiles of time average vacuum were related to controllable parameters such as flush volume and frequency and were compared with the homogeneous and Lockhart–Martinelli head loss models. The chosen two-phase flow models tended to underpredict time average vacuum losses near the inlet valve and over-predict farther downstream. The homogeneous model has occurred conservative in most cases.

The following correlation for the volume of standard air admitted in a single flush through the urinal (U1 in Fig. 1) during its intermittent usage was proposed:

$$V_{G1} = 3.55 t_o \left(p_a - p\right)^{0.5} \tag{1}$$

where V_{G1} is the volume of air admitted per flush, Sdm^3, t_o is open time valve, s, $p_a - p$ is vacuum at nearest transducer (X9 in Fig. 1) just before the valve opens [kPa].

The coefficient of determination R^2 was equal to 0.85 (sample size $n = 11$); although the basic trend was correct, the scatter was large. Their data are shown below in our model validation chapter. At least part of the scatter was attributed to variations in the water volume (0.5–3.3 dm^3 per flush) but the authors concluded that further research and more data are needed.

This paper aimed to elaborate more comprehensive, but still simple model of vacuum service lateral hydraulics for design and operation applications. Our new simplified model of sump emptying was formulated and validated on the Gray et al. [5] experimental data and CFD simulations. Recommendations concerning limited and closer to optimal air to sewage ratios, based on the new simplified model, were presented to the designers and operators.

2. METHODS

Sewage transport in vacuum systems is possible thanks to liquid plugs driven by the pressure differential. The traveling plug problem in a circular pipe is complex due to a moving and deforming control volume. Mathematical models for liquid plugs and slugs traveling in pipelines, even those one-dimensional, are rather sophisticated [7–10]. In our simplified model, sewage (liquid) and air (gas) mass and momentum conservation laws were applied. Liquid plug velocity was calculated using the Darcy–Weisbach equation, given friction factor, plug length, pressure differential (head loss), and pipe diameter. The model was validated using accessible empirical data and simulations using CFD. It is worth mentioning that the term plug, as a coherent liquid cylindrical volume, was traditionally used in patent literature [1, 11]. On the other hand, the plugs in the theory of two-phase flow are defined as a liquid with elongated gas bubbles that move at very low velocities along the top of the pipe. At higher gas velocities and void fractions, one can observe an intermittent appearance of high-velocity liquid slugs, which bridge the whole pipe cross-section. Tay and Thorpe [12], analyzing gas-liquid slug flows in pipe bends, for the coherent liquid cylinder, called here a liquid plug, used the term piston.

2.1. SIMPLIFIED MODEL FORMULATION

Assumptions

• The vacuum valve open time is equal to the sum: $t_o = t_{oL} + t_{oG}$, i.e., the liquid is sucked first during t_{oL} and then the air – during air admittance time t_{oG}.

• At the moment of valve opening, the liquid volume in the sump is equal to V_L and it is evacuated through the vertical riser and horizontal service lateral to the vacuum main as a single, long liquid plug.

• The liquid plug of length $L_p = V_L/A$ (A is the cross-sectional area of service lateral) is traveling at a constant mean velocity, similarly as assumed by Dukler and Hubbard [7]:

$$U_L = \sqrt{\frac{2D(p_a - p)}{\rho_L f_D L_p}} \tag{2}$$

where D is the service lateral internal diameter and ρ_L is liquid (sewage) density.

• The pressure differential between the rear and front of the plug ($p_a - p$) is constant during the valve open time t_o.

• The Darcy–Weisbach friction factor f_D can be tuned (calibrated) in the field by measuring a real emptying time of liquid t_{oL} and estimated, knowing that $t_{oL} = L_p/U_L = V_L/(U_L A)$, from the rearranged equation (2)

$$f_D \approx \frac{2D(p_a - p)t_{oL}^2 A^3}{\rho_L V_L^3} = \frac{(p_a - p)t_{oL}^2 \pi^3 D^7}{32\rho_L V_L^3} \tag{3}$$

• After a rapid closing of the vacuum valve, the whole or a part (if any) of the liquid plug in the service lateral is stopped, and then it is quickly flattened and sucked into the vacuum main.

• Inertial effects and hydrostatic energy losses during liquid flow in the riser are ignored.

• Air resistance at flows with liquid plug is included in the liquid resistance.

There are two main stages (A, B) of the emptying process depending on vacuum valve open time and plug travel time relations, as follows:

• The liquid plug is fully or partly stopped in the service lateral: the travel time of the plug front t_{pf} is longer than the valve open time t_o: $t_o < t_{pf} < t_{pf}^*$, where t_{pf}^* the plug front travel time to the end of the service lateral. When the liquid plug is already partly or almost fully sucked into the vacuum main: $t_{pf}^* < t_o \le t_{pr}^* + L_p/U_L$, where t_{pr}^* is the plug rear travel time to the end of the service lateral (Fig. 2A2*).

• The liquid plug has been *alr*eady fully sucked into the vacuum main, but the airflow is continued: $t_{pf}^* + L_p/U_L = t_{pr}^* < t_o$ (Fig. 2B).

Stage A

This stage occurs when $t_{pf} + \dfrac{L_p}{U_L} = t_{oG} = t_o - t_{oL} \le t_{oG}^* = \dfrac{L_s + H_r}{U_L}$, where s is the travel time of the liquid plug front.

