The effects of centrifugal blowers, control valves, attenuating devices and reservoir resonance on organ pipe flutter

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Abstract

The aim of this research is to investigate the noticeable organ pipe flutter that may, under certain conditions, exist on a sounding organ pipe. The effectiveness of a pipe organ wind system is notoriously difficult to predict. For many years pipe organ builders have been aware of organ pipe flutter and several have tried to address the problem with little success. For pureness of tone it is important that the wind system is perfectly steady and without any imperfections that may cause organ pipe flutter. A survey of 83 UK pipe organs, was conducted by 8 organ tuners, confirms that 57% of the pipe organs surveyed had organ pipe flutter. Organ pipe flutter is particularly noticeable when tuning pipework or playing single notes. During this condition there is “no flow” in the duct connecting the blower to the reservoir. Using a specially constructed test apparatus, built from pipe organ components, this research examines the conditions necessary to produce organ pipe flutter, and how organ pipe flutter may be eliminated. Employing a microphone to measure a sounding test organ pipe and an accelerometer to measure the vibration of a reservoir top, various pipe organ wind system elements are examined and correlated with the excitation and attenuation of the reservoir top vibration and organ pipe flutter. The reservoir acts as a mass spring system. For weighted wind systems the mechanical mass, which may exceed 100kgs, is the dominant factor. For sprung reservoir wind systems, the mass is approximately 25% of that for a weighted system and is less dominant. Results indicate that under certain conditions, the blower excites the reservoir at its natural resonant frequency with sufficient amplitude to cause unwanted amplitude modulation on a sounding organ pipe. Results are systematically presented for weighted and sprung reservoir wind systems, organ blowers and the effects of blade frequencies, reservoir control valves, and attenuating devices inserted between the blower and the reservoir, to determine their effect on reservoir top vibration and the development of organ pipe flutter. With this knowledge, the pipe organ builder will be able to build pipe organs with improved wind systems and flutter free pipework.

1 Introduction

1.1 General introduction

For many years Pipe Organ Builders have been aware of imperfections that exist in pipe organ wind systems. The purpose of this research is to examine in detail organ pipe flutter and determine how it is created and how it may be mitigated. A knowledge of the workings of a pipe organ is necessary to understand the phenomena of organ pipe flutter.

There are three basic parts to most wind instruments: the control, the resonator and most important for this class of instrument, a supply of air. Most musical wind instruments are designed to play single notes and use a single resonator. The control is provided by a series of holes or valves to select different notes. Air is supplied by the player who develops a technique of winding, that best suits each instrument. There are several exceptions where the instrument is designed to play more than one note at a time and the air supply needed to play the instrument is beyond the means of a single person. In these instruments, some form of mechanical system is employed and normally takes the form of a simple bellows and feeder arrangement.

This simple arrangement of pressurised air, held in an enclosure with some form of mechanism to admit air into the pipes under the control of a player, is the basis of all pipe organs. From early times, the pipe organ has continued to develop to reach the form that we find today. Much of the development has centred on the action that controls the pipes. It is now possible to have many thousands of pipes under the control of a single player. The largest pipe organ in the world contains over 33,000 pipes and is controlled by seven keyboards and a pedalboard Barnes [1]. It is not surprising that the pipe organ is often referred to as the “King of Instruments”. For comparison, the largest pipe organ in the United Kingdom is in Liverpool Anglican Cathedral and has 10,268 pipes controlled by five keyboards and a pedalboard. Fortunately, the vast number of organs are of a more modest size and contain two or three keyboards with substantially fewer pipes. A typical two manual pipe organ contains approximately 1,000 pipes. Each pipe produces a single note and the speaking length varies from 10mm to 10m. The case or display pipes conceal most of the pipework and represent only a small proportion of the total number of organ pipes. It is possible to play many pipes simultaneously, so it is important that all the pipes are in tune. The pipes are tuned in octaves until no beats are audible. This requires each organ pipe to “speak” without any imperfections so that the tuning beats can be clearly heard. Any imperfections in the wind system that may cause one or both organ pipes to speak with a slight flutter makes fine tuning of the organ very difficult.

Vital for the organ to function is a supply of air which is generated by a simple feeder arrangement or an electrically driven fan supplying a reservoir. The reservoir is pressurised using weights or springs applied to the reservoir top.

This research examines the presence of organ pipe flutter in 83 pipe organs using information collected by 8 organ tuners. Also, using a specially constructed test apparatus, that represents a working pipe organ wind system, the cause of organ pipe flutter will be determined, together with methods for its control and elimination.

1.2 Thesis outline

Following this introductory Chapter, Chapter 2 covers briefly the evolution and development of pipe organ wind systems.

Chapter 3 focuses on the extent of organ pipe flutter and the results of a Pipe Organ Survey conducted by 8 organ tuners on 83 pipe organs.

Chapter 4 describes in detail the methodology and test apparatus necessary to determine the amount of organ pipe flutter on a test organ pipe. Some preliminary results and observations are presented.

Chapter 5 defines frequency and amplitude modulation together with the psychoacoustics of organ pipe flutter.

Chapter 6 examines blower and reservoir combinations.

In Chapter 7, the theory of reservoir resonance is examined.

Chapter 8 investigates how different blowers produce different levels of organ pipe flutter.

Chapter 9 investigates the development of blade tone and its effect on organ pipe flutter.

In Chapter 10, the effects of impeller inter-blades are examined with reference to organ pipe flutter.

Chapter 11 examines the effects that different control valves have on reservoir top vibration and organ pipe flutter.

In Chapter 12, having considered the cause of organ pipe flutter, possible methods and devices used by organ builders that limit or eliminate pipe organ flutter are examined.

In Chapter 13 the salient points raised by the research are discussed and the areas for future research outlined.

The Appendix A, B, C, D, E & F contain additional supporting material.

1.3 Novelty of this research

The following outcomes, which are presented throughout the thesis, are novel contributions to the field of pipe organ building to reduce reservoir top vibration and organ pipe flutter.

- A logical examination of a pipe organ wind system with reference to its historic development and construction.
- Demonstrating that reservoir top vibration causes organ pipe flutter.
- Working in the frequency domain, the various elements of a pipe organ wind system are systematically examined and correlated with the excitation and attenuation of reservoir top vibration and organ pipe flutter.

The results of this research will extend the knowledge base and allow pipe organ builders to better understand the generation and attenuation of organ pipe flutter and improve the design and construction of pipe organ wind systems.

2 The evolution and development of pipe organ wind systems

In this Chapter the significance of the various components used in a pipe organ wind system and this research are examined.

2.1 Early Greek pipe organ

As early as 250 B.C. Ctesibius, a Greek engineer, constructed an hydraulic or water pipe organ which is described in detail by Barnes [1]. Figure 1 shows the workings of the water organ. Air pressure is developed from a column of water to drive air through a set of tuned pipes. The operation of the two handles K & L moves the two pistons M & N in the cylinders F & G and pumps air into the chamber C. The air is pressurised by the water in the tank B and prevented from escaping back into the atmosphere by the two non-return valves Q & R. The pressurised air is held in the chamber E until the lever T moves a slide, allowing the air to pass into the chamber S. The player presses the spring loaded key V to allow air into the pipes mounted on the board U.

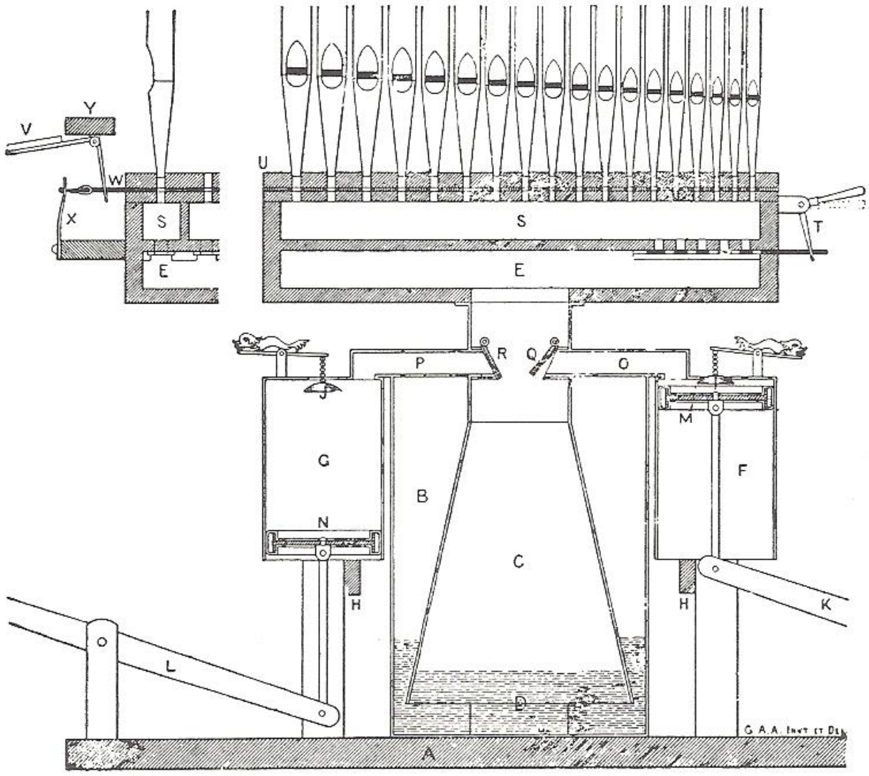

Figure 1 The Hydraulic organ of Ctesibius Circa 250 BC (After Barnes [1])

2.2 Wind systems of the 16 Century

All organs use some form of reservoir to pressurise the wind supply Elvin [2] Audsley [3]. A simple arrangement of feeders and bellows from a sixteenth century organ in Halberstadt Cathedral is shown in Figure 2.

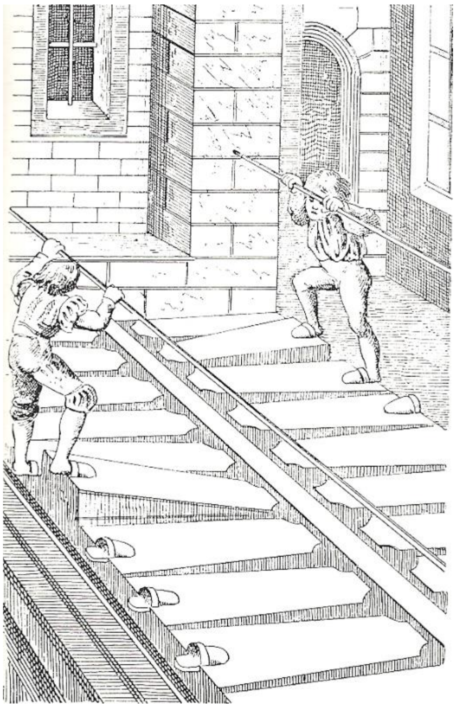

Figure 2 Feeders in Halberstadt Cathedral Circa 1500 (After Elvin [2])

2.3 Hand blown double rise weighted reservoir

For smaller organs the arrangement of the reservoir and feeders is shown in Figure 3. The air pressure inside the reservoir is pressurised by placing weights on the top of the reservoir.

The most important feature of any pipe organ wind system is that the air pressure in the reservoir must remain constant, otherwise the pitch and tuning of the pipes will vary. This is achieved by having two sets of ribs, one folding outwards the other folding inwards. The outward folding ribs pull the reservoir top down and the inward folding ribs push the reservoir top up, so that together, they compensate each other. This allows the top of the reservoir to move up and down without changing the outlet pressure of the reservoir. Two or more feeders are connected to some form of handle that can be operated by one person, also shown in Figure 3.

In this application no control valve is needed, and the maximum height of the reservoir is restricted by the flap safety valve. A thin cord connects the top of the reservoir over a series of pulleys to a small brass weight in clear view of the person operating the blowing handle. The brass weight acts as an indicator to show the fullness of the reservoir and it is important that the blowing person keeps the indicator between two predetermined positions

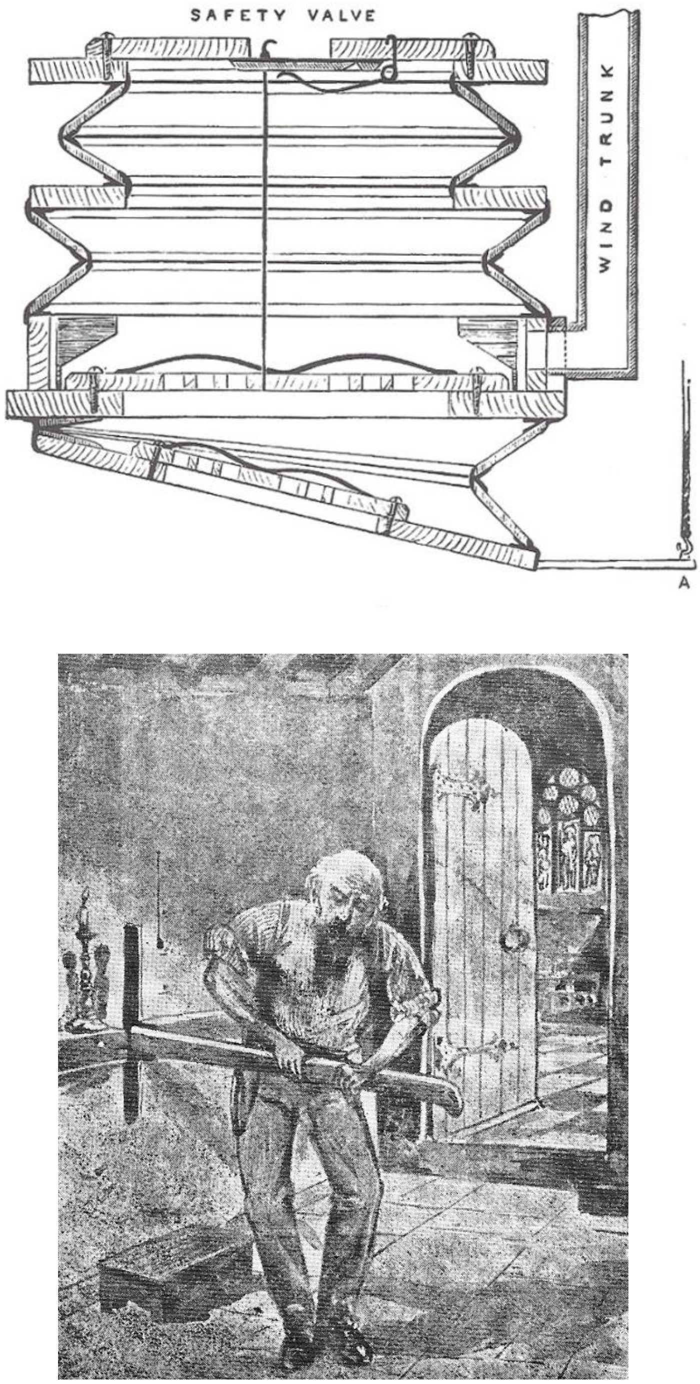

Figure 3 A double rise reservoir with feeder and a typical hand blowing arrangement (After Barnes [1] & Elvin [2])

2.4 Water engine operated feeders

Technology very rarely stands still. With the advent of the Industrial Revolution, the power of water, steam and gas replaced muscle power as a prime mover. The use of a non-human power source allows greater flexibility and the construction of larger organs. However, it is still possible for new organs to be made with a traditional hand blowing wind system. A typical arrangement using a water engine to operate the feeder is shown in Figure 4. The flow of water into the valve is controlled by the vertical rod connected to the top of the reservoir. As the reservoir fills, the vertical stroke of the water engine slows.

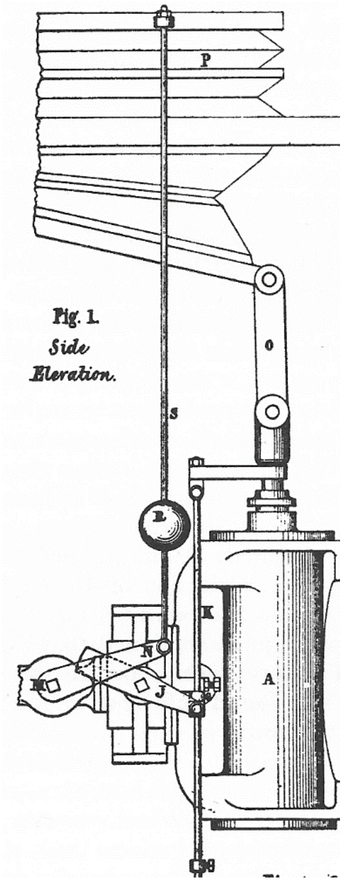

Figure 4 A set of bellows and feeder powered a water engine. (After Elvin [2])

2.5 Gas engine driven feeders

Not all areas had a copious supply of water or distribution system. The invention of the gas engine and its development in France during 1861 provided an alternative form of propulsion for areas poorly served by municipal water companies. In 1878 Crossley Brothers of Manchester, introduced a new Otto silent gas engine and offered sizes of 1, 2, 31/2 and 8 horse power models. Figure 5 shows a typical Crossley Brothers Limited gas engine connected to a pipe organ wind system. In1884 a 31/2 H.P. engine was supplied to St Giles Cathedral, Edinburgh. Also at this time, several other major organs had gas engines fitted.