The volume of air pushed into the vacuum main by the liquid plug, before the vacuum valve closes, is the sum of slightly compressed air in the riser and the air under vacuum in the service lateral, which recalculated to standard conditions, gives

$$V_{G1S} = H_r A + (L - h_r) A \frac{p}{p_a} = \left(H_r + (L - h_r) \frac{p}{p_a} \right) A \tag{4}$$

where H_r is the height of the riser (Fig. 2), L is the distance from the initial liquid level in the riser to the liquid plug front, measured along the service lateral axis ($L = tU_L$); its maximum value is denoted as L^*, p is absolute pressure in the vacuum main, p_a is barometric pressure.

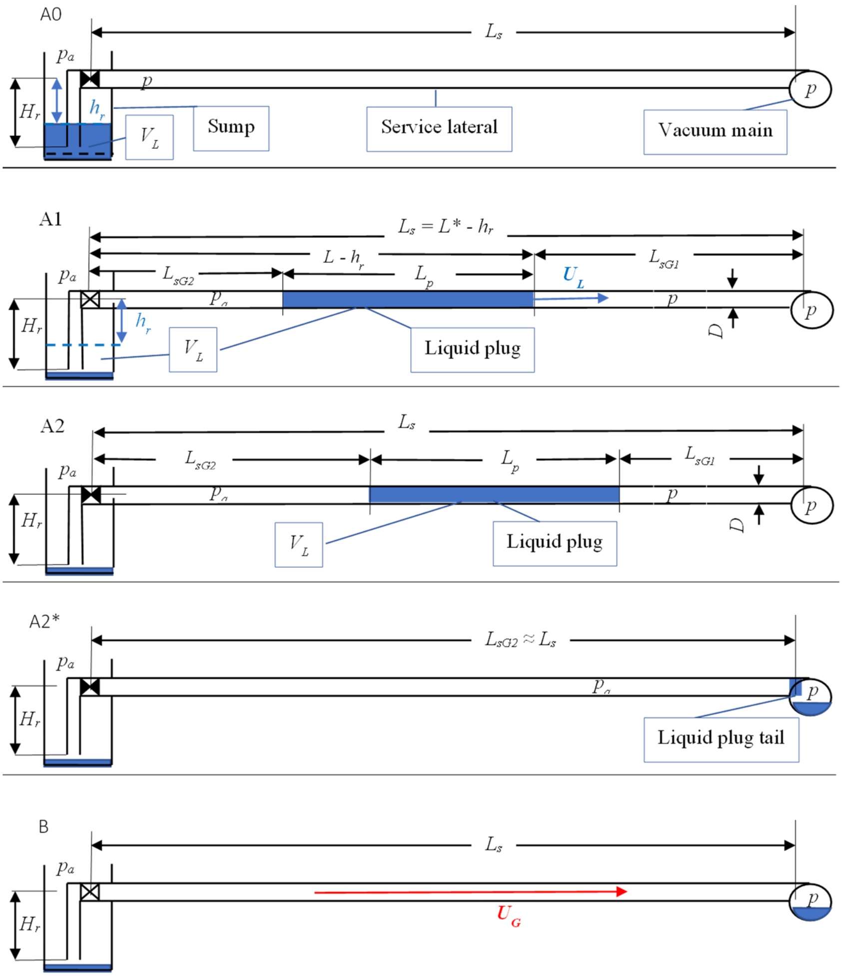

Fig. 2. Emptying stages: A0 – initial stage before valve opening ($t = 0$), A1 – liquid plug location after the vacuum valve opening $(t < t_o^* = t_{pr}^* = t_{pf}^* + L_p/U_L)$, A2 – intermediate location of the liquid plug just after the vacuum valve closing $(t \approx t_o < t_{pf}^*)$, A2* – final location of the liquid plug just after the vacuum valve closing $(t \approx t_o^*)$, B – air flow only $(t > t_o^*)$

The air mass in the lateral just after the vacuum valve closes and before liquid plug decays is the sum

$$m_{G'} = \rho_{G1} L_{sG1} A + \rho_{G2} L_{sG2} A$$

and after the liquid plug decaying

$$m_{G''} = \rho_{G1} L_S A$$

From the air mass budget, one can calculate the air mass delivered to the vacuum main, after the vacuum valve closes and the liquid plug decay

$$\Delta m_G = m_{G'} - m_{G''} = \rho_{G1} L_{sG1} A + \rho_{G2} L_{sG2} A - \rho_{G1} L_s A$$
$$= \rho_{G2} L_{sG2} A - \rho_{G1} \left(L_s - L_{sG1} \right) A$$

Thus, the volume of standard air delivered to the vacuum main after the valve closing is equal to

$$V_{G2S} = L_{sG2} A - \left(L_s - L_{sG1} \right) \frac{p}{p_a} A = \left(L - L_{sL} - h_r - \left(L - h_r \right) \frac{p}{p_a} \right) A \tag{5}$$

Using equations (4) and (5), the air to liquid ratio can be expressed by the following quotient

$$ALR = \frac{V_{G1S} + V_{G2S}}{V_L} = \frac{\left(H_r + L - L_p - h_r \right) A}{V_L} \tag{6}$$

When the liquid plug is almost fully sucked into the vacuum main ($L_p \approx 0$) by neglecting the difference $H_r - h_r$, it holds

$$ALR = \frac{L_s A}{V_L} \tag{7}$$

from which the minimum design length of the service lateral is derived as

$$L_{s\min} = \frac{V_L ALR_{\min}}{A} = \frac{4 V_L ALR_{\min}}{\pi D^2} \tag{8}$$

The recommended maximum open valve time

$$t_o^* = t_{oL} + t_{oG}^* = \frac{V_L}{U_L A} + \frac{L_s + H_r}{U_L} = \frac{1}{U_L} \left(\frac{4 V_L}{\pi D^2} + L_s + H_r \right) = \frac{L_p + L_s + H_r}{U_L} \tag{9}$$

Longer open valve times than t_o^* lead to a free air inflow (Stage B) and excessive energy consumption.