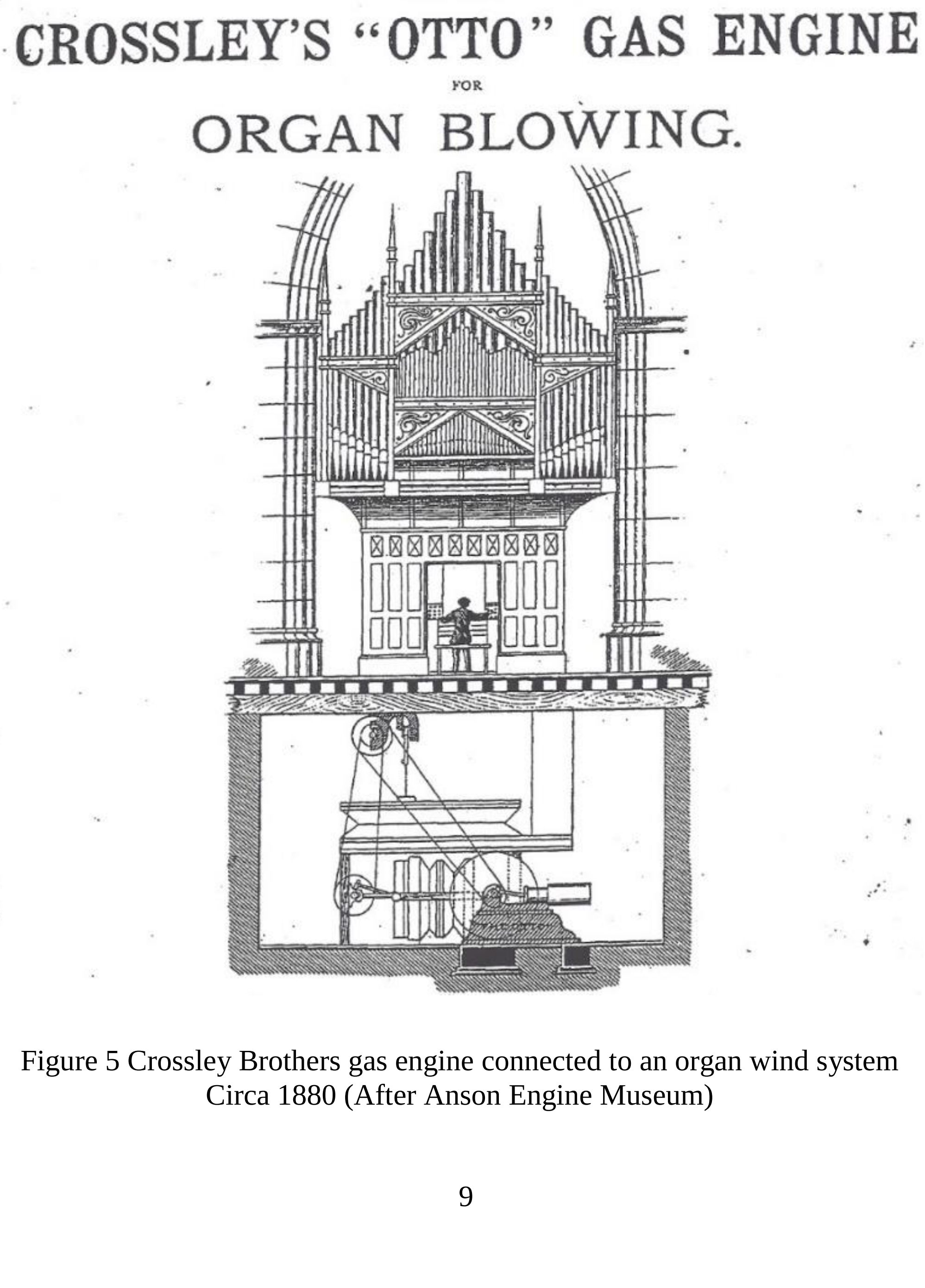

Figure 5 Crossley Brothers gas engine connected to an organ wind system
Circa 1880 (After Anson Engine Museum)

2.6 Electrically driven feeders

The development of municipal electricity supply companies, during the last decade of the nineteenth century, opened the flood gates to a plethora of electrically powered devices. The electric motor was one such device which provided a clean and more reliable form of propulsion than the water or gas engines. Figure 6 shows a double rise weighted reservoir supplied by three feeders, connected to a 3-crank shaft driven by an electric motor. The move from the reciprocating motion of the water and gas engines, to the rotational motion of the electric motor, proved a convenient power source to drive the crank mechanism. The speed of the motor was controlled by a simple feedback system that connected the top of the reservoir to a rheostat, that controlled the speed of the motor.

Older organ builders and tuners considered the most stable wind was supplied by a large double rise reservoir supplied by a 3-crank feeder system similar to that shown in Figure 6. Such an arrangement is still giving good service supplying wind to the organ at Colne Parish Church in Lancashire England.

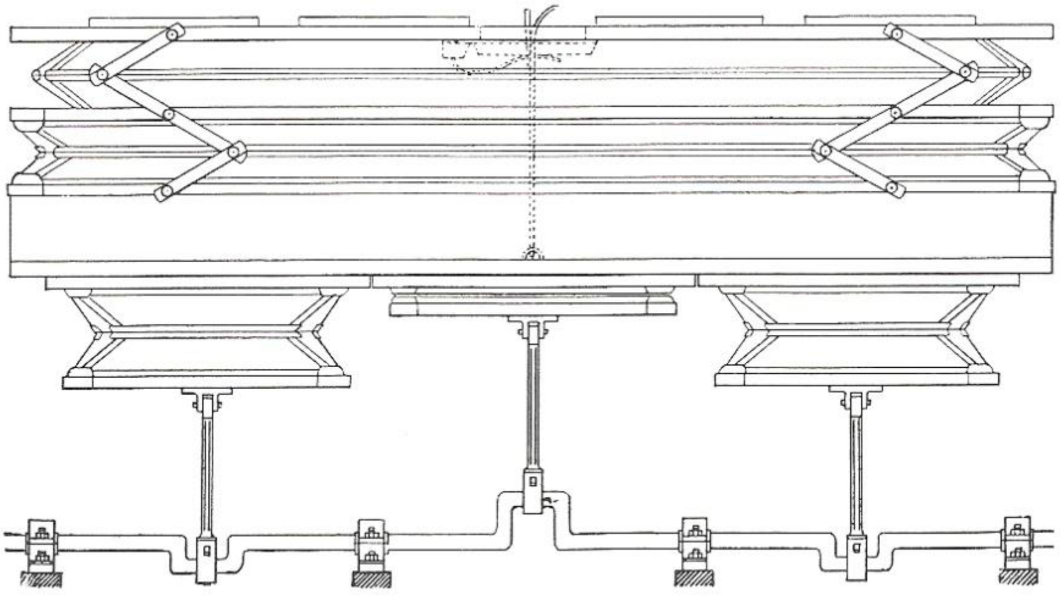

Figure 6 A double rise reservoir with 3-crank feeders (After Barnes [1])

2.7 Electrically driven rotational fans

A small hand operated fan designed to replace hand operated blacksmiths bellows is shown in Figure 7.

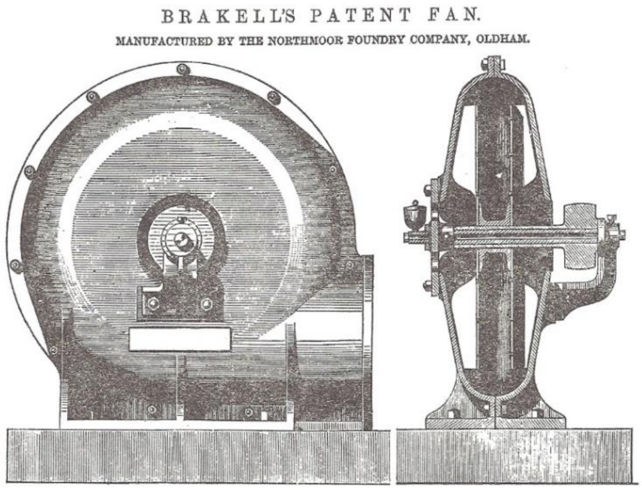

Figure 7 Hand operated blacksmiths fan (After Elvin [2])

Laycock and Bannister was an old family organ building company established in 1842. Using the rotational principal of the hand operated blacksmiths fan they produced a very primitive blower, shown in Photographs 1, with a cruciform impeller. The Layban impeller was 600mm in diameter and 175m wide. The outer case was 975 x 975 x 200mm internal width and constructed from solid pine with mild steel angle reinforcement. This blower dates from the early 1900's.

Photograph 1 The Layban outer case and cruciform impeller

An early rotational blowing plant, that was designed especially for organ blowing applications, was introduced in 1902 by The Kinetic Company [4]. The unit, shown in Photograph 2, consisted of a series of metal impellers mounted on a common shaft to give the desired pressure and flow rate. The drawback with this design was that the early 8 pole motors ran at 750 rpm and each stage was only capable of raising the pressure by 50mm water gauge.

Photograph 2 Kinetic company impeller arrangement

Whilst all new technologies have some period during which reality fails to live up to expectations, the move from feeder driven blowing was relatively pain free, and the concerns of pulsating wind affecting the tone of the pipe work were not borne out [2]. From this first design of a special fan unit for organ blowing applications, further developments in fan design have continued and further details are contained in Appendix A.

Backward curved centrifugal fans are now the norm in organ building applications and only in exceptional cases is a feeder arrangement employed. The use of feeders is usually exclusively limited to the restoration of old instruments; where it is important to keep historical correctness, or new historic copies and reconstructions.

Figure 8 shows an arrangement to feed low and high-pressure wind to an organ and is typical of most electrically driven blower systems. The system includes two single rise sprung reservoirs and two control valves which are examined in Chapter 11.

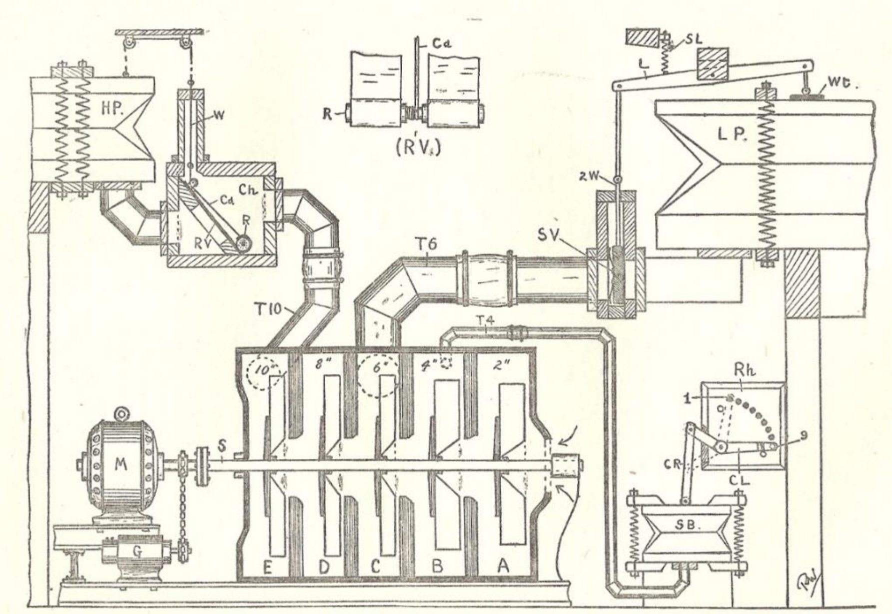

Figure 8 A multi-stage low and high pressure Kinetic blowing plant
(After Whitworth [4])

Initially the noise from the large fan assemblies necessitated the blowing plants to be housed in a separate room. Typical locations were cellars or church towers. In some installations, the route taken by the ducting from the blowing plant to the organ chamber could approach 30m. To control and cut off the air supply to the reservoir, a control valve is fitted, either at the blower or at the reservoir.

The first fans used large low speed motors that were directly attached to the impeller or alternatively, the fan was driven by a pulley and belt. To develop the required flow rates and pressure the impellers tended to be large with few blades. As fan and motor designs improved, the footprint of the units allowed inclusion within the organ chamber or close by in a noise reducing cabinet. Key to this development was the development of smaller more powerful 1500 & 3000 rpm single and three phase electric motors.

In a modern wind system, the basic function of the reservoir is no longer that of storage. The reservoir now acts to regulate the air pressure, and in this role, from a purely dynamic consideration, it can be made considerably smaller Norman [5] Moyes [6]. Modern installations tend to have short connecting trunks, typically less than 1m long and small reservoirs.

Recently, with the introduction of electronic inverters, the speed of the blower motor can be controlled to better meet the demand of the organ. Nothing is new, this is a modern version of how the early systems were controlled. In the bottom right hand corner of Figure 8, a control rheostat, which is operated from a single rise pneumatic motor, matches the motor speed with the wind demand. Whilst speed control solves some of the noise problems, it is not a complete fix.

2.8 Control valves

The move from the feeder system to the use of electrically driven blowers, means that some form of control is needed to shut off the supply of air once the reservoir has reached the correct height and wind pressure. This is achieved using three basic types of control valve.

2.8.1 The guillotine control valve

The guillotine control valve is shown in Figure 8, labelled SV. It is the simplest form of control valve and comprises a vertical plate connected to the top of the reservoir by a thin cord. The cord allows the vertical plate to move up and down in synchronisation with the reservoir top. When the reservoir is empty, the valve is in the “up” position allowing air into the reservoir. As the reservoir rises the valve moves down and closes when the reservoir is full.

2.8.2 The roller blind control valve

The roller blind is shown in Figure 8 connected to the HP reservoir and is similar to the guillotine valve, but the flat vertical plate is replaced by an inclined flexible blind that rolls up and down as the reservoir rises and falls.

2.8.3 The internal control valve

The internal control valve can take several forms but its main difference to the other two valves is that it is located inside the reservoir. Its connection to the reservoir top can be by a strong cord or by a system of metal linkages. The valve is usually arranged to have some form of mechanical advantage so that the movement of the reservoir top is geared to give large valve opening for small reservoir top movements.

Details of the control valves used in this research are detailed in Chapter 11.

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3 Pipe organ survey

3.1 Preamble

Initially, a series of listening tests were considered to determine organ pipe flutter. Whilst this would provide useful information on the human perception of organ pipe flutter it would not address the fundamental question of which types of wind systems are more likely to cause organ pipe flutter.

As an alternative to a series of listening tests, information regarding the occurrence of organ pipe flutter and wind system information could be obtained from actual pipe organs. The Pipe Organ Survey is structured to confirm the prevalence of organ pipe flutter and wind systems information. Using this approach, organ pipe flutter information is provided by organ tuners who are aware of organ pipe flutter and best placed to collect the information contained in the survey. The survey contains 31 questions spread over four sections, and each pipe organ was surveyed as part of a regular tuning visit without any cost. In the next section the survey is described in detail together with the rationale behind each question.

The University of Salford Ethics panel approved the survey in January 2017 and the survey paperwork was sent to over 20 organ tuners in March 2017. During the course of 2017, 8 organ tuners responded with details of 83 pipe organs. Further information is contained in appendix B.

3.2 Questionnaire Details

3.2.1 Section 1 Location and general information of the organ

Location				
Builder		Date		
Number of stops	Swell	Great	Pedal	
Position		Size of building	Small.	Medium	Large.
Approximate reverberation time in seconds		Less than 1	1 to 3.	over 3
Temperament		Equal	Other.	

Comments

Should further general information be needed about the instrument the location can be used to access the National Pipe Organ Register.

The original organ builder gives some indication about the style of the organ, particularly wind pressures and voicing style. Low wind pressures generally support classical voicing styles whilst higher wind pressures would indicate more romantic tendencies.

The date gives some indication of the period of the organ. Organ building is no different to any other art form. There are good periods and periods that would be best “glossed over”.

The number of stops gives some idea of the requirements of the wind system.

The position of the organ is very important in the projection of the sound. In the United Kingdom, west end positions are not common. At the west end the pipework is allowed to speak unimpeded into the body of the church and tends to favour lower wind pressures. Often the organ is crammed into a chamber at floor level or elevated on a gallery in the north or south choir. The proximity of a large stone or brickwork pillar often impedes the passage of sound into the nave. To overcome this, the organ builder often needs to use higher wind pressures.

The size of the organ should be in scale to the size of the building. The size of the building will also have a great influence on the reverberation time. The reverberation time is a function of the volume and absorbent surfaces of the room and can be calculated using Sabine's equation. Unlike the orchestra, the organ flourishes in a lively acoustic, and over three seconds adds grandeur to the instrument and absorbs some of the power that would otherwise be overpowering. It is always encouraging to hear the pipework answering back.

Often the temperament is ignored. Generally, organs used for playing and accompanying a wide repertoire are tuned to equal temperament. It is only in certain applications that the repertoire will dictate that the organ will be tuned to some un-equal temperament. Normally these organs will be historic instruments or modern copies specifically used for early music making.

3.2.2 Section 2 Scale and organ pipe flutter on four selected organ pipes

The scale of an organ pipe or its width is the main feature that gives the pipe its distinctive

No Flutter =NF **Slight flutter = SF** **Moderate Flutter =MF** **Extreme Flutter = EF**

Organ Pipe		Scale	Soundboard Degree of Flutter	Body of the church Degree of Flutter
Great 8ft open 6″	c 49			
Great 4ft principal 6″	c 37			
Gt or Sw 8ft stp flute 6″	c 49			
Gt or Sw 8ft string 6″	c 49			

Comments.

sound or timbre. A wide scale pipe reinforces the fundamental and low harmonics and has few if any high harmonics. Flute stops normally have wide scales. The other extreme is a narrow scale pipe which has little fundamental but is rich in high harmonics. String stops normally have narrow scales and fall into this category. Scales which are neither wide nor narrow give rise to the diapason tone which is regarded as the true pipe organ tone or timbre. The diapason stop is rich in fundamental and harmonics. Further information regarding pipe scales is contained in Appendix C

The four pipes chosen all have the same frequency of approximately 1kHz but differ in scale and timbre.

The first pipe is from the great 8ft open diapason rank

The second pipe is from the great 4ft principal rank and is slightly smaller in scale than the 8ft open diapason.

The third pipe is from the great or swell 8ft stopped flute rank. The choice of the swell division has been given because often a stopped flute is not available on the great organ.

The fourth pipe is from the great or swell 8ft string rank.

The organ tuner is asked to listen to the pipe at the soundboard, near field, and use the four descriptors to describe the level of flutter. The procedure is repeated in the body of the church. Some tuners have reported that often pipe flutter is a problem at the soundboard yet is not apparent in the body of the church.

3.2.3 Section 3 Wind system details

SR = Single Rise DR = Double Rise W = Weighted Sp = Sprung

Example: A Swell 4 ft. x 6 ft. double rise weighted reservoir = 4x6 DR W

	Break down	Swell	Great
Reservoir type & construction			
Full Reservoir height inc well (inches)			
Approx. Wind pressure (inches) WG			
Cut-off valve details			
Blower make		Motor HP.	1 or 3 phase
Blower speed		Low 1500 rpm	High 3000 rpm
Blower location			
Approximate distance from blower to reservoir (feet)			

Comments

Just as it is unusual to find two pipe organs with identical specifications, so it is unusual to find two pipe organs with identical wind systems. This section of the survey seeks to detail the type of wind system, of which the major components are the blower and reservoir type. Some instruments may have a single reservoir, others may have a reservoir to each division. Generally, the pipe organ has a single blower supplying all divisions.

The reservoir type and construction will have a direct effect on the resonant frequency of the wind system and this is examined in Chapter 7. For a weighted wind system, the wind pressure and the filled height of the reservoir will determine the resonant frequency of the reservoir. For a sprung reservoir system, the resonant frequency will be determined by the mass of the reservoir top and will not change if the reservoir wind pressure is changed. This portion of the survey will give a clear indication of the natural resonant frequency of the wind system.

The influence of the cut-off valve is not fully understood. This question has been included to obtain more information and determine the significance of the various designs on reservoir top vibration and organ pipe flutter.

The blower details and location provide the final information for this section.

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3.2.4 Section 4 Tremulant stop details

Swell stop selected for use with the tremulant				
Type of tremulant				
Speed & effectiveness of the tremulant	Slow	Fast	Good	Poor

If possible, please give the approximate number of vibrations per second.

Comments

The Tremulant is a device designed to create a vibrato effect on the sounding organ pipe and it is normal to control the depth and frequency by mechanical or electronic means. More detailed information about the tremulant can be found in Appendix D. Musically, the tremulant can, if correctly adjusted, add to the musical performance. Unfortunately, sometimes little care has been taken in the setting of the tremulant and in this situation, many are rendered ineffective and useless.

It is unusual to find a tremulant on each division and if one is included it is more than likely to be located on the swell organ. The tremulant is designed to be used with a single stop normally an imitation reed or a flute. The ear is most sensitive to amplitude modulation of 4 Hz and it will be interesting to see how the frequencies obtained from the survey compare with this optimum frequency.

Tremulants of the wind dumping type are the most common. These were originally pneumatically operated and are very difficult to adjust. The historic Dom Bedoes tremulant, that is designed to be mounted inside the connecting trunk, is very rarely found.

The Austin Universal wind-chest produces flutter free pipework and requires a special tremulant to modulate the pipework. This takes the form a mechanism fitted above the pipework composed of a series of slowly rotating horizontal blades. More detailed information about the Austin Universal wind-chest can be found in Appendix E.

3.3 Results

The results for the four sections of the organ building survey are shown in Tables 1 to 10

3.3.1 Section 1 Location and general information of the organ

The results for section 1 are shown in Tables 1 & 2 and are grouped by each organ tuner.

Ref	date	swell	great	pedal	total	east	west	small	medium	large	less 1	1 to 3	over 3	equal	other
1	1890	8	6	2	16	x			x			x		x	
2	1905	5	5	2	12	x		x			x			x	
3	1905	7	6	2	15	x			x		x			x	
4	1902	5	5	1	11	x		x			x			x	
5	1908	9	6	2	17		x	x			x			x	
6	1989	11	6	5	22		x		x			x		x	
7	1894	7	6	10	23	x		x			x			x	
8	1973	9	7	5	21	x		x			x			x	
9	1965	8	9	6	23		x		x			x		x	
10	1925	4	4	2	10	x		x			x			x	
11	1900	4	4	2	10	x		x			x			x	
12	1960	5	6	4	15	x			x		x			x	
13	1905	9	7	2	18	x			x		x			x	
14	1963	7	7	4	18		x	x				x		x	
15	1861	10	8	3	21	x			x		x			x	
16	1864	6	8	1	15	x			x			x		x	
17	1893	7	6	1	14	x			x		x			x	
18		4	5	1	10	x		x			x			x	
19	1900	3	3	1	7	x		x			x			x	
20		13	9	7	29	x			x			x		x	
21	1890	7	8	2	17	x			x			x		x	
22	1880	9	10	8	27		x		x			x		x	
23	1920	3	5	1	9		x		x		x			x	
24	1905	6	7	2	15		x		x			x		x	
25	1975	10	7	3	20	x			x		x			x	
26		9	6	4	19		x		x			x		x	
27		5	5	1	11	x		x			x			x	
28	1983	7	8	1	16	x			x				x		x
29	1991	9	10	7	26	x			x			x		x	
30	1900	5	8	1	14	x			x			x		x	
3`	1890	4	7	3	14	x		x				x		x	
32	1990	6	10	7	23		x	x			x			x	
33	1920	5	4	1	10		x		x			x		x	
34	1920	9	7	3	19	x			x			x		x	
35	1858	6	6	1	13	x			x			x		x	
36	1964	7	8	3	18	x		x			x			x	
37	1959	10	11	6	27	x			x			x		x	
38	1982	10	11	6	27	x			x			x		x	
39	1999	10	9	5	24		x		x			x		x	
40	1960	7	7	3	17	x			x		x			x	

Table 1 Section 1 summary organs 1 to 40

Ref	date	swell	great	pedal	total	east	west	small	medium	large	less 1	1 to 3	over 3	equal	other
41	1996	6	7	2	15		x		x		x			x	
42		7	7	3	17		x		x			x		x	
43	1990	7	7	2	16	x			x		x			x	
44	1989	8	7	7	23		x		x			x		x	
45	1978	4	4	1	9	x		x			x			x	
46	1986`	6	7	1	14	x			x		x			x	
47	2016	16	10	11	37	x				x		x		x	
48	1864	6	11	2	19				x			x		x	
49	1890	4	5	1	10				x			x		x	
50	1980	9	8	7	24	x			x			x		x	
51	2004	6	7	1	14		x		x			x		x	
52	1864	5	6	1	12		x		x			x		x	
53	1957		3		3			x			x			x	
54	1900	5	5	1	11		x		x			x		x	
55	1889	4	5	1	10	x			x			x		x	
56		9	7	7	23		x		x			x		x	
57	1880	4	4	1	9	x			x			x		x	
58	1890	10	9	2	21	x			x			x		x	
59	1900	3	4	1	8	x			x			x		x	
60	1900	7	6	3	16	x			x			x		x	
61	1890	13	9	10	32	x			x			x		x	
62	1890	8	6	1	15	x			x			x		x	
63	1905	5	5	1	11	x			x			x		x	
64	1912	4	4	2	10	x			x			x		x	
65	1880	7	7	4	18		x		x			x		x	
66	1963	5	4	1	10		x		x			x		x	
67	1890	8	7	2	15	x			x			x		x	
68	1890	9	7	2	18		x		x			x		x	
69	1883	5	5	1	11	x			x			x		x	
70	1880	9	8	1	18	x			x			x		x	
71	1890	8	7	2	17	x			x			x		x	
72	1872	10	7	3	20	x			x			x		x	
73	1874	9	10	3	22					x				x	
74	1870	7	8	2	17	x			x			x		x	
75		9	8	5	22		x		x			x			
76		6	5	2	13	x			x			x		x	
77	1900	7	6	2	15	x			x			x		x	
78		5	5	1	11	x			x			x		x	
79	1969	6	6	6	18	x			x			x		x	
80	1937	6	6	3	15		x		x			x		x	
81	1989	3	2		5			x			x			x	
82	1937	6	6	3	15		x	x				x		x	
83	1865	7	7	4	18	x		x				x		x	

Table 2 Section 1 summary organs 41 to 83

3.3.2 Section 2 Scale and organ pipe flutter on four selected organ pipes

3.3.2.1 Pipe scale results

The results of the pipe scales for each of the four organ pipes are shown in Tables 3 & 4.

Ref	open mm	princ mm	flute mm	string mm
1	20	16	12	12
2	22	16	16	13
3	19	14	16	13
4	19	16	16	13
5	22	19	13	13
6	22	16	25	13
7	19	13	tri	13
8	16	19	13	13
9	16	13	19	13
10	19	16	sqr	10
11	19	19	19	19
12	19	13	19	13
13	19	16	sqr	10
14	17	17	sqr	11
15	17	13	13	13
16	17	16	sqr	10
17	19	16	sqr	10
18	17	13	sqr	11
19	19	22	sqr	13
20				
21	19	19	17	13
22	18	17	20	9
23	11	10	12	8
24	21	20	17	9
25	10	9	8	7
26	10	9	6	7
27	10	9	sqr	7
28	8	7	10	7
29	10	9	sqr	7
30				
31				
32				
33				
34				
35				
36	30	21	30	19
37	34	23	30	20
38	33	21	23	17
39	29	23	23	19
40	30	25	30	19

Table 3 Pipe scales organs 1 to 40

Ref	open mm	princ mm	flute mm	string mm
41	19	16	16	13
42	19	16	19	13
43	19	16	16	13
44	19	16	19	11
45	19	16	16	11
46	19	16	19	13
47	22	19	sqr	16
48	16	13	13	13
49	17	17	16	16
50	19	16	19	16
51	16	16	13	13
52	13	13	10	8
53	13	16	10	10
54	13	13	16	13
55	16	13	13	13
56	13	13	16	16
57	13	16	16	16
58				
59	13	13	19	13
60	13	16	16	13
61	16	13	13	13
62	13	13	13	16
63	13	13	16	13
64	16	16	13	10
65	16	16	13	8
66	13	16	19	13
67	16	13	16	13
68	13	13	13	16
69				
70	16	16	16	16
71	13	16	16	13
72	16	13	13	13
73	16	16	13	13
74	19	16	16	16
75	16	13	13	16
76	16	13	13	13
77	19	16	19	16
78				
79	19	19	16	13
80	28	25	25	20
81		24	sqr	
82	30	28	22	20
83	28	25	sqr	21

Table 4 Pipe scales organs 41 to 83

3.3.2.2 Organ pipe flutter results

The results for 652 sounding pipes are shown in Tables 5 & 6 and are grouped by each organ tuner. The first column for each stop is at the soundboard and the second is in the body of the church. Also included, fres Hz, is an approximation of the resonant frequency of the reservoir using equations 5&6 in Chapter 7.

Ref	8ft open	8ft open	4ft princ	4ft princ	8ft stp fl	8ft stp fl	8ft string	8ft string	fres Hz
1	SF	NF	NF	NF	SF	SF	MF	SF	10.7
2	SF	MF	SF	SF	SF	SF	NF	NF	11.0
3	NF	NF	NF	NF	SF	SF	NF	NF	11.9
4	SF	SF	NF	NF	SF	MF	NF	SF	12.3
5	MF	MF	MF	SF	SF	SF	SF	NF	11.9
6	SF	SF	NF	NF	SF	SF	NF	NF	15.1
7	MF	MF	MF	SF	MF	SF	EF	MF	-
8	SF	SF	SF	NF	SF	SF	NF	NF	12.9
9	NF	NF	NF	NF	SF	NF	NF	NF	Schwim
10	SF	SF	SF	SF	NF	NF	NF	NF	13.5
11	NF	NF	NF	NF	SF	SF	NF	NF	13.0
12	NF	SF	NF	NF	SF	SF	NF	NF	SR W/S
									-
13	SF	SF	SF	SF	SF	SF	SF	SF	10.7
14	NF	NF	NF	NF	SF	SF	NF	NF	Comp
15	MF	MF	SF	SF	SF	SF	SF	SF	12.4
16	SF	SF	SF	SF	MF	MF	SF	SF	12.9
17	SF	SF	SF	SF	SF	SF	NF	NF	12.3
18	MF	MF	SF	SF	SF	SF	SF	SF	12.5
19	SF	SF	SF	SF	SF	SF	SF	SF	14.1
									-
20	SF	NF	SF	NF	MF	NF	SF	NF	9.3
21	SF	NF	SF	NF	MF	SF	SF	NF	11.0
22	NF	NF	NF	NF	SF	NF	NF	NF	12.3
23	SF	SF	SF	SF	SF	NF	NF	NF	12.9
24	NF	NF	NF	NF	SF	SF	SF	NF	9.3
25	MF	MF	MF	MF	MF	MF	MF	MF	10.6
26	SF	NF	SF	SF	SF	NF	SF	SF	11.0
27	SF	NF	SF	NF	MF	NF	NF	NF	12.3
28	EF	SF	EF	SF	MF	SF	MF	NF	13.5
29	MF	NF	MF	NF	NF	NF	NF	NF	12.3
30	MF	MF	MF	MF	MF	MF	SF	SF	10.7
31	MF	MF	MF	MF	MF	MF	MF	MF	10.1
32	MF	MF	MF	MF	MF	MF	SF	SF	12.3
33	SF	SF	SF	SF	MF	MF	MF	MF	10.7
34	SF	SF	NF	NF	SF	SF	NF	NF	12.3
35	NF	NF	NF	NF	NF	NF	NF	NF	10.7
36	NF	NF	NF	NF	NF	NF	NF	NF	12.9
37	NF	NF	NF	NF	NF	NF	NF	NF	11.8
38	NF	NF	NF	NF	NF	NF	NF	NF	12.8
39	NF	NF	NF	NF	NF	NF	NF	NF	14.3
40	NF	NF	NF	NF	NF	NF	NF	NF	9.1

NF No Flutter SF Slight Flutter MF Moderate Flutter EF Extreme Flutter

Table 5 Section 2 summary of organ pipe flutter results organs 1 to 40

Ref	8ft open	8ft open	4ft princ	4ft princ	8ft stp fl	8ft stp fl	8ft string	8ft string	fres Hz
41	NF	NF	NF	NF	NF	NF	NF	NF	
42	NF	NF	NF	NF	NF	NF	NF	NF	13.7
43	NF	NF	NF	NF	NF	NF	NF	NF	12.9
44	NF	NF	NF	NF	NF	NF	NF	NF	
45	NF	NF	NF	NF	NF	NF	NF	NF	
46	NF	NF	NF	NF	NF	NF	NF	NF	13.5
47	NF	NF	NF	NF	NF	NF	NF	NF	13.2
48	NF	NF	NF	NF	NF	NF	NF	NF	12.3
49	NF	NF	NF	NF	NF	NF	NF	NF	13.5
50	NF	NF	SF	NF	NF	NF	NF	NF	11.0
51	NF	NF	NF	NF	NF	NF	NF	NF	11.4
52	NF	NF	NF	NF	NF	NF	NF	NF	11.0
53					MF	MF			
54	NF	NF	NF	NF	NF	NF	NF	NF	11.9
55	NF	NF	NF	NF	NF	NF	NF	NF	11.9
56	SF	SF	SF	NF	SF	NF	SF	NF	12.5
57	NF	NF	NF	NF	NF	NF	NF	NF	12.9
58	NF	NF	NF	NF	NF	NF	NF	NF	12.1
59	SF	SF	SF	SF	SF	SF	SF	SF	13.5
60	NF	NF	NF	NF	NF	NF	NF	NF	13.2
61	MF	MF	MF	MF	MF	MF	MF	MF	
62	NF	NF	NF	NF	NF	NF	NF	NF	11.9
63	NF	NF	NF	NF	NF	NF	NF	NF	12.9
64	NF	NF	NF	NF	NF	NF	NF	NF	12.3
65	SF	SF	SF	SF	SF	SF	SF	SF	12.9
66			SF	NF	SF	NF	SF	NF	
67	NF	NF	NF	NF	NF	NF	NF	NF	12.9
68	NF	NF	NF	NF	NF	NF	NF	NF	12.9
69	NF	NF	NF	NF	NF	NF	NF	NF	13.5
70	NF	NF	NF	NF	NF	NF	NF	NF	12.3
71	SF	SF	SF	SF	SF	SF	SF	SF	13.5
72	MF	MF	MF	MF	MF	MF	MF	MF	12.9
73	NF	NF	NF	NF	NF	NF	NF	NF	12.9
74	NF	NF	NF	NF	NF	NF	NF	NF	12.9
75	NF	NF	NF	NF	NF	NF	NF	NF	
76	SF	SF	SF	SF	SF	SF	SF	SF	
77	SF	SF	SF	SF	SF	SF	SF	SF	12.9
78	NF	NF	NF	NF	NF	NF	NF	NF	12.9
79	NF	NF	NF	NF	NF	NF	NF	NF	
80	NF	NF	NF	NF	NF	NF	NF	NF	9.6
81			NF	NF	SF	SF			13.5
82	NF	NF	NF	NF	NF	NF	NF	NF	9.3
83	SF	SF	SF	SF	SF	SF	SF	SF	10.7

NF No Flutter SF Slight Flutter MF Moderate Flutter EF Extreme Flutter

Table 6 Section 2 summary of organ pipe flutter results organs 41 to 83

3.3.3 Section 3 Wind system

The results for section 3 are shown in Tables 7 & 8 and are grouped by each organ tuner.