Stage B

When $t_o > t^*_{pr}$, the arrival time of the liquid plug rear t^*_{pr} is earlier than the vacuum valve closing time t_o. The air volume, pushed first into the vacuum main by the liquid plug, before the vacuum valve closes, recalculated to standard conditions, can be expressed by Eq. (4). Next, a free air volume (after liquid plug suction to the main), delivered during time interval $t_o - t^*_{pr}$, can be assessed as

$$V_{G2S} = \frac{1}{\rho_{G2}} \sqrt{\frac{D}{f_D R_G T L^*} \left(p_a^2 - p^2\right)} A \left(t_o - t^*_{pr}\right) \tag{10}$$

where t^*_{pr} is the arrival time of the liquid plug rear to the outlet, measured from the beginning of the vacuum valve opening $(t^*_{pr} = (H_r + L_s)/U_L)$.

Decompression of the air in the service lateral, from the absolute pressure p_a to p, after valve closing, delivers the following standard air volume

$$V_{G3S} = L_s A \frac{p_a - p}{p_a} \tag{11}$$

Finally, the air-to-liquid ratio can be calculated as

$$ALR = \frac{V_{G1S} + V_{G2S} + V_{G3S}}{V_L} = \frac{\left(L_s + h_r \frac{p}{p_a}\right) A + \frac{A}{\rho_{G2}} \sqrt{\frac{D}{f_D R_G T L^*} \left(p_a^2 - p^2\right)} \left(t_o - t^*_{pr}\right)}{V_L} \tag{12}$$

or

$$ALR = \frac{\left(L_s + h_r \frac{p}{p_a} + \frac{t_o - t^*_{pr}}{\rho_{G2}} \sqrt{\frac{D}{f_D R_G T L^*} \left(p_a^2 - p^2\right)}\right) A}{V_L} \tag{13}$$

where ρ_{G2} is approximately equal to the air density at barometric pressure p_a and temperature T.

Generally, stage B should be avoided, even at the expense of increasing the service lateral length L_s.

The plug model has been validated on experimental data obtained by Gray et al. [5]. The value of the Darcy–Weisbach friction factor for relative roughness $k/D = 0.001$ and $Re = U_L D/\nu > 165\ 000$ is equal to 0.02. The plug velocities were calculated using f_{D_TP}

$= f_D\phi^2 = 0.02 \times 2.75 = 0.055$, which agrees well with the above experimental data. Liquid plugs of length L_p = 0.3–1.7 m embrace intermittently one bend 90°, only. Pipe length equivalent to that single minor loss is $\zeta D/f_D = 1.13\times0.05/0.02 = 2.8$ m/m, which along whole L_s = 56 m is approximately twice smaller due to straight reaches without bends within the plug length L_p, giving for L_p = 1 m: $f_D \approx 0.02\ (1+1.4)/1 \approx 0.05$.

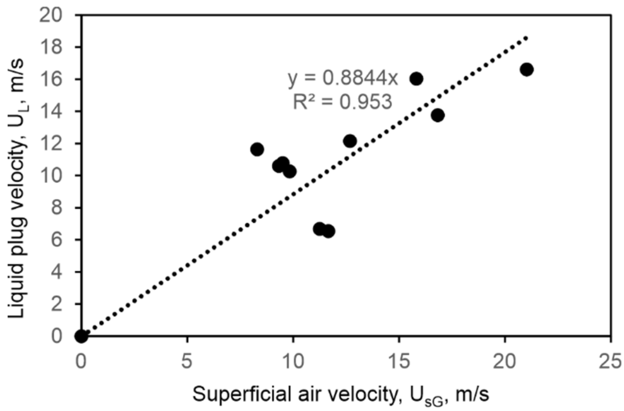

Fig. 3. Liquid plug velocity U_L calculated using Eq. (2)
in function of air superficial velocity U_{sG} measured by Gray et al. [5]

The comparison of the calculation results with the experimental data has shown that the plug model is promising in the prediction of hydraulic conditions in the service lateral during emptying small amounts of water or wastewater to the vacuum main or tank. Reasonably good agreement has been achieved in the prediction of the mean liquid plug velocity, taken as the measured air velocity (Fig. 3). Acceleration of the plug tail during its shortening (shrinking) closely to the vacuum main is limited by the locally increased pressure in the main due to the evacuated liquid. Besides, the liquid acceleration term in two-phase flows is usually not significant in the energy budget compared with the frictional one [13].

Similarly, the assessment of the admitted air volume and air to liquid ratio *ALR* seems to be also practically acceptable (Fig. 4). Despite an even larger scatter of data compared to the simplified empirical model expressed by Eq. (1), our model gives almost the same relationship with the coefficient of determination R^2 = 0.50. A better agreement has been achieved for shorter emptying times. The plug model over-estimates by 58 and 67% the admitted air volume in two experiments with short cycle periods (5 s) and by 16–20% at small portions of water (2 × 0.5 dm^3 per flush), but it underestimates (by 35%) the admitted air volume in one experiment with a relatively large portion of water (3.3 dm^3 per flush). The highest discrepancy can be attributed to the probable transport of two plugs instead of a single one, as assumed in our plug model. Unclear remains the impedance of water and air suction by the urinal elements (strainer, trap, etc.).

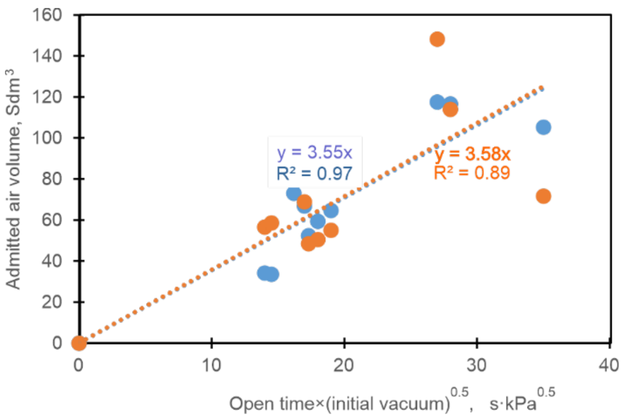

Fig. 4. Results of calculation (orange dots) and experimental data (blue dots) obtained by Gray et al. [5]

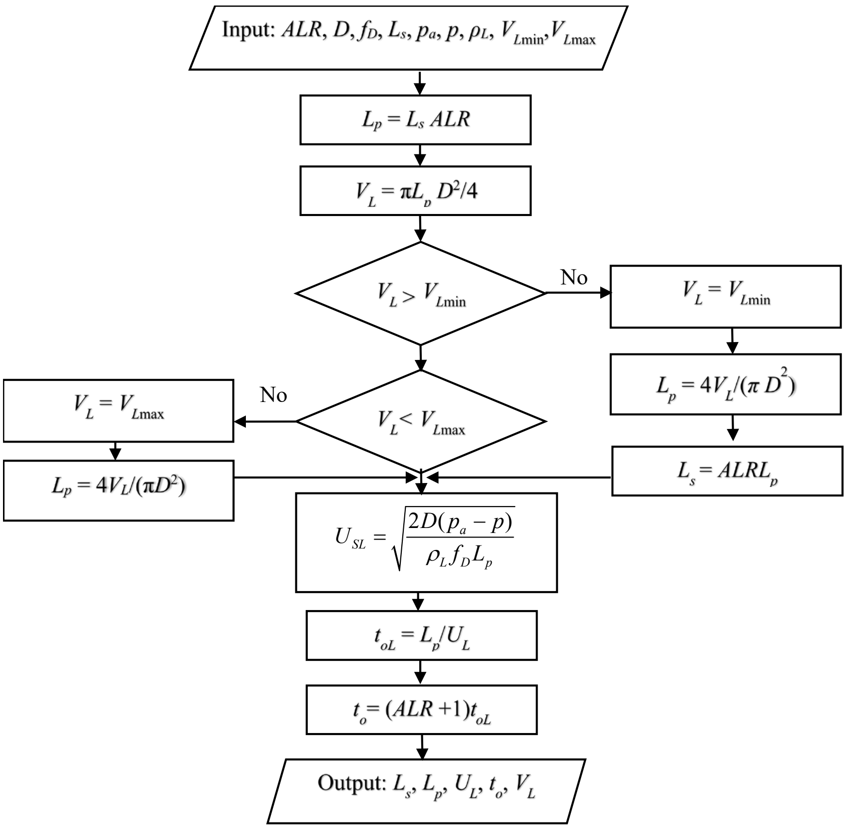

Fig. 5. Flow-chart for determining service lateral length L_s
(without a free air admittance – stage B), active sump volume V_L and valve open time t_o

Based on the above theory, a flow chart for calculating service lateral length L_s, active sump volume V_L, and valve open time t_o (Fig. 5), as the basic controllable parameters of the vacuum sewerage network.

There is also a second condition: the air of volume V_{air} and sewage V_L inserted via a service lateral during one sump emptying should rise the local pressure p_1 in the vacuum main between consecutive lifts by $\Delta p \approx 2$ kPa, up to $p_2 = p_1 + \Delta p$, for pushing a sewage plug through the downstream lift.

Treating the air as an ideal gas, from the equation of state, one obtains

$$p_1 V_p + m_{air} R_{G\,air} T = p_2 \left(V_p - V_L\right) \tag{14}$$

where $R_{G\,air}$ is the specific gas (air) constant (287 J·kg^{-1}·K^{-1}), T is the absolute temperature of the air, K, V_p is the air volume in the service laterals, and vacuum main, m^3.

Substituting $m_{air} = \rho_a V_{air}$ and rearranging

$$V_{air} = \frac{\Delta p V_p - p_2 V_L}{\rho_a R_{G\,air} T} \tag{15}$$

where ρ_a is the atmospheric air density, depending on temperature and local site altitude.