Ref	DR	SR	Wedge	Schwim	comp	Jelly	Weighted	Sprung	Height	W press	Guillotine	Internal	Roller B	1ph	3ph	1500	3000	dist ft
1	8x4						x		16	3	x			x			x	3
2	6x4						x		14	3.25	x			0.5			x	4
3	8x4						x		12	3.25	x			0.75			x	2
4	7x5						x		12	3	x				0.5		x	2
5	8x6						x		13	3				0.75			x	2
6			10x4				x		8	3	x			0.75			x	4
7	5x3						x		12		x			0.5			x	4
8	7x3						x		12	2.75	x			0.75			x	2
9				x				x		3.25		x		0.5			x	2
10	8x3						x		10	3	x			0.5			x	2
11	10x3						x		10	3.25	x			0.5			x	6
12		3x3					x	x	8			x		0.5			x	3
							10	1										
13	x						x		16	3			x	0.75		x		3
14				comp				x		3.25		x		0.25			x	2
15	x						x		13	2.75	x			1		x		2
16	x						x		12	2.75	x			0.5		x		4
17	x						x		12	3			x	0.5		x		2.5
18	x						x		14	2.5	x			0.5			x	2
19	x						x		10	2.75	x			0.5			x	2
							6	1										
20	5x2						x		16	4	-							6
21	x						x		14	3.25	x			0.5		x		3
22		3.5x3.5						x	12	3	x			1		x		2
23	5x4						x		11	3				0.5		x		1
24	7x5						x		18	3.5	x			1		x		6
25	7x7,5						x		13	3.75	x				1	x		25
26	7x5						x		18	2.5	x			0.5		x		1
27	6x3.5						x		12	3	x			0.25		x		6
28		x	x						12	2.5	x			0.5			x	1
29	7x3.5						x		12	3	x				3	x		20
							8	1								8	15	
30	8x5						x		16	3			x	0.5		x		1
31	6x4						x		18	3			x	0.75		x		6
32						2.5x2.0		x	12	3			x	1			x	1
33	6x4						x		16	3			x	0.5		x		1
34		4x3					x	x	12	3		x						
35	8x4						x		16	3	x			0.5		x		1
36		x					x		11	3	x			1		x		1
37		x					x	x	10.5	3.75			x		2		x	4
38		x						x	9	3.75			x	1			x	2
39				x					9	3	x				V/S			3
40	3x2						x		19	3.5	x				1	x		4

Table 7 Section 3 summary organs 1to 40

Ref	DR	SR	Wedge	Schwim	comp	Jelly	Weighted	Sprung	Height	W press	Guillotine	Internal	Roller B	1ph	3ph	1500	3000	dist ft
41		8x3		x						3		x	x	0.75			x	2
41		5x3					x		9	3.25			x	1			x	10
43	8.5x5						x		11	3	x			0.75			x	3
44	4x3			x						3		x		0.75			x	15
45					x					3				0.25			x	2
46	6x4						x		10	3	x			0.75			x	2
47		6x4					x		9	3.5		x			6	x		6
							7	2										
48	6x5						x		12	3			x		0.75	x		5
49	5x4						x		10	3			x		0.25		x	3
50		4x4					x		15	3	x			0.5			x	2
51		4x3					x		14	3								4
52	6x5						x		15	3			x	0.25		x		7
53														0.16			x	2
54	6x6						x		12	3.25	x			0.75		x		4
55	5x5						x		13	3				0.75		x		8
56		4x4						x	10	3.5					1		x	10
57	6x6						x		11	3	x			0.25			x	4
58	7x6						x		12	3.125			x	0.75		x		5
59	4x4						x		10	3	x			0.25			x	3
60	6x5						x		10.5	3	x			1		x		3
61				x						3		x		0.5			x	4
62	6x5						x		12	3.25			x	0.75			x	
63	5x5						x		11	3	x			0.5			x	3
64	5x4						x		12	3	x			0.5			x	8
65	5x5						x		11	3	x			0.5			x	6
66		3x2					x	x		3			x	0.5			x	3
67	6x6						x		11	3			x	0.5			x	4
68	5x5						x		11	3	x			0.75		x		5
69	5x5						x		10	3			x	0.25			x	5
70	6x6						x		12	3			x	0.75		x		12
71	6x5						x		10	3			x	0.5			x	10
72	6x6						x		11	3			x	1			x	13
73	7x6						x	x	11	3			x	1			x	8
74	6x6						x		11	3			x	0.75			x	8
75	6x6						x			3			x	0.75		x		10
76	5x5						x		10		x			0.75			x	6
77	6x5						x		11	3	x			0.5			x	4
78	5x4						x		11	3	x			0.75		x		6
79				x						3				0.5			x	
							26	2										
80	3x6						x		17	3.5			x	1.5		x		20
81	2.5x1.5						x		12	2.5				1		x		1
82	4x1.5						x		18	3.5			x			x		20
83	7x4.5						x		17.5	2.75			x			x		16

Table 8 Section 3 summary organs 41to 83

3.3.4 Section 4 Tremulant stop

The results for section 4 are shown in Tables 9 & 10 and are grouped by each organ tuner.

Ref	stop	Dump	fan	Pan	slow	fast	good	poor	freq
1	8ft ged	x				x	x		3
2	N>A								
3	8ft ged	x				x	x		5
4	V diap	x				x		x	4
5	N/A								
6	8ft st dia	x				x	x		4
7	N/A								
8	8ft st dia	x				x	x		4
9	N/A								
10	8ft R fl	x				x	x		4
11	8ft L ged	x				x	x		7
12	8ft fl	x				x	x		5
13	N/A								
14			x		x		x		2
15	N/A								
16	N/A								
17	N/A								
18	N/A								
19	8ft ged	x				x	x		-
20	N/A								
21	8ft fl	x			x		x		5
22	oboe	x				x		x	8
23	N/A								
24	N/A								
25	fl 8	x				x	x		6
26	ged	x			x		x		4
27	N/A								
28	N/A								
29	fl 8	x				x		x	7
30	N/A								
31	N/A								
32							x		
33	N/A								
34	N/A								
35	N/A								
36	st diap	x				x	x		5
37	N/A								
38	sw open				x		x		3
39	sw open	x				x	x		3
40	N/A								

Table 9 Section 4 tremulant summary organs 1to 40

Ref	stop	Dump	fan	Pan	slow	fast	good	poor	freq
41	8 fl			x	x		x		2
42	8 fl	x			x		x		
43	8 ged	x			x		x		3
44	8 fl			x	x		x		
45	8 ft		x		x		x		
46	N/A								
47	8 fl	x			x		x		3
48	N/A								
49	N/A								
50	NlA								
51	N/A								
52	8open	x					x		
53	8 ged					x			
54	N/A								
55	N/A								
56	viola			x		x		x	
57	N/A								
58	N/A								
59	N/A								
60	N/A								
61	N/A								
62	N/3								
63	N/A								
64	N/A								
65	N/A								
66	fl	x			x		x		
67	N/A								
68	open	x				x		x	
69	N/A								
70	N/A								
71	N/A								
72	open	x			x		x		
73		x				x	x		
74	N/A								
75	N/A								
76	open	x			x		x		
77	N/A								
78	N/A								
79			x		x			x	
80	N/A								
81	N/A								
82	N/A								
83	N/A								

Table 10 Section 4 tremulant summary organs 41to 83

3.4 Analysis

3.4.1 Section 1 General information

The oldest organ surveyed was built in 1858.

The average number of stops was 16.5.

70% of the organs were sited at the east end of the church in north or south organ chambers, the remainder were placed at the west end.

Most of the buildings were deemed to be of medium size (75%) with a reverberation time of 1 to 3 seconds (67%).

Only one organ had non-equal temperament.

3.4.2 Section 2 Organ pipe results

3.4.2.1 Organ pipe scale

The scales of each organ pipe are shown in Figures 9, 10, 11 & 12 together with the average scale shown in red. The average scale of the various pipes:

8ft open diapason	18mm
4ft principal	16mm
8ft stopped flute	16.5mm (note this is a stopped pipe)
8ft string	13mm

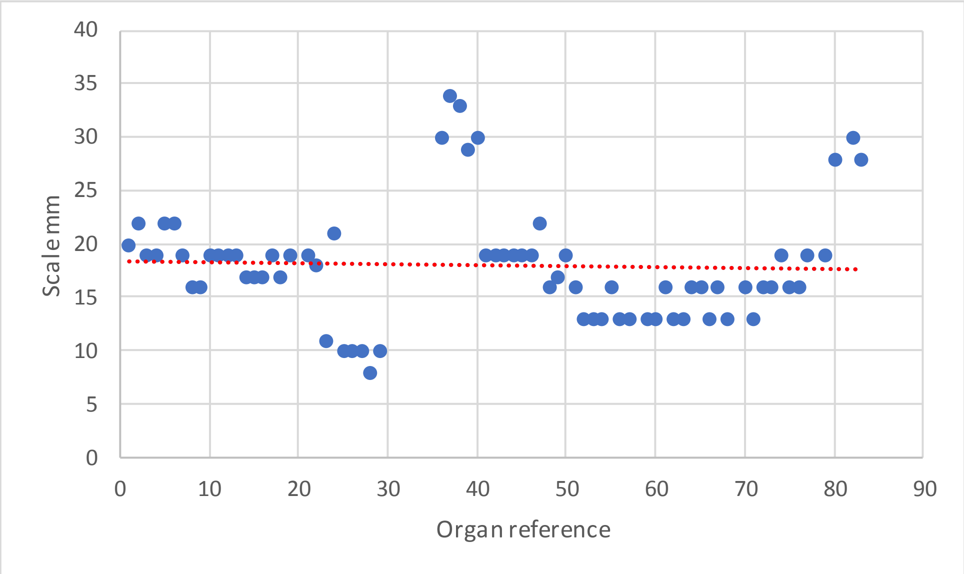


Figure 9 8ft open diapason pipe scale

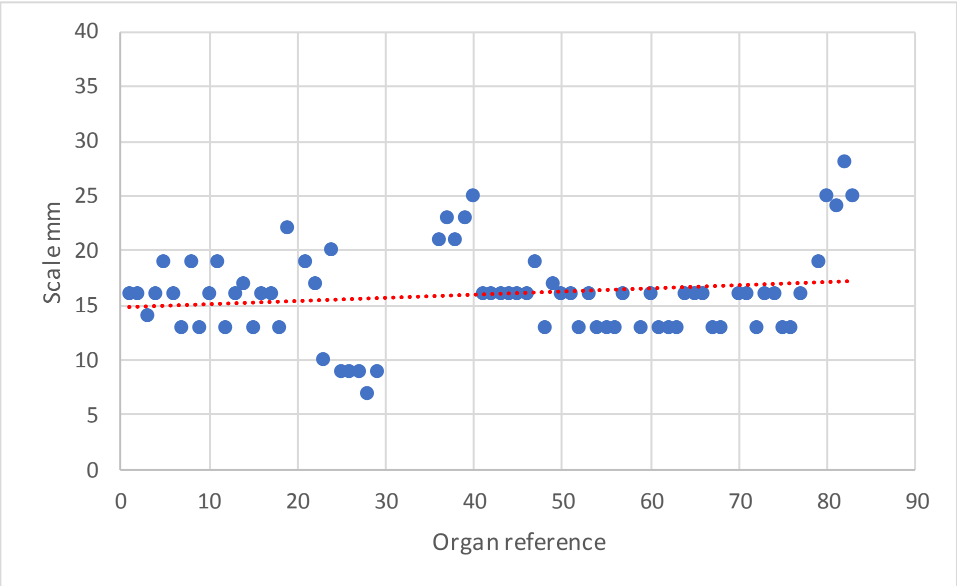


Figure 10 4ft principal pipe scale

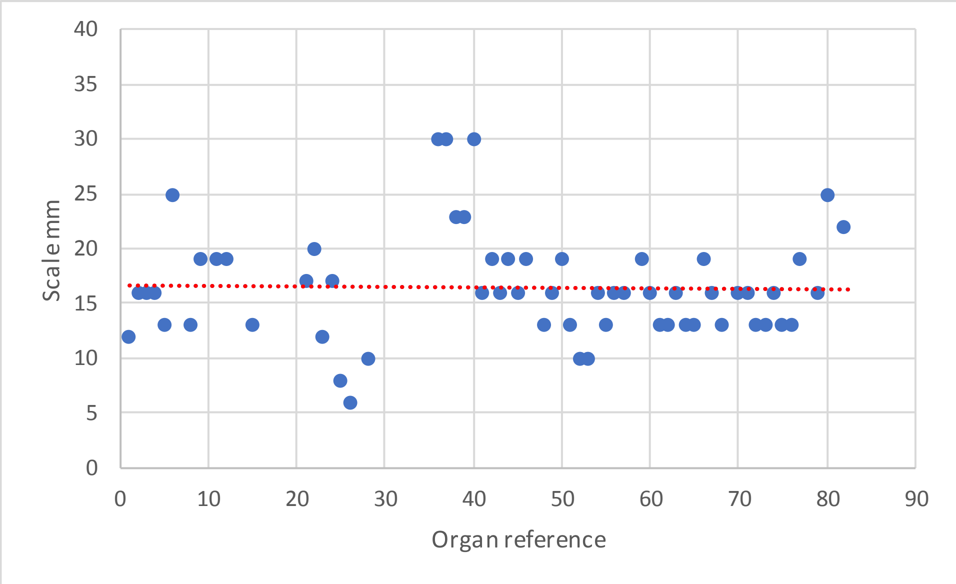


Figure 11 8ft stopped flute pipe scale

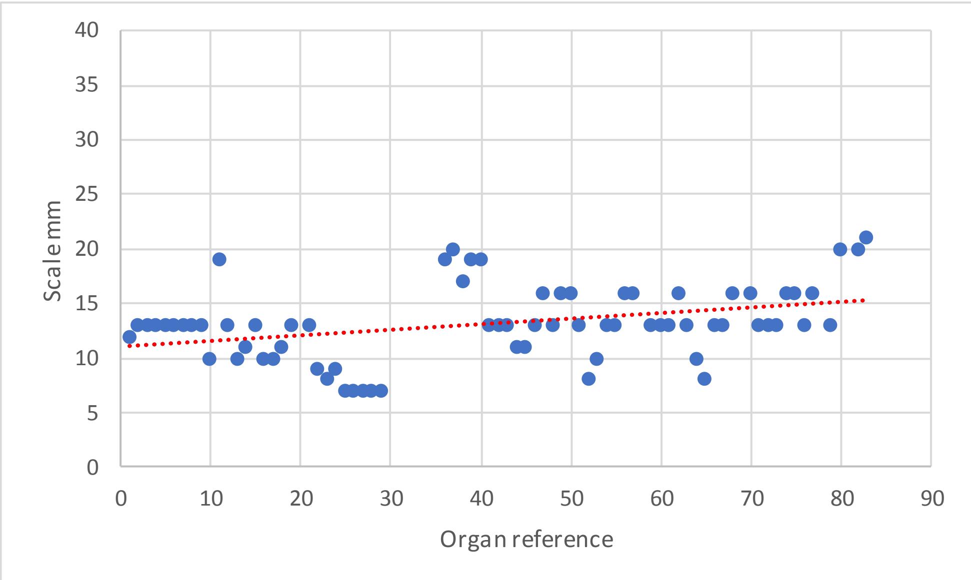


Figure 12 8ft string pipe scale

3.4.2.2 Organ pipe flutter on four selected pipes

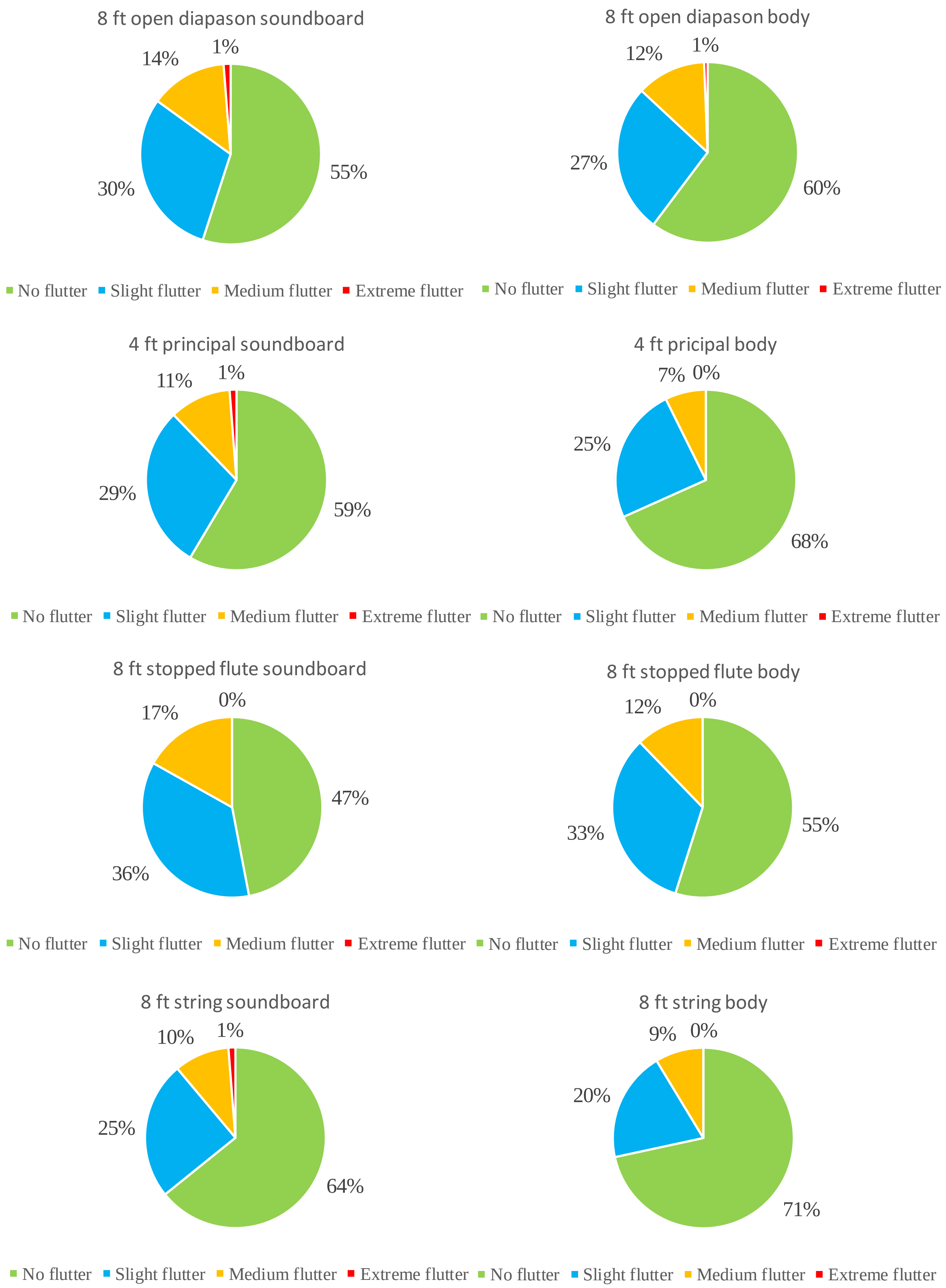


Figure 13 Organ pipe flutter at the soundboard and in the body of the church

The details of Tables 5 & 6 are shown in Figure 13 for the degrees of flutter on the four organ pipes at the soundboard and in the body of the church. Figure 14 shows the total amount of flutter for each organ pipe by combining the soundboard and body of the church results. The total combined results for all pipes are shown in Figure 15.

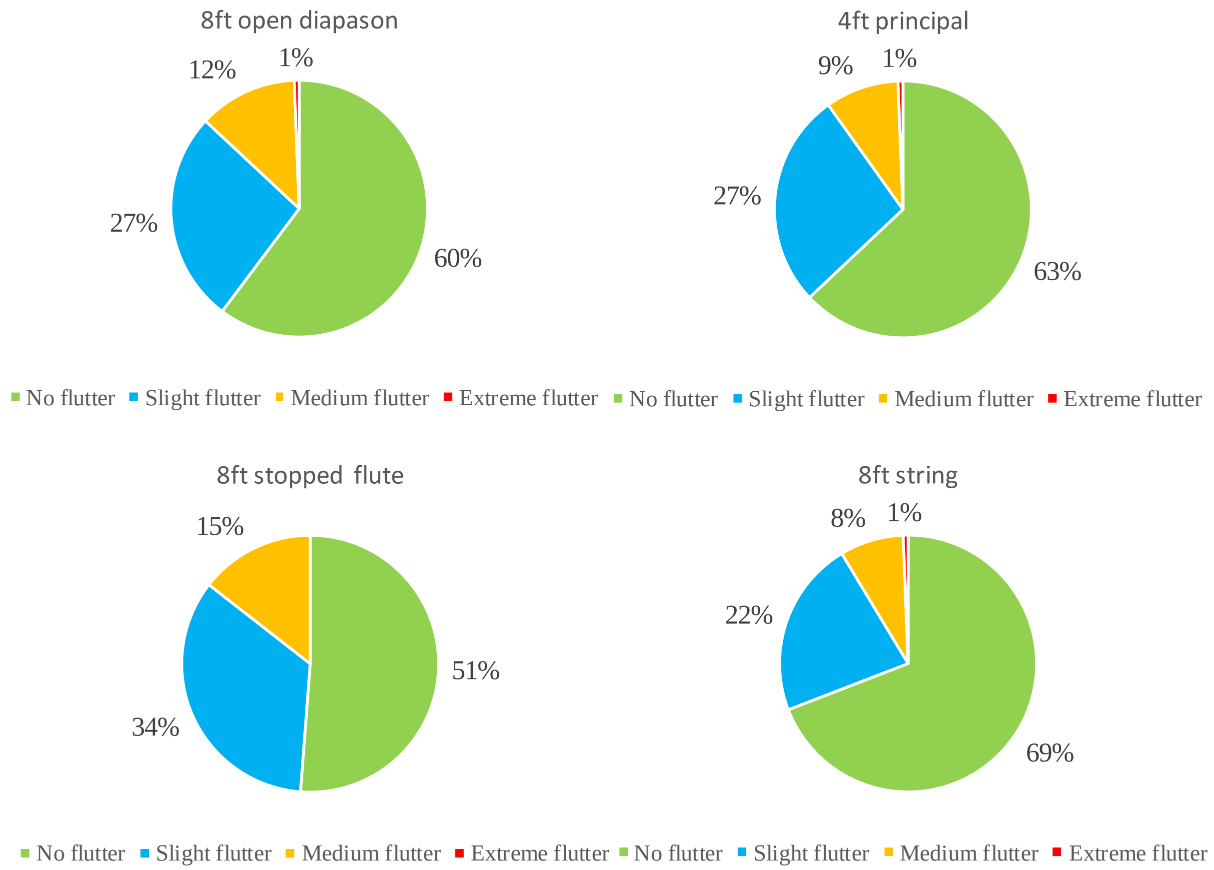


Figure 14 Organ pipe flutter for each organ pipe.

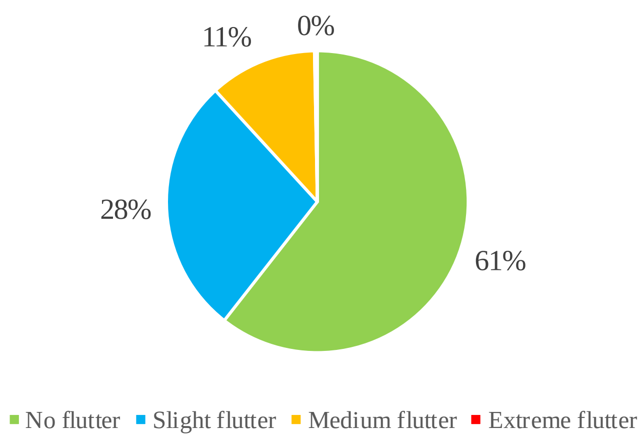


Figure 15 Total organ pipe flutter for all pipes

The overall organ pipe flutter results for the four pipes are shown in Figure 16.

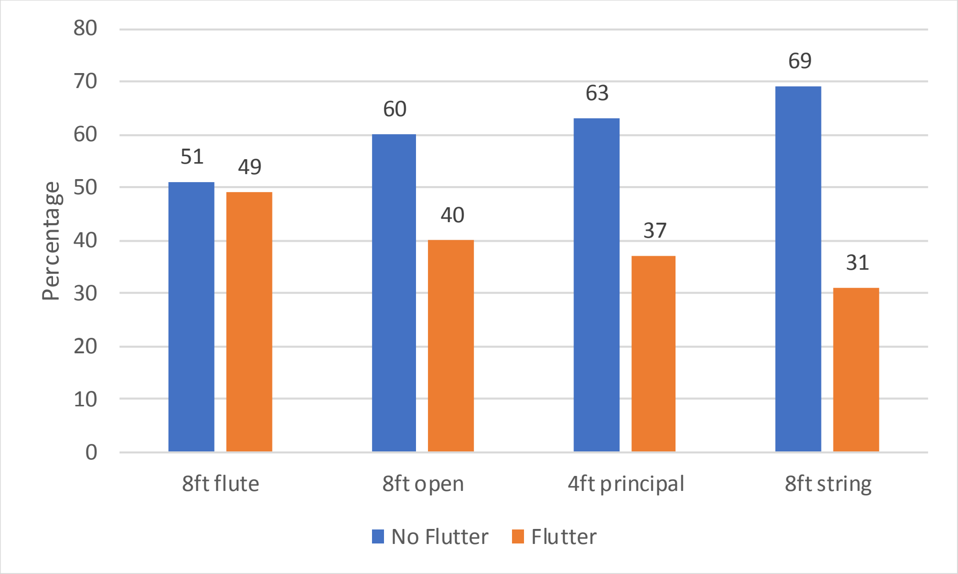


Figure 16 Overall organ pipe flutter for each organ pipe

3.4.3 Section 3 Wind system details

The results for reservoir loading, reservoir height, wind pressure, control valves, blower speed and blower to reservoir distance are shown in Figures 17 to 22.

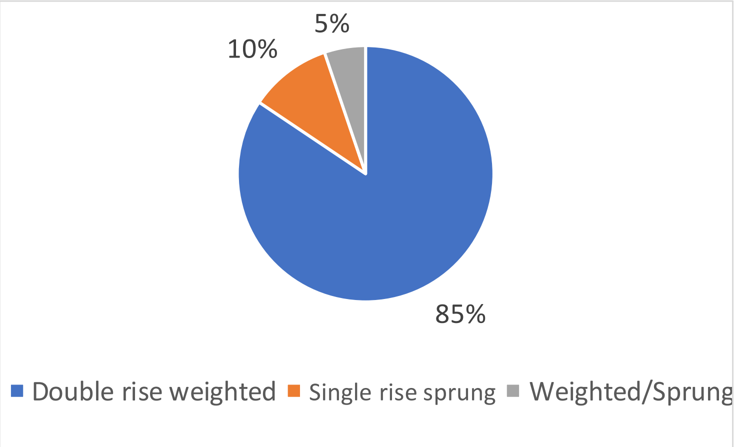


Figure 17 Reservoir loading results

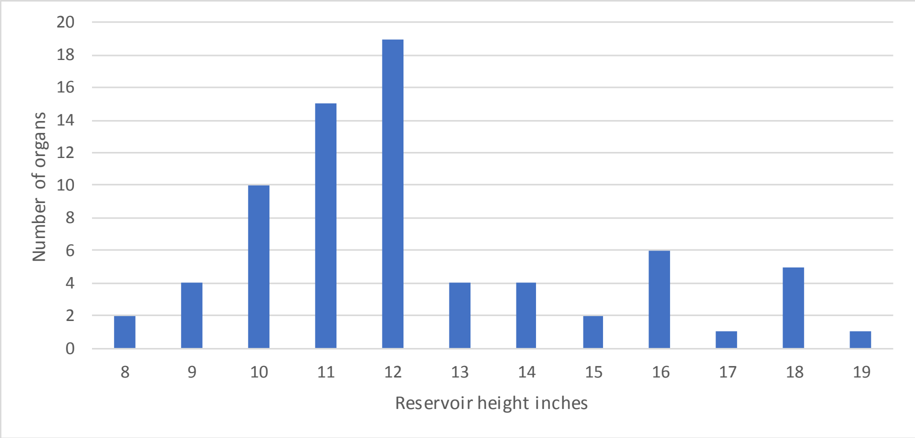


Figure 18 Reservoir height

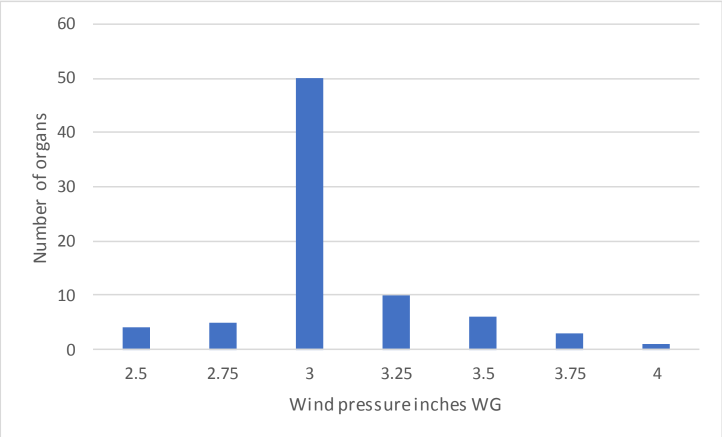


Figure 19 Reservoir wind pressure

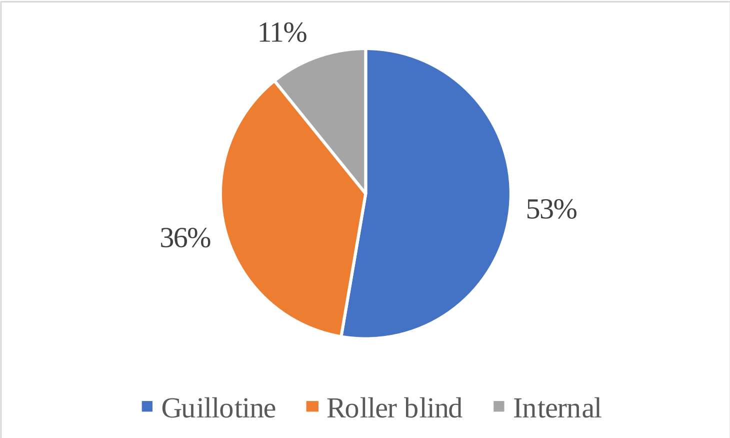


Figure 20 Control valve details

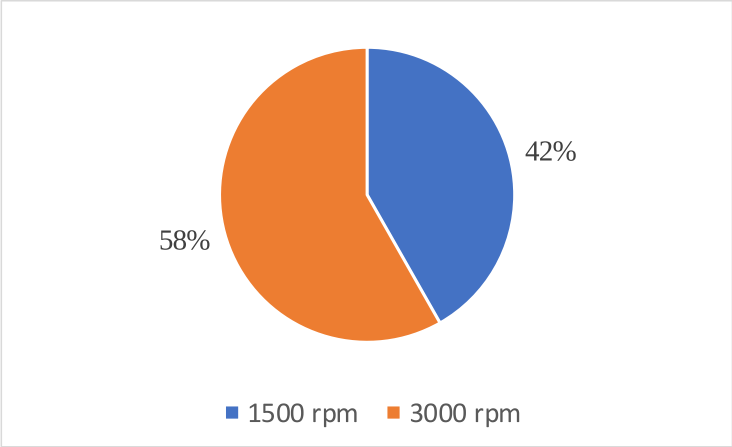


Figure 21 Blower speed

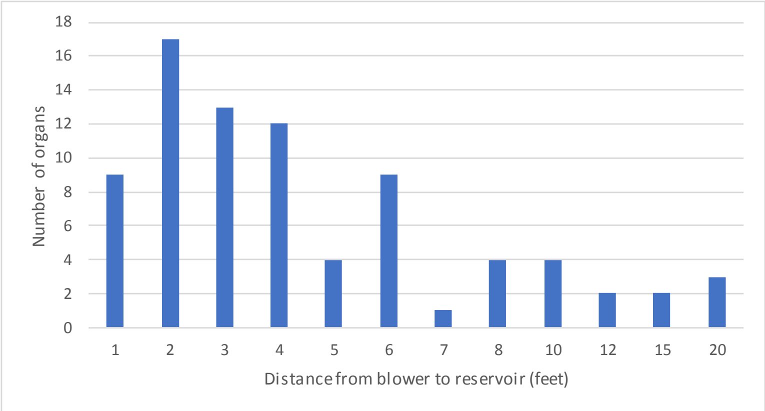


Figure 22 Blower to reservoir distance

3.4.4 Section 4 Tremulant details

The tremulant frequencies are shown in Figure 23.

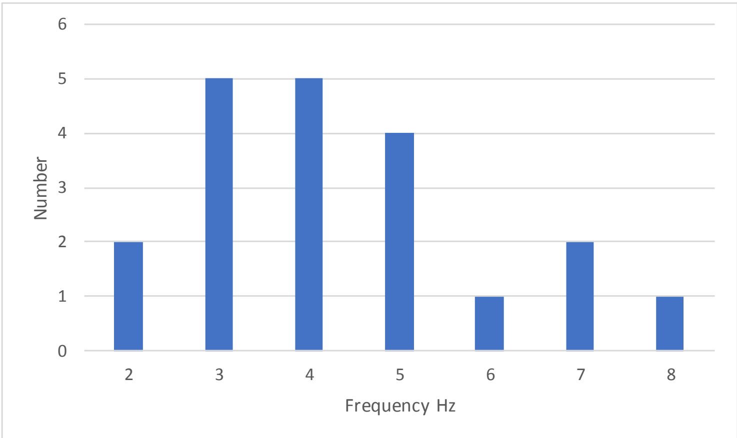


Figure 23 Tremulant frequency

3.5 Discussion

The survey is possibly the largest detailed pipe organ wind survey to have been conducted in the United Kingdom. Tables 1 & 2 show that the survey covered a fair representation of small two manual and pedal pipe organs found in the UK. The oldest organ surveyed was built in 1858 and the average number of stops was 16.5. 70% of the organs were sited at the east end of the church in north or south organ chambers. This is not surprising for many organs are located near the choir at the east end. The majority of the buildings (75%) were deemed to be of medium size with a reverberation time of 1 to 3 seconds (67%).

Over one third (39%) of the sampled pipes had flutter and the tuners detected slightly more organ pipes with flutter at the soundboard than in the body of the church. The variation in the results of the 8 tuners is interesting. Two tuners (36-40 Table 5) & (41-47 Table 6) found no pipes with flutter. One tuner surveyed 32 organs (48-79 Table 6) and found 11 organs (34%) had organ pipe flutter. This variation may be due to the random nature in which the organs were selected. Consideration needs to be given to quality of the selected pipework. Not all organs are regulated and tonally finished to the same standard. Also, some pipes may have gone off speech due to the accumulation of years of dust in the pipe mouths. The results also indicate that pipe scale may have some effect, with the least flutter associated with small scale pipes. Figures 9 to12 show the large divergence in pipe scales particularly for organs 36 to 40, 80 to 83, and no pipe scales were reported for organs 30 to 35. Some flute pipes were made from wood and no scale was recorded. Despite these unknown quantities, the survey confirmed the main objective that organ pipe flutter exists on 47 organs.

The results for the wind system show that 85% of the organs have double rise weighted reservoirs. The most common reservoir height was 12 inches (300 mm) and approximately 60% of the organs had a wind pressure of 3 inches (75mm) water gauge. The most popular cut-off device (53%) was the simple guillotine valve and 58% of the organs had a blower speed of 3000 rpm. Many of the blowers are placed inside the organ or very close to the organ case.

Only 31 organs had a working tremulant and 80% had traditional wind dumping units. The most common speed was approximately 4Hz, which is the frequency that humans perceive amplitude modulation to be most sensitive Zwicker [7].

3.6 Conclusion

The survey shows that over one third (39%) of the organ pipes surveyed had some form of flutter. The survey also indicates that pipe scale may have some influence and further research is needed to confirm this.

4 Pipe Organ Wind Instability

4.1 Preamble

This Chapter describes in detail the methodology and test apparatus necessary to determine the amount of organ pipe flutter on a test organ pipe. Some preliminary results and observations are also discussed.

Chapter 2 described briefly the evolution of the pipe organ wind system, and many organ builders select a system of winding that best supports their style of pipe regulation and voicing. Also important is the ability for the wind system to handle large variations in wind demand. Some choose traditional wedge, others may use single or double rise reservoirs with the possible inclusion of some form of concussion unit. For new organs, some may choose a more compact design employing wind-chest regulation using a Schwimmer arrangement.