When V_{air} is lower than $V_L ALR_{min}$ then a free air admittance – stage B is necessary to pump up the local air pressure in the vacuum main by the differential pressure $\Delta p \approx 2$–3 kPa. It explains why numerous smaller lifts are recommended by AIRVAC (2018) over one large lift and why long stretches between two consecutive lifts with no service laterals should be avoided.

2.2. SIMULATION USING CFD

Ansys Fluent [14] was used for the simulation of emptying a typical sump via the simplified service lateral as shown in Fig. 2. The following values of variables and parameters were taken into account: active sump volume $V_L = 38$ dm^3 (together with liquid inside the suction pipe), initial sewage depth 0.25 m above suction pipe edge and final sewage depth: 0.045 m below suction pipe edge, the internal diameter of pipeline $D = 75$ mm, vertical pipe length $H_r = 1.25$ m, service lateral length $L_s = 10$ m, roughness height, $K_s = 0$ mm, roughness constant, $C_s = 0.5$, horizontal pipe slope = 0%, atmospheric pressure $p_a = 100$ kPa, air temperature $T = 288$ K, and absolute air pressure in vacuum main $p = 50$ kPa.

3. RESULTS

Calculations according to the flow chart in Fig. 5 were performed for typical service laterals of length $L_s = 5$–100 m and diameter $D = 75$ mm with a vertical riser of length

$H_r = 1.2$ m. It was assumed that the sump of active volume $V_L = 38$ dm^3 was emptied without any inflow to the valve pit. The interface valve open time t_o was taken 6.0 and 12.0 s. Three values of pressure differential Δp were tested: 30 kPa (minimum), 50 kPa (mean) and 70 kPa (maximum).

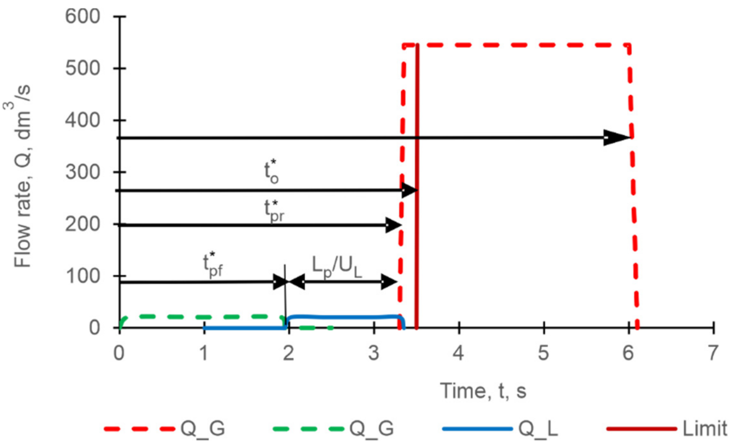

Fig. 6. Course of air and liquid flows at the inlet to the vacuum main for the interface valve; open time $t_o = 6$ s, pressure differential $\Delta p = 50$ kPa, length of horizontal service lateral $L_s = 10$ m. t_o^* denotes the recommended valve open time

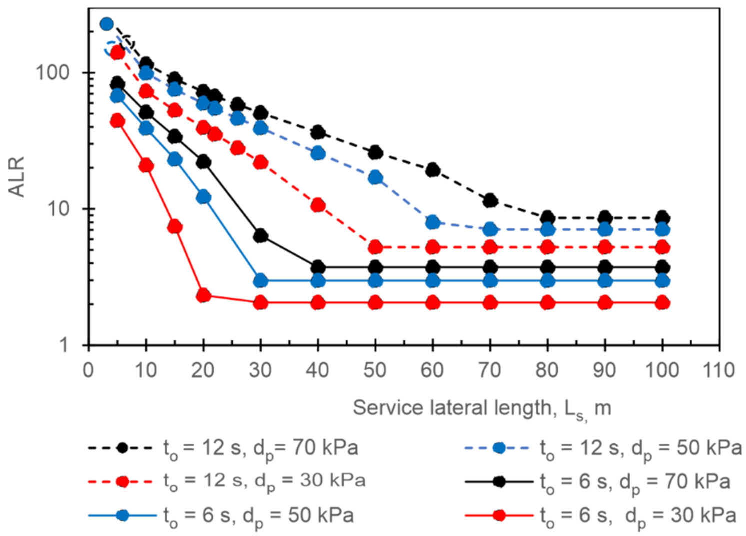

Fig. 7. Air to liquid ratio of fluids admitted to the vacuum main during emptying 38 dm^3 of liquid as a function of service lateral length L_s, valve open time t_o (6 and 12 s), pressure differential $dp = \Delta p = p_a - p$ and friction factor $f_D = 0.025$

The course of air and liquid flow at the inlet to the vacuum main for the interface valve open time t_o = 6 s, Δp = 50 kPa, L_s = 10 m is shown in Fig. 6. At the beginning, the rarified air is pushed by the liquid plug to the vacuum main. After passing the rear of the liquid plug to the vacuum main $(t > t_o^*)$, there is a peak of intensive air flow which can be responsible for a high (very often – too high) inflow of air. Such a high admittance of air results in unacceptably high values of *ALR* (Fig. 7), especially, when the valve open time is too long.