Pipe organ wind systems have been considered, by some, to have some parallels with building ventilation systems. The main difference is that the pipe organ wind system terminates in free air at the pipe mouth, and the passage of air from the fan to the pipe is pressurised with many restrictions along its path. The air is pressurised by a weighted or sprung reservoir fitted with a variety of control valves, some more sensitive to pressure changes than others. The performance of a pipe organ wind system is notoriously difficult to predict. Identical wind systems do not always produce the same end results and often additional time is needed to make any necessary adjustments.

Research by Carlsson[8] concentrated on the dynamic behaviour of pipe organ wind, in particular how pressure fluctuations during playing, affect the pitch and speech of flue pipes.

The installation of a Turbulence Attenuator in the wind system immediately downstream of the blower was studied by Ngu[9] who also made changes to the blower impeller, increasing the number of blades from 12 to 24.

Most of the research to-date has examined the changes in organ pipe wind pressure for various playing conditions in the time domain. Research by Steenbrugge [10] into the operating regimes, voicing practices and other work in this area of research, assumes a pure air supply similar to that produced by an orchestral player. The pipe organ survey shows that this condition is not always achieved, and organ pipe flutter exists.

4.2 Methodology

4.2.1 Introduction

The pipe organ survey described in Chapter 3 confirms what organ builders and tuners have noticed for many years, that some organs have a slight unsteadiness in organ pipe tone that is particularly noticeable when tuning the organ pipes. The organ tuners call this unsteadiness "organ pipe flutter."

To investigate organ pipe flutter, it is necessary to have a wind system that is representative of a working pipe organ wind system and capable of producing both flutter and flutter-free pipework. Also, very important is that any measurements are taken under controlled conditions.

The key elements of a pipe organ wind system are shown in Figure 24 and consists of:

- Pipe organ blower
- Single rise reservoir
- Connecting pipe
- Test organ pipe
- Tremulant
- Instrumentation.

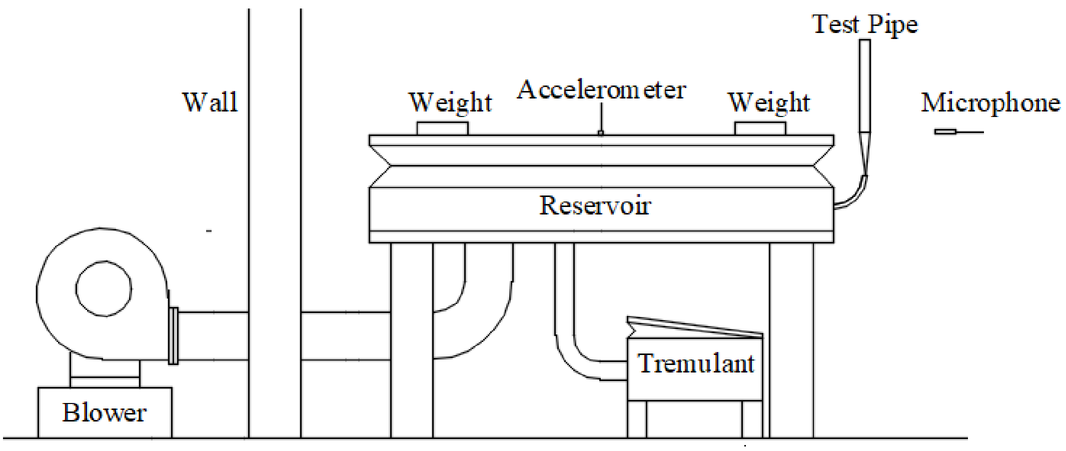


Figure 24 Test apparatus

4.2.2 Pipe organ blower

Air is supplied to the reservoir from an organ blower located in an adjoining room. The siting of the blower in a separate room allows the use of different blowers, some of which radiate more noise. The principal test blower, blower 6 (Chapters 8), shown in Photograph 3, is a modern 3000 rpm unit fitted with a .75kw electric motor controlled by a variable speed invertor. A 2m long 100mm diameter flexible plastic pipe connects the blower to a 100mm diameter plastic pipe connected to the reservoir.

Photograph 3 3000 rpm blower with flexible connecting pipe

4.2.3 Reservoir

The test reservoir, shown in Photograph 4, consists of a 1200mm x1200mm single rise reservoir with a working height of 400mm. The reservoir is fitted with a simple internal control valve connected to the reservoir top.

Normally a single rise reservoir would be sprung loaded. The use of a single rise test reservoir allows the use of weights or springs to pressurise the air inside the reservoir so that a comparison may be made between weighted and sprung loading.

The blower is connected to the reservoir by a 100mm plastic pipe with flexible rubber connecting joints.

Photograph 4 Single rise reservoir with measurement equipment

4.2.4 Internal control valve

The test reservoir is fitted with an internal control valve shown in Photograph 5.

Photograph 5 Internal control valve

4.2.5 Reservoir springs

Springs may be used as an alternative to weights, as a means of pressurising the air inside the reservoir and four springs are fitted, one at each corner. One of the springs is shown in Photograph 6. The pressure inside the reservoir can be varied by adjusting the spring tension using M8 threaded steel rods attached to each spring.

Photograph 6 Reservoir springs

4.2.6 Tremulant

The test apparatus is fitted with a traditional wind dumping tremulant which is designed to deliberately create an amplitude modulated wind supply. The tremulant shown in Photograph 7 is typical of many fitted to Victorian and Edwardian pipe organs. The tremulant unit is controlled by a low voltage electronic controller that allows adjustment of the frequency and depth. Further information about the tremulant is contained in appendix D

Photograph 7 Wind dumping tremulant

4.2.7 The test organ pipe

The test organ pipe is arranged to stand vertically from one side of the reservoir using a small length of 12mm copper tube bent at a right-angle. The pipe is a six-inch treble c pipe, note 49, taken from an 8ft. great open diapason rank of pipes. The pipe is made from spotted metal and is fitted with a traditional tinplate tuning slide. A Photograph of the test pipe is shown in Photograph 8.

Pipe details.

Pipe diameter	22.5mm (scale)
Mouth wide	15.7mm
Windway	0.1mm
Cut-up	4.6mm
Toe hole diameter	6.4mm

Photograph 8 The test organ pipe

4.2.8 Instrumentation

4.2.8.1 Pressure measurement

The pressure in the reservoir and the connecting pipe from the blower to the reservoir is measured using two electronic pressure transducers connected to a data acquisition unit. The data acquisition unit is connected to the USB port of a laptop computer running Omega Daqview software. The pressure instrumentation is calibrated to measure the wind pressure in Pascals and mm water gauge.

Data acquisition unit Omega OMB DAQ -55 USB data acquisition unit.

Pressure transducer 1 Omega PX278-30DSV differential pressure transducer.

Pressure transducer 2 Omega PX278-30DSV differential pressure transducer.

4.2.8.2 Noise and vibration measurement

The sounding test organ pipe and reservoir top vibration are measured using a two-channel Class 1 pc-based noise and vibration measurement system running 01dB dBFA32 Symphonie software. The interface between the computer and the microphone or accelerometer is by a 01dB Symphonie data acquisition unit which connects to the laptop computer PCMCIA port. The unit measures from dc to a maximum frequency of 20kHz.

01dB Symphonie unit SN 01009

Both measurement systems are shown in Photograph 9.

Photograph 9 Measurement equipment

4.2.8.3 Organ pipe high frequency microphone measurement

Channel 1 connects to a class 1 measurement microphone, shown in Photograph 10, placed 150mm from the pipe mouth. Alternatively, the microphone can be placed inside the reservoir.

Microphone	B&K 2669 Microphone pre-amp	SN 1865497
	01 dB Microphone MCE 212	SN 17989
Microphone Calibrator	Cirrus CR511E class 1L	SN 038876

Photograph 10 Measurement microphone

4.2.8.4 Reservoir top low frequency vibration accelerometer measurements

Channel 2 connects to an icp accelerometer, shown in Photograph 11, mounted centrally on the reservoir top. All reservoir vibrations are acceleration re (dB 1.000e06m/s^2PWR)

Accelerometer 1	B&K 4507 -004	SN 32102
Accelerometer 2	MTN 1800 -004	SN 358848

Photograph 11 Accelerometer placed centrally on reservoir top

4.2.9 Other Applications

The 01dB software provides a series of windowing and overlap options together with automatic anti-aliasing filters. The system also provides a comprehensive set of analysis tools but in some circumstances, other applications can provide a more convenient platform. Measurement data can be exported as a text or WAV file for importing into MATLAB, Audacity or Adobe Audition for post processing and analysis. The Adobe Audition software is particularly convenient for inspecting frequency spectra.

4.3 Preliminary results

4.3.1 Organ pipe microphone results

4.3.1.1 Time domain

The fundamental frequency of the test organ pipe is just over 1kHz and generates both even and odd harmonics.

Using the microphone, connected to channel 1, a 10 second period was recorded using the 01dB dBFA software. The signal of the sounding test pipe in the time domain is shown in Figure 25 and it is significant that it gives little information about the level and character of any organ pipe flutter or modulation of the sounding pipe.

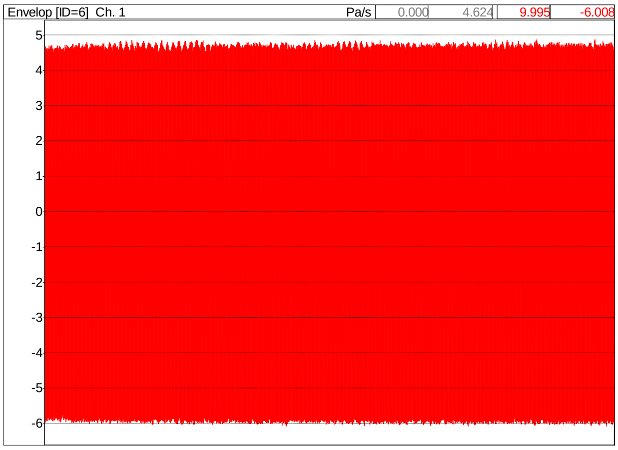


Figure 25 Time domain of the test organ pipe

4.3.1.2 Frequency domain.

Alternatively, the signal can be analysed in the frequency domain and the frequency spectrum, together with harmonics up to 20kHz are shown in Figure 26.

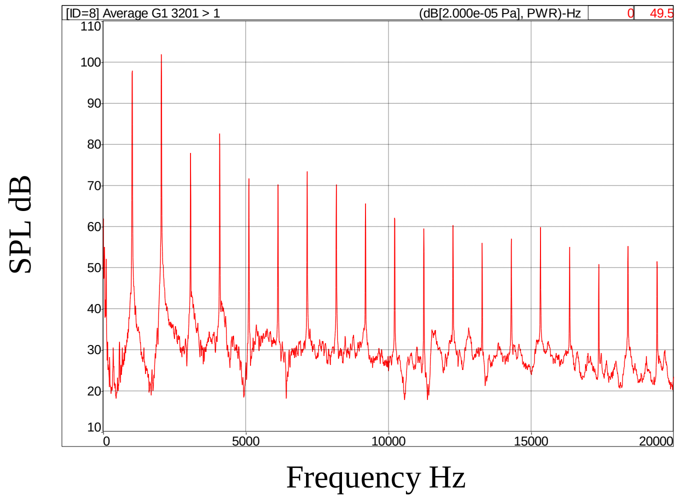
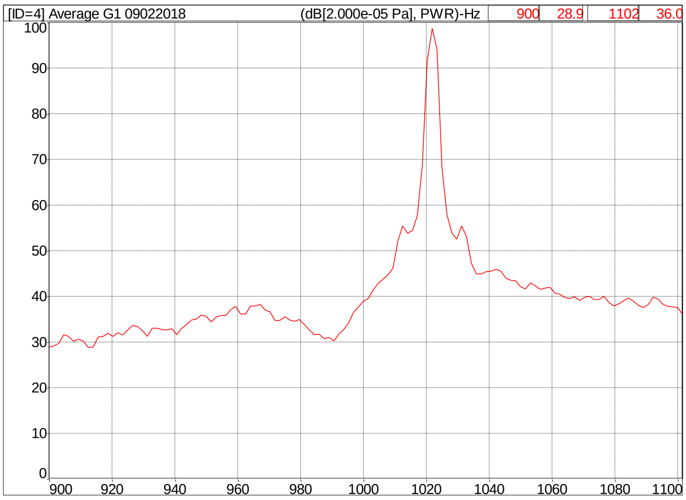


Figure 26 Frequency spectrum of the test organ pipe

4.3.2 Frequency resolution

The 01dB software allows the frequency resolution to be changed from 401 to 3201 lines of FFT. It is also possible to zoom in and examine in more detail a section of the frequency spectrum. Figures 27 & 28 show a zoomed portion of the frequency spectrum, centred on the fundamental frequency of 1020Hz, using 801 and 3201 lines of FFT. The frequency span is 190Hz (910 to 1100Hz) with corresponding frequency resolutions of 0.24Hz & 0.06Hz. This means that 3201 lines of FFT must be used to see, in detail, the presence of any side-bands on each side of the fundamental frequency.

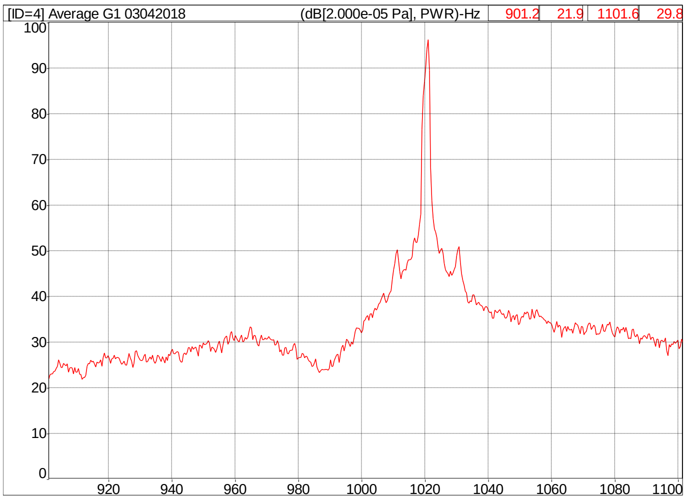
SPL dB



Frequency Hz

Figure 27 Test pipe fundamental 801 lines FFT

SPL dB



Frequency Hz

Figure 28 Test pipe fundamental 3201 lines FFT

Figure 28 shows two side-bands at approximately 10Hz on each side of the fundamental frequency. Figure 29 also shows a strong peak at 10Hz which, is the same frequency as the two side-bands.

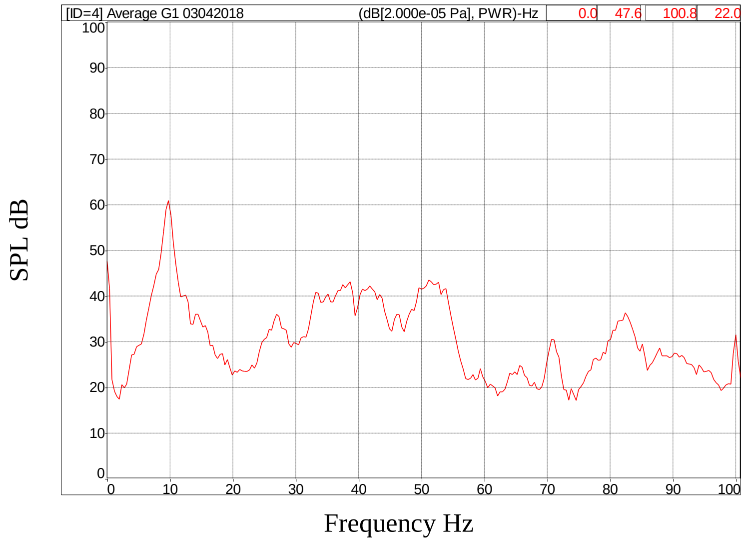


Figure 29 Test pipe low frequency spectrum 3201 lines FFT

4.3.3 Reservoir top accelerometer vibration results

The corresponding reservoir top vibration levels for frequencies below 100Hz are shown in Figure 30 for 3201 lines FFT. The maximum level occurs at a frequency of 10 & 39Hz.

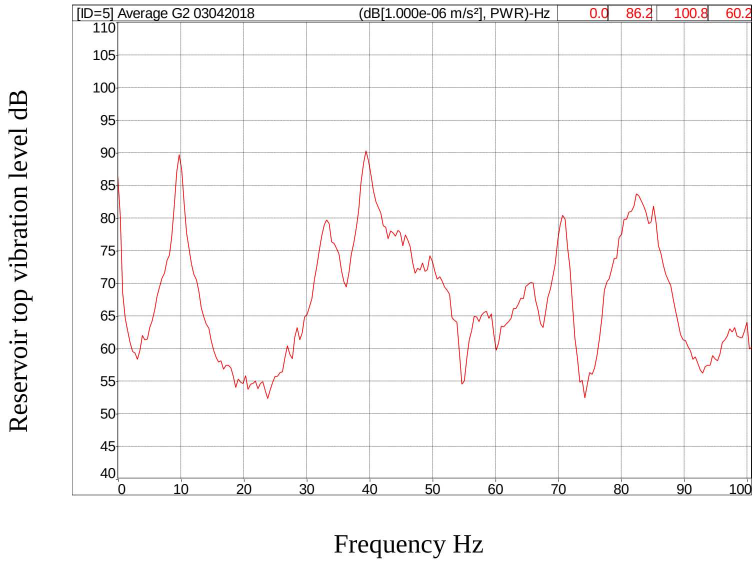


Figure 30 Reservoir top frequency spectrum 3201 lines FFT

4.4 Background noise level results

It is important that any signal must not be compromised by any background noise. Figure 31 shows the frequency spectrum of the test organ pipe (blue) and the background noise with the blower switched off (red), and on (green).