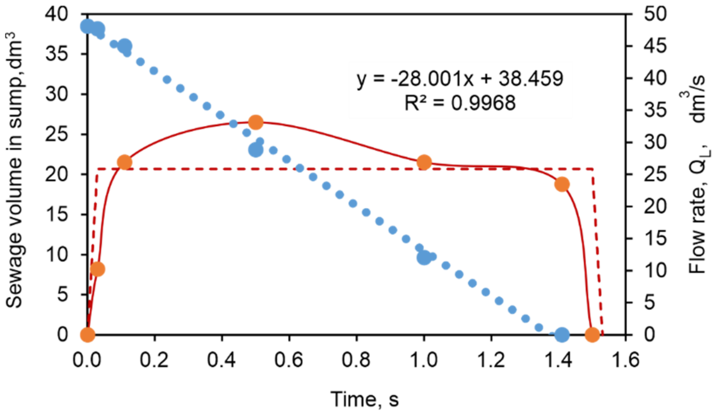

Fig. 8. Time dependences of liquid flow rate computed by CFD (solid orange line) with that calculated as a product of liquid plug velocity (f_D = 0.025, Δp = 50 kPa) and the pipe cross-section area (orange dashed line) and temporal course of sewage volume in sump above the suction (inlet) edge (dotted blue line)

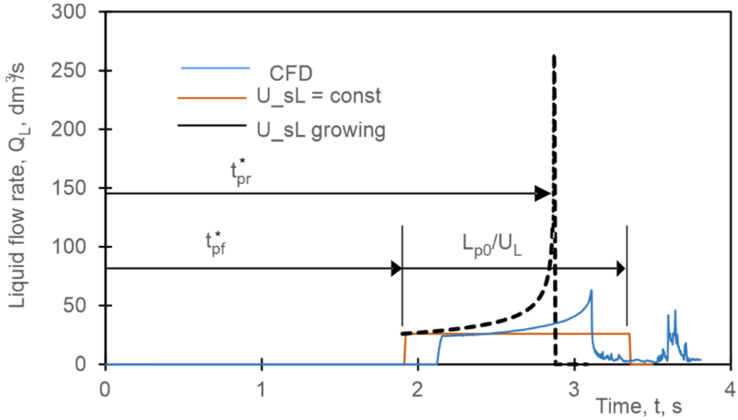

Fig. 9. Time dependences of liquid flow rates at the inlet to the vacuum main, calculated by three different methods, for the interface valve open time t_o = 6 s, pressure differential Δp = 50 kPa, and length of horizontal service lateral L_s = 10 m

Comparison of liquid flow rates, during emptying 38 dm^3 of liquid at Δp = 50 kPa, computed by CFD (solid orange line in Fig. 8) with that calculated – as a product of liquid plug velocity (for $f_D = 0.025$) and the pipe cross-section area (orange dashed line) shows a reasonable agreement for a cross-section located directly downstream from the elbow.

A similar comparison of liquid flow rates at the inlet to the vacuum main, calculated by the applied three different methods, is shown in Fig. 9. It can be seen that the plug front travel time to the end of the service lateral t^*_{pf} is close to 2.0 s and the CFD estimate it is about 10% longer than that calculated by the simplified methods. That discrepancy can be explained by neglecting the inertial effects in the latter case.

During the final phase of the stage A1 the liquid plug length is reduced at a rate

$$-\frac{dL_p}{dt} = U_L(t) = CL_p^{-0.5} \tag{16}$$

where $C = (2D\Delta p/\rho L f_{\mathrm{D}})^{1/2}$.

After integration with respect to the liquid plug length L_{pL}, which is reducing in the time interval $[t^*_{pf}, t^*_{pr}]$ from L_{pL0} to 0, one obtains

$$t^*_{pr} - t^*_{pf} = \frac{L_{pL0}^{1.5}}{1.5C} < \frac{L_{p0}}{U_L} \tag{17}$$

The inequality (17) implies that the assumption in our simplified method is on the safe side, i.e., the recommended valve open time is longer than in reality.

4. DISCUSSION

Typically, a service lateral (connection) length is dependent on valve-pit and main or branch line location. No recommendations have been published relating the service lateral length to hydraulic parameters, e.g., sump volume or pipeline slope. In Australian guidelines [15], there is a recommendation that a service lateral with a lift should be ca. minimum of 4 m and a maximum of 30 m long. The maximum length of a service lateral of internal diameter 75 mm (3″) according to AIRVAC [6] is 91.5 m (300 feet). Roediger's construction guidelines [16] recommend laying service laterals with a slope of a minimum of 0.2% in the direction of flow. Special solutions (not specified) are necessary for house connection lines longer than 20 m.

The active sump volume for older 50 mm (2″) AIRVAC valves was 19 dm^3 (5 gal), whereas for recently preferred 75 mm valves 38 dm^3 (10 gal.) [6]. Sewage level in the sump in the range $H_r - h_r$ = 25–35 cm is typically chosen. Regarding the valve (and so the capacity to evacuate the sewage), there are two types of RoeVac G-type collection chamber: G65 – 2.5″ with a small sump of volume 30 dm^3 and G75 3″ with a big sump of volume 60 dm^3 [16]. The following hydraulic loads shall not be exceeded: for G65 V_L = 20–30 dm^3 sewage at a rate of 1–3 dm^3/s per evacuation cycle (maximum 30 s) and for G75 V_L from 30 to 50 dm^3 sewage at a rate of 1–3 dm^3/s per evacuation cycle (maximum 30 s). Sewage level differential in the sump ranges from 20 to 40 cm. Lab experiments on the flow capacity of the RoeVac G65 interface valve made by Kalenik [17] showed that with increasing open times (from 6 to 12 s), the liquid flow Q_L through the vacuum lateral had increased only slightly (from 18 to 23 dm^3/s at Δp = 70 kPa), however, the airflow Q_G had increased significantly (from 64 to 243 dm^3/s, respectively). The ratios *ALR* were also significantly higher (3.6 vs. 10.6) for longer open times. Water evacuation rates (2–3 dm^3/s), higher for shorter valve open times, were coherent with the above-mentioned producer's estimates.

It can be noticed that in the AIRVAC's practice, the greater the active sump volume, the larger the internal diameter of the valve and the vacuum service lateral is, in proportion: $V_{L1}/V_{L2} = 38/19 = 2 \approx (D_1/D_2)^2 = (75/50)^2 = 2.25$ to achieve nearly the same liquid plug length L_p. The created plugs are relatively long ($L_p > 100D$) and therefore they are more stable than shorter plugs. On the other hand, they should not be too long, because then the condition of $ALR > 2$ would not be fulfilled. Such a situation may happen when a large sewage volume, e.g., from the bath, is delivered to the sump during its emptying at a slow rate. To avoid waterlogging, some operators have drilled a small hole (5–6 mm) in the suction pipe in the valve-pit sump to deliver the air simultaneously with the sewage. That practice is questionable because the two-phase flow is homogeneous in the riser only, but then it transforms to churn flow and separates quickly in nearly horizontal service laterals. That mode of sump emptying needs further research.

Valve open time should be constant, consistent with the setting by the operator. Unfortunately, some pneumatically driven controllers are not stable, and once set the valve open time may change depending on the pressure differential, temperature, vibrations, etc. Figure 10 depicts variability in the open time of four randomly chosen RoeVac membrane valves of diameter 2.5″ in a small vacuum sewerage system in western Poland. The valves No. 1003 and 1013 were located closely to the vacuum station, whereas the valves No. 1122 and 1124 – 1.7 km upstream the station. The open time of the RoeVac valve unit is factory set to 5 s. It is usually corrected in the field by trial and error.

The length of one complete AIRVAC (3") valve cycle is about 6–8 s, consisting of 2–3 s for the liquid suction, followed by 4–5 s of air admittance (AIRVAC 2018). However, in a field study performed in the years 1976–1977 in Bend (Oregon, USA) with participation of AIRVAC personnel, the valves most distant from the collection station were set with considerably longer open times than recommended by AIRVAC design literature. The measured liquid evacuation time t_{oL} for those pits with relatively high active sump volumes (42–64 dm^3) ranged from 0.7 to 1.3 s, whereas the valve open times – from 6.0 to 11.7 s [18]. Air pumps on and off limits were set at 47 and 67 kPa of vacuum. The whole vacuum network was relatively short (563 m including service laterals, made of PVC pipes of diameter 75 mm) and served 8 homesteads, only. The liquid flow rates for the six most distant valve pits ranged from 46 to 89 dm^3/s and the corresponding liquid plug velocity from 10.7 to 19.4 m/s, which are close to those measured by Gray et al. [5] and estimated by Eq. (2) (Fig. 3).

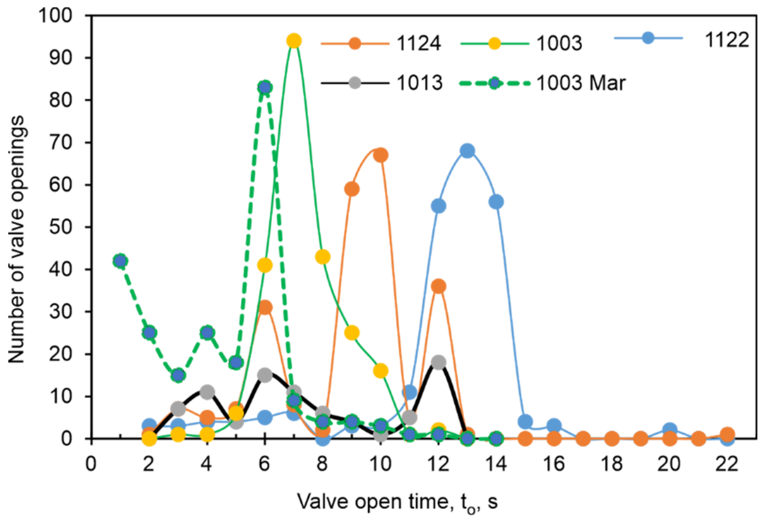

Fig. 10. Variations of open time t_o of valves No. 1013, 1123 and 1124 in April 2021 and valve No. 1003 in March and April 2021

During further transport of sewage in the vacuum main, every lift generates head losses, hence the longer distance from the vacuum station the greater energy is used from the expanding air and a higher *ALR* is required. The recommended valve open times shown in Fig. 11 are corresponding to the Stage A2*, when the liquid plug is almost fully sucked into the vacuum main (Fig. 2A2*) and almost the whole service lateral is under atmospheric pressure: at that very moment $(t = t_o^*)$ the vacuum valve is closed. When the service lateral volume $((L_s + h_s)A)$ had been too small to provide $ALR \geq 2$, a short interval of Stage B was admitted. The vacuum valves' open times for $ALR \geq 4$, i.e., for longer main length and distances from the vacuum station, should be longer than those depicted in Fig. 11.