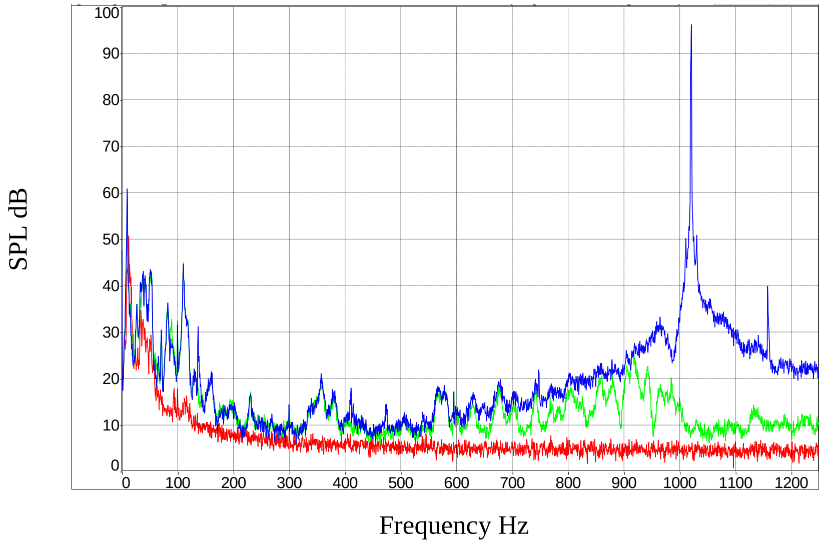


Figure 31 Frequency spectrum of sounding pipe and background noise 3201 lines FFT

The corresponding reservoir top vibration levels, using the accelerometer connected to channel 2, are shown in Figure 32 for the sounding pipe (blue), background noise blower off (red), and blower on (green).

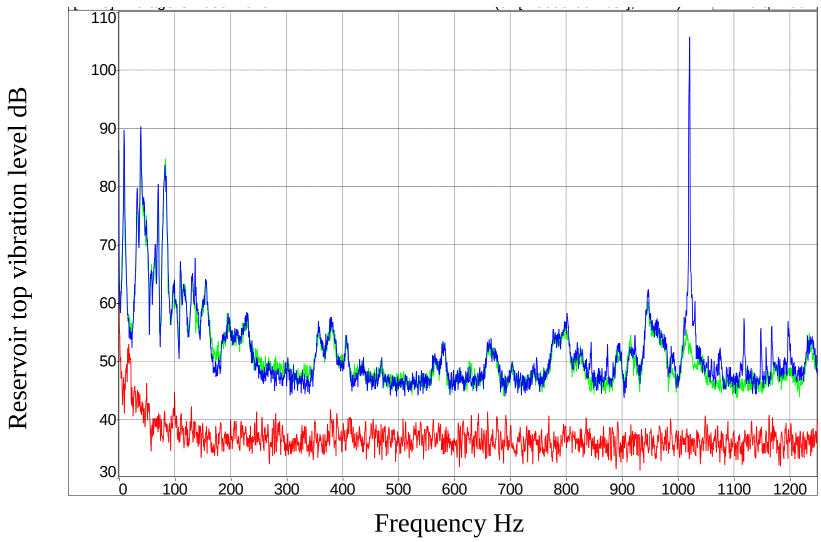


Figure 32 Reservoir top vibration levels for the sounding test pipe and background noise 3201 lines FFT

4.5 Conclusions

Figures 27 & 28 show that 3201 lines of FFT must be used in all measurements.

Figures 31 & 32 show that the background levels are well below the signal levels for both the microphone and accelerometer.

The correlation between the side-band frequency (10Hz) using the organ pipe microphone and the 10Hz low frequency microphone level, together with the 10Hz frequency of reservoir vibration using the accelerometer is good. This indicates that the low frequency vibration of the reservoir is the cause of the flutter and amplitude modulation that we can hear on the sounding organ pipe.

5 Modulation

5.1 Preamble

The sounds experienced in everyday living are constantly changing in frequency and amplitude giving rise to frequency and amplitude modulation. Figures 33 and 35 show frequency and amplitude modulated waves consisting of a carrier wave onto which another signal is super-imposed.

Sustained sounds generated by a musical instrument or singer often contain small frequency fluctuations. These may be naturally occurring or deliberately engineered, and the vibrato is often used by musicians to enhance their own musical performance.

In the laboratory, the perceptions of such variations are studied using either frequency or amplitude modulated sine waves, and a direct comparison between the sensations produced by frequency modulation and amplitude modulation was conducted by Zwicker [11] and Moore [12]

5.2 Frequency Modulation

In a frequency modulated signal the carrier's frequency is varied in proportion to the modulating signal's magnitude, but the amplitude of the signal does not vary. This relationship is shown in equation 1 and Figure 33.

$$\text{Frequency modulation} = \sin(2.\pi.f_c.t - \beta.\cos 2.\pi.g.t) \quad (1)$$

Where f_c = carrier frequency. g = modulating frequency. β = frequency modulation index

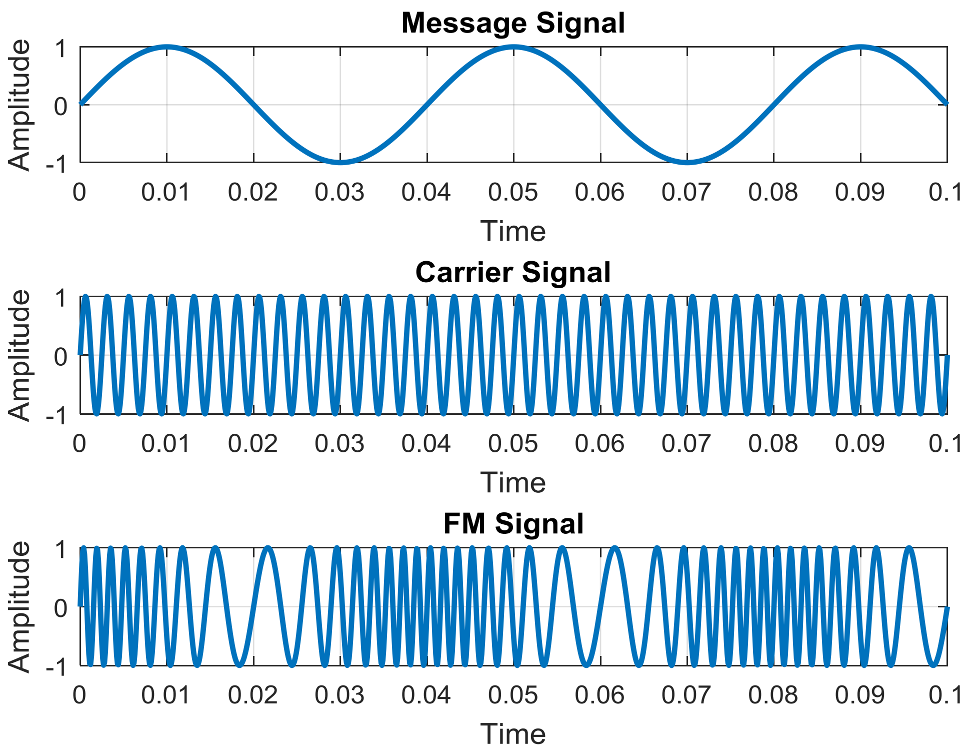


Figure 33 Frequency modulation

Vibrato is often used by musicians to enhance their musical performance and was studied by Seashore [13]. Prame [14] measured the vibrato rate of ten singers and found that the average vibrato rate was 6 Hz. Frequency modulation was also studied by Rossing et al [15] Moore [16] Backus [17] and Deutsch [18].

The just noticeable frequency modulation as a function of modulation frequency for a 1kHz centre frequency is shown in Figure 34. This shows that the most sensitive frequency modulation occurs at a frequency of approximately 4Hz Zwicker [7]

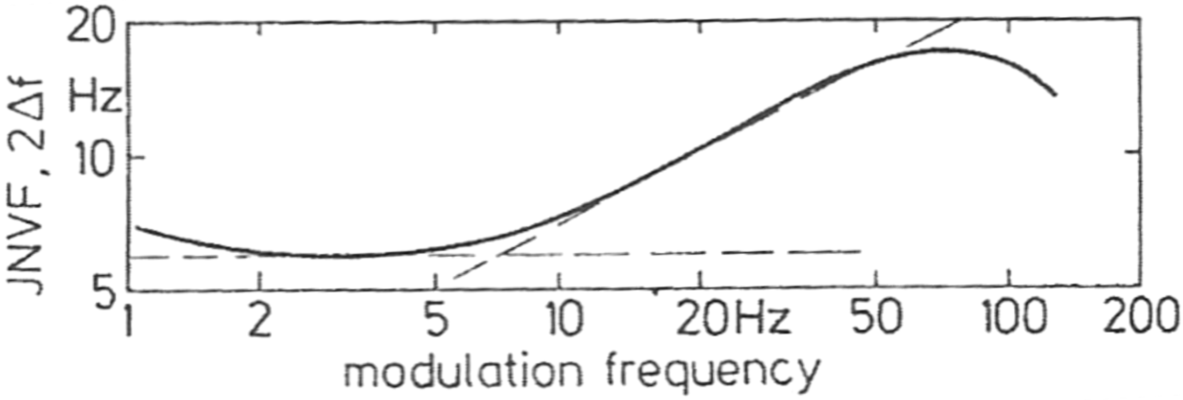


Figure 34 The just noticeable frequency modulation as a function of modulation frequency for a 1kHz centre frequency (After Zwicker [7])

Early sound films had problems with sound quality due to the slight variation in the speed of the film transport mechanism. To overcome this, a standard was introduced by the Society of Motion Picture Engineers [19] .This included a series of test films, specifically made to test the level of frequency modulation, so that the mechanism could be adjusted to produce an acceptable level of sound clarity.

Reel to reel tape recorders also suffer from the same problem. To overcome this, sound engineers used recording speeds of 7.5 or 15 inches per second. The unwanted frequency modulations, wow (0.5 to 6Hz) and flutter (6 to 100Hz), can now be removed using DSP techniques Maziewski [20] Czyzewski et al [21].

5.3 Amplitude modulation

In an amplitude modulated signal the carrier's amplitude is varied to follow the magnitude of the modulating signal keeping the carrier frequency constant. This relationship is shown in equation 2 and Figure 35.

$$\text{Amplitude modulation} = (1+m.\sin 2.\pi.g.t).\sin(2.\pi.f_c.t) \quad (2)$$

Where f_c = carrier frequency. g = modulating frequency. m = amplitude modulation index

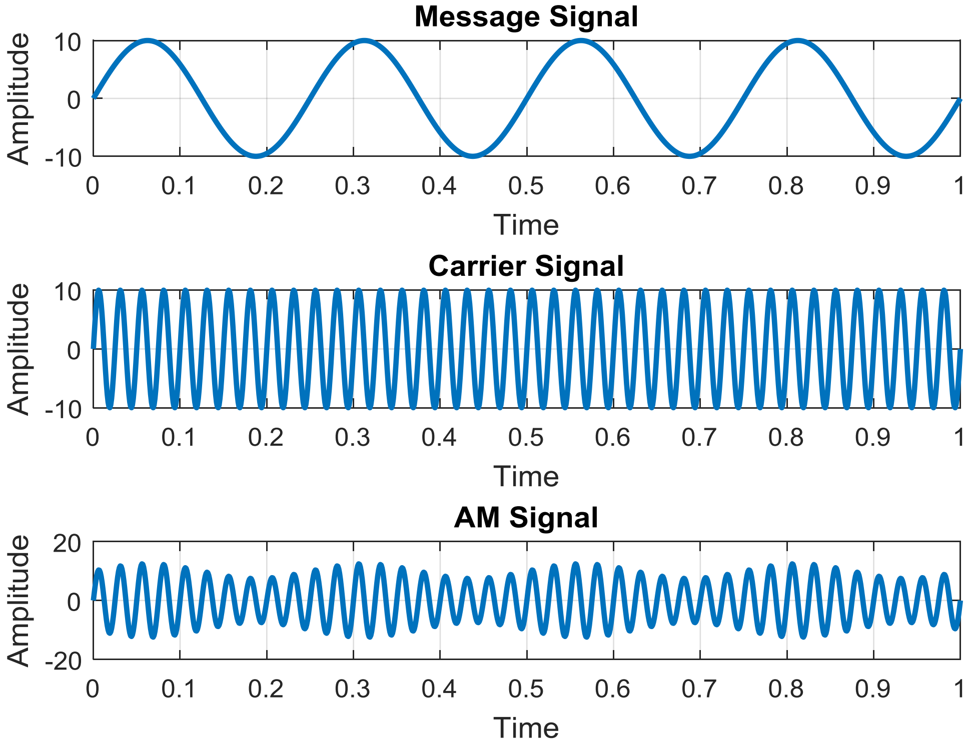


Figure 35 Amplitude modulation

The tremolo is the amplitude modulation version of the vibrato, Olson [22], and is designed to gently move the reservoir top and create a mildly undulating tone. The tremolo or tremulant pipe organ stop was first used in the 16th century and is specially designed to impart strong amplitude modulation, typically around 4Hz. More information regarding the tremulant stop is contained in Appendix D.

The effects of amplitude modulation are well known, and soldiers are often asked to "break step" when crossing a bridge. Also, the collapse of the Tacoma Narrows bridge, caused by wind induced vibration, had a significant influence on modern bridge design.

The relationship between the just noticeable degree of amplitude modulation as a function of the modulating frequency for a 1kHz tone at 40dB & 80dB, and white noise (dotted line), is shown in Figure 36, Zwicker [7]. It shows that the ear is most sensitive to a modulating frequency of approximately 4Hz.

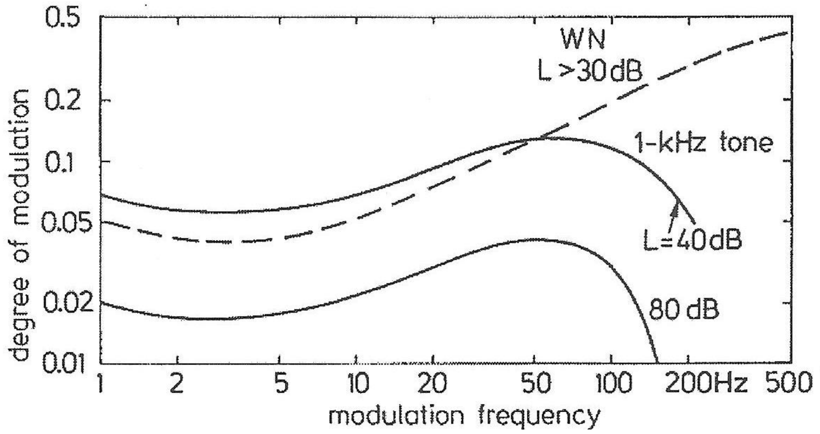


Figure 36 The relationship between the just noticeable degree of amplitude modulation as a function of the modulating frequency for a 1kHz tone at 40dB & 80dB and white noise dotted line (After Zwicker [7])

The preliminary findings detailed in Chapter 4, show that the organ pipe flutter results from the reservoir vibrating at a frequency of approximately 10Hz. From Figure 36, the sensitive to low frequency amplitude modulation at this frequency is approximately 2% for a level of 80dB or 7.5% for a level of 40dB.

Figure 26 (Chapter 4) shows the frequency spectrum of the sounding test pipe, at a distance of 150mm from the pipe mouth, with a level of 98dB for the first harmonic (1kHz) and 101dB for the second harmonic (2kHz). These levels represent the levels that would be experienced by the tuner at the soundboard. Generally, the output level of the pipework is regulated to produce the desired sound level in the body of the church. Dependent on the position of the organ and the acoustics of the church this may mean that the pipework needs to be driven hard with very high levels at the soundboard. Not all stops respond to this treatment and this is where the scale of the stop is of importance. Figure 13 (Chapter 3) shows that the amount of flutter detected in the body of the church is less than that at the soundboard for each of the four organ pipes.

5.4 Amplitude Modulation index

The amplitude modulation index m = Vm/Vc (3)

Where Vm = the modulation voltage and Vc the carrier voltage

Percentage of modulation = modulation index x 100

Alternatively, the level of the carrier and modulation frequency may be expressed in terms of Pascals.

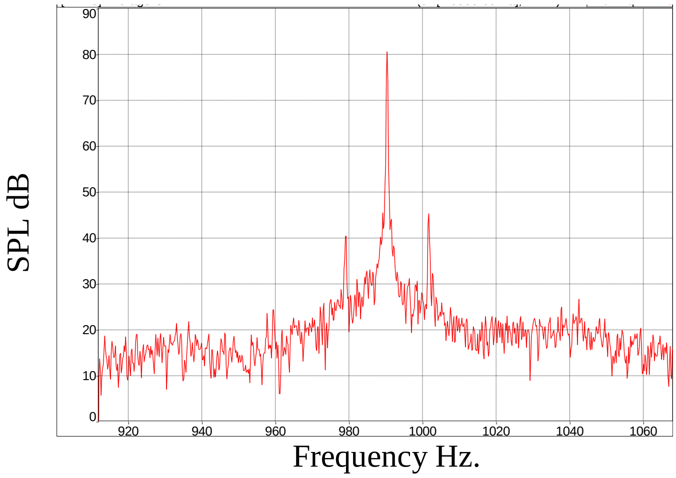


Figure 37 Typical fundamental frequency spectrum showing amplitude modulation side-bands

Figure 37 shows the spectrum of the fundamental frequency for a reservoir pressure of 60 mm WG. The level of the fundamental is 80.6dB and the level of the lower and upper side-bands are 40.4 and 45.3dB respectively.

SPL = 20 Log10(P1/Pref). P1 = pressure in Pa. Pref = reference pressure = 0.00002

P1 pressure = 0.00002 x $10^{(SPL/20)}$

80.6dB = 0.2140Pa. 40.4dB = 0.0021Pa. 45.3dB = 0.0037Pa.

The modulation index of the lower side-band = 0.0021/0.214 = 0.0098 or 0.98%

the upper side-band = 0.0037/0.214 = 0.0173 or 1.73%

Alternatively, the modulation index can be calculated using the difference between two SPL's from Table 11.

As an example, from Figure 37, Lower side-band diff = (80.6 - 40.4) = 40.2dB

From Table 11 % modulation = 1.00 %.