Typically, the vacuum pumps are set to operate under a vacuum of 54–68 kPa, i.e., the absolute pressure p_{vs} in the collection tank varies in the range of 33–47 kPa. The corresponding standard air to liquid volume ratios *ALR* in the collection tank ranges therefore from 2.0 to 5.2 (Table 3) and the same range must be provided during sumps emptying through service laterals. The *ALR* recommended by AIRVAC are close to those given by EN 16932 (Table 1) for relatively high population density, only. In sparsely populated areas the required *ALR* might be even twice higher.

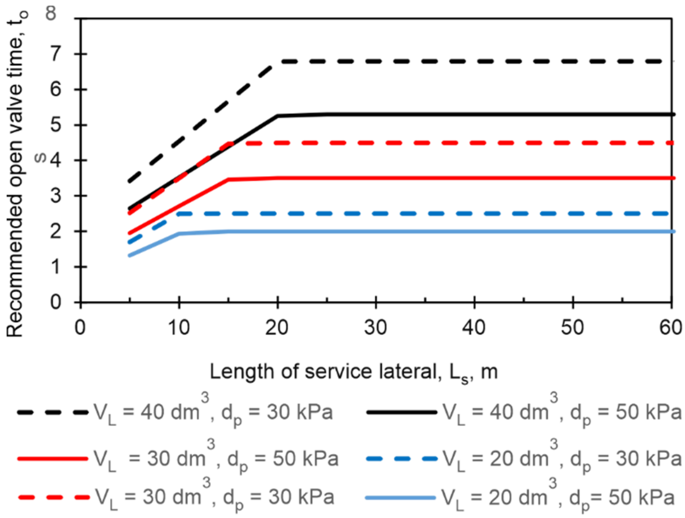

Fig. 11. Recommended valve open times t_o for a given sump volume V_L, service lateral length L_s, pipe diameter D = 75 mm, friction factor f_D = 0.025 and air to liquid ratio $ALR \geq 2$

Table 3

Air pumping speed to liquid peak flow ratio A_Q recommended by AIRVAC [6] and corresponding standard air to liquid volume ratios ALR

Longest line length [m]	$A_Q = \frac{Q_{vp}}{Q_{max}}$	$ALR = \frac{A_Q p_{vs}}{p_a}$
0–1524	6	2.0–2.8
1524–2134	7	2.3–3.3
2134–3048	8	2.6–3.8
3048–3658	9	3.0–4.2
>3658	11	3.6–5.2

The proposed simplified method can be treated as a rule of thumb for the design and operation of a vacuum sewerage network, both indoor and outdoor. Particularly questionable is the assumption of a constant pressure gradient acting on the liquid plug. The inflow of liquid from the service lateral to the vacuum main must increase the absolute pressure in the vacuum main, therefore the decreasing pressure differential is acting on the shrinking liquid plug, thus decreasing a potential error in the estimation of the optimal vacuum valve open time. As yet CFD 3D simulations, even limited to two-phase flow, last too long and need enormous memory resources to be practically useful as a design tool. The development of CFD in the future should be helpful in the simulation of whole sewerage networks, not only of their small parts.

Further field studies are needed firstly on small systems, where corrections of hydraulic operational parameters are relatively easy to introduce, and their results can be monitored and interpreted in detail. The actual values of *ALR* should be determined for each connection and compared with the recommended value $ALR = t_o/t_{oL} - 1$ for $t_o = t_o^*$. The mean *ALR* must be equal to its counterpart in the vacuum station. Apart from valve open time and pressure inside vacuum mains, it would be of interest to monitor volumes of evacuated sewage via continuous measurements of sewage levels in sumps. Modern wireless valve pit monitoring systems allow monitoring valve opening and closing operations as well as liquid levels in the holding sump.

In sewerage practice, the flow is three-phase, therefore density and viscosity of the mixture are typically higher and the frictional loss is greater than in the two-phase flow. The mixture must be treated then as a non-Newtonian fluid [19]. It is especially important in the case of black water (sewage from toilets and urinals) collection, where the air to liquid ratio is relatively high.

Up-to-date monitoring systems can make automatically real-time adjustments to prevent problems from occurring (e.g., too low vacuum levels at the end of the main), and reduce operating costs. However, to optimize any sewerage system an adequate hydraulic simulation model seems to be indispensable.

5. CONCLUSIONS

- Active sump volume, service laterals geometry (diameter, length, roughness), vacuum level, and valve open time have the most significant effect on the air to sewage ratio and further on, on hydraulic conditions of the sewage transport through the branches and mains to the vacuum station.
- The standard air to liquid volumetric ratio is roughly equal to the ratio of admittance times of these two phases, provided that the liquid plug is not fully sucked from the service lateral to the vacuum main during the valve open time. Otherwise, the former ratio can be significantly greater than the latter, thus increasing energy consumption.
- Sizing properly the active sump volume and valve open time for a given service lateral length, one can provide the required air to liquid ratios along the vacuum main, thus minimizing the energy consumption by the vacuum pumps.
- Based on the air-to-liquid ratio in the pipe equal to 2:1, recommended by AIRVAC, the minimum service lateral length for average vacuum in the main equal to 50 kPa, is a quotient of the active sump volume and the pipe cross-sectional area.
- Further research is needed to elucidate the role of service laterals in shaping optimal conditions (minimum air to liquid ratios) of sewage transport, especially in the field and/or on a technical scale in small pilot systems.

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