SPL difference (dB)	% modulation
55	0.18
54	0.20
53	0.22
52	0.25
51	0.28
50	0.32
49	0.35
48	0.40
47	0.45
46	0.50
45	0.56
44	0.63
43	0.71
42	0.79
41	0.89
40	1.00
39	1.12
38	1.26
37	1.41
36	1.58
35	1.78
34	2.00
33	2.24
32	2.51
31	2.82
30	3.16
29	3.55
28	3.98
27	4.47
26	5.01
25	5.62
24	6.31
23	7.08
22	7.94
21	8.91
20	10.00
19	11.22
18	12.59
17	14.13
16	15.85
15	17.78

Table 11 Percentage amplitude modulation Lookup Table

5.5 Amplitude modulation simulation

To understand the effect of amplitude modulation on a signal, a MATLAB script has been developed (am1dbtest) to display a defined modulated signal in both the time and frequency domain. Several carrier frequencies from 62 to 8000Hz can be selected, which correspond to the various stop pitches starting at the 8ft. In addition, modulating frequencies can be selected that are relevant to those experienced in an organ reservoir. The 10 to 20Hz is the typical modulation frequency produced by reservoir top vibration, 10Hz for weighted reservoirs and 20Hz for sprung reservoirs. The level of modulation is adjustable from a very low level of 1 percent to a level of 10 percent. This range can then be increased tenfold to give a maximum modulation of 100 percent.

The modulating frequencies below 10Hz are used for simulating the modulation produced by the tremulant stop.

Carrier frequency. 62,125,250,500,1000,2000,4000,8000Hz.

Modulation Frequency. 2,3,4,5,6,7,8,9,10,12.5,15,17.5,20,25,30Hz.

Percentage Modulation. .01,.02.03,.04,.05,.06,.07,.08,.09,.1

Modulation multiplier 1 or 10

Figure 38 shows a carrier frequency of 1kHz, the same as the pipe fundamental frequency, and a modulation frequency of 10Hz, the same as the reservoir top frequency. A 1% level of amplitude modulation produces two distinct visible side-bands, that are not audible at 10Hz on each side of the carrier frequency of 1kHz. A 2% level of amplitude modulation is shown in Figure 39 and is just audible.

Figure 40 shows a modulating frequency of 4Hz with 10% modulation which is typical of a tremulant stop.

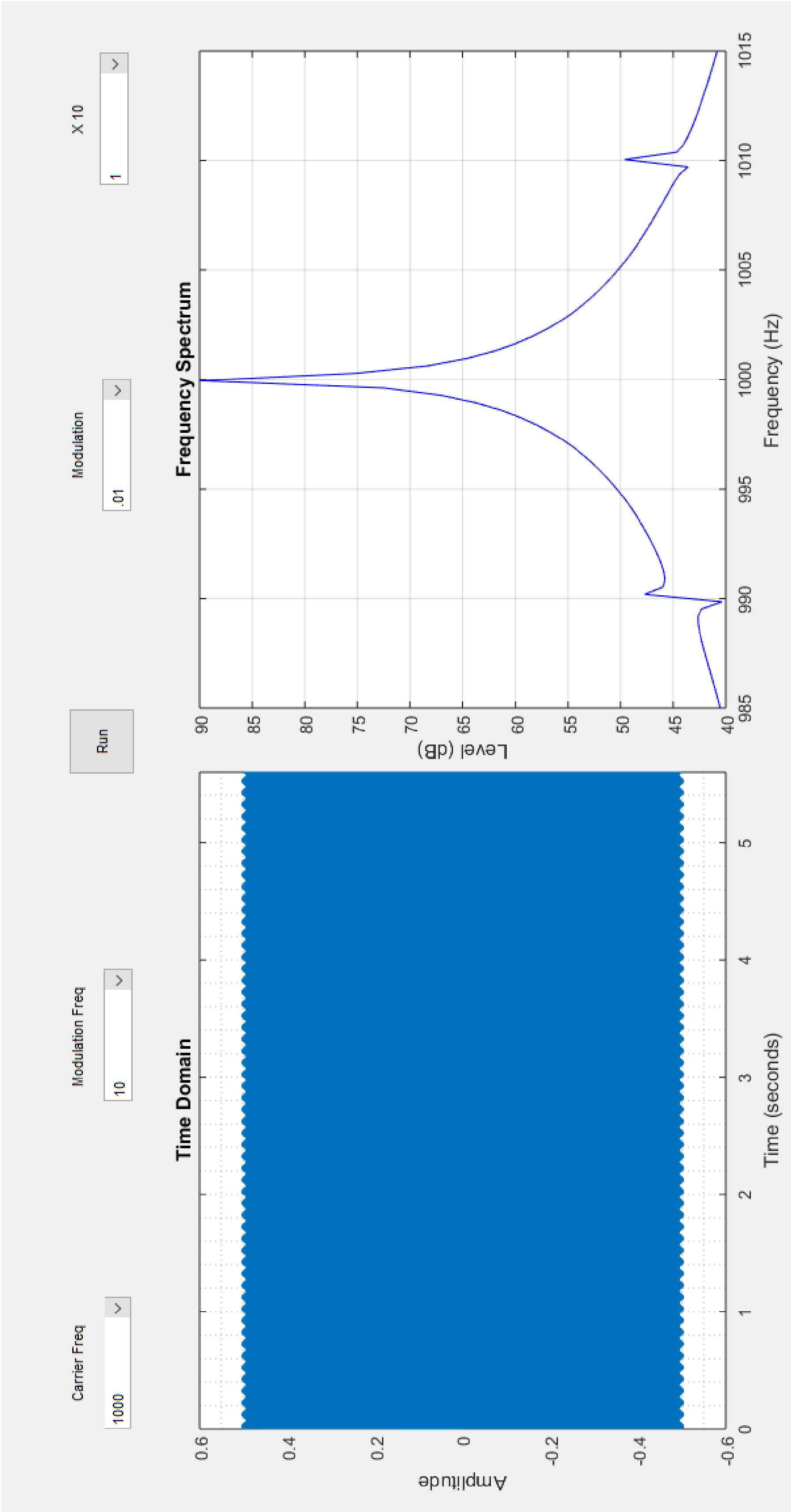


Figure 38 1% amplitude modulation

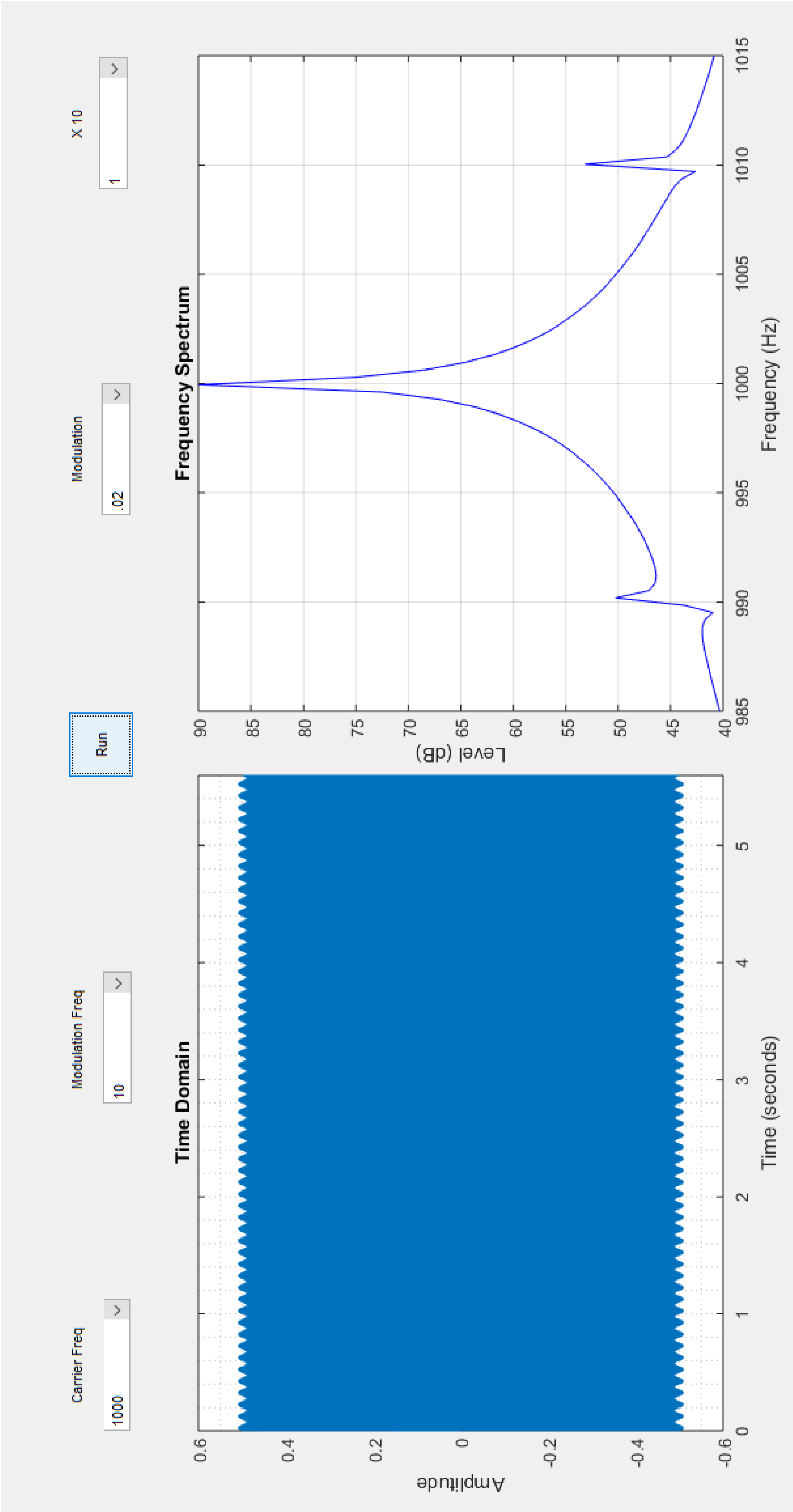


Figure 39 2% amplitude modulation

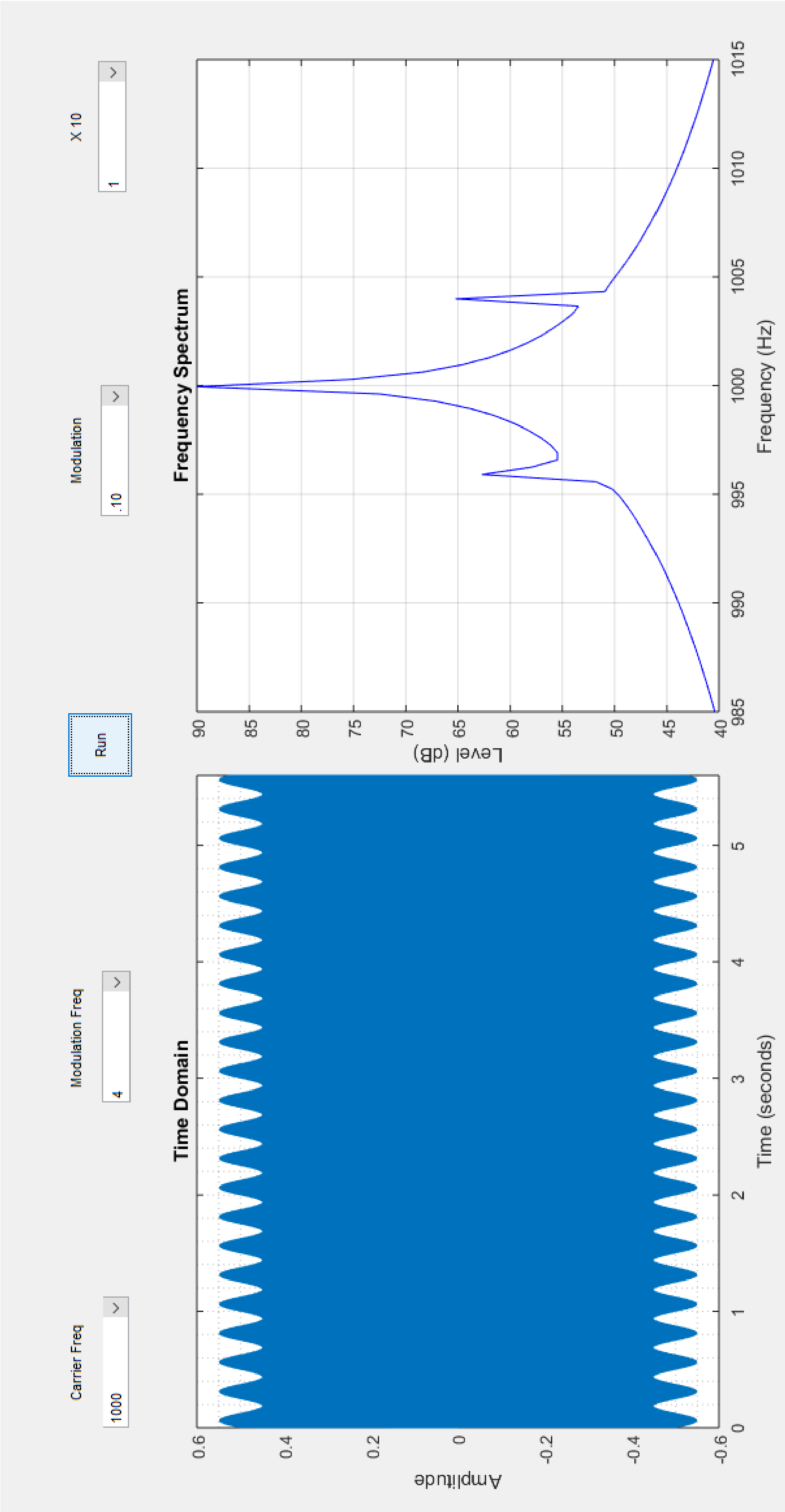


Figure 40 10% amplitude modulation

5.6 Side-band modulation

Condition monitoring of machinery often uses side-band modulation analysis [23]. Small defects in the rotational components of machines, such as gear and bearing wear, can be detected using an accelerometer and specially designed software that analyses the rotating components and produces a frequency spectrum. Any faults, such as cracked or broken teeth, appear on the frequency spectrum as a series of side-bands equally spaced on each side of the rotational speed of the gear train.

Figure 41 shows a typical amplitude modulated signal and frequency spectrum with high amplitude side-bands around the tooth-meshing frequencies, for a gearbox with several defective teeth. This is very similar to the frequency spectrum of a sounding organ pipe with its many harmonics, each with side-bands at the reservoir resonant frequency. The organ pipe side-bands have very high energy levels, and so only one principal side-band is observed.

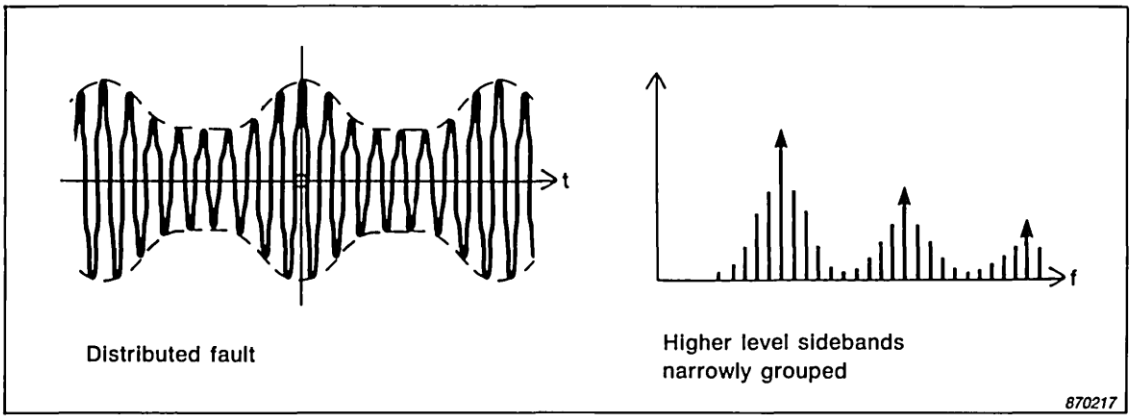


Figure 41 Amplitude modulated and frequency components for a defective gearbox (After Angelo [23])

5.7 Alternative methods to determine % amplitude modulation

Several methods can be employed to determine the percentage amplitude modulation of the sounding treble c open diapason test pipe.

5.7.1 Method 1

The 01dB measurement platform is a calibrated system. Calibration has been achieved using a 94dB microphone calibrator. Using the 01dB post analysis software, the frequency spectrum of the sounding pipe can be selected as shown in Figure 42. The Figure shows the selection of the 5th harmonic of 5kHz. The zoom feature allows the two side-bands to be clearly seen and measured. Figure 43 shows the zoom spectrum for the 5th harmonic with a level of 70.4dB with lower and upper side-band levels of 43.2 and 45.3dB respectively. This gives a difference of 27.2 and 25.1dB. From Table 11 the corresponding percentage modulation is approximately 4.5% and 5.5%.

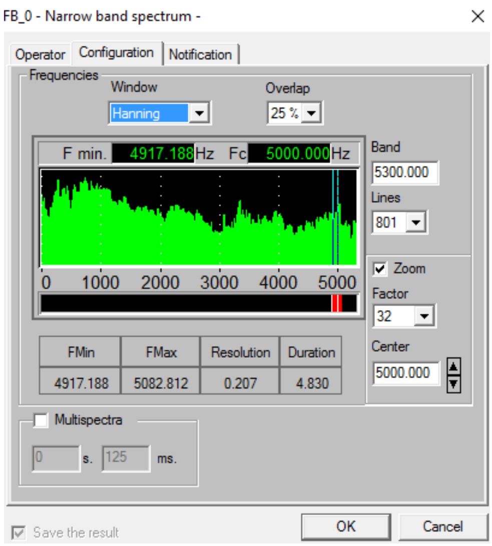


Figure 42 01dB measurement window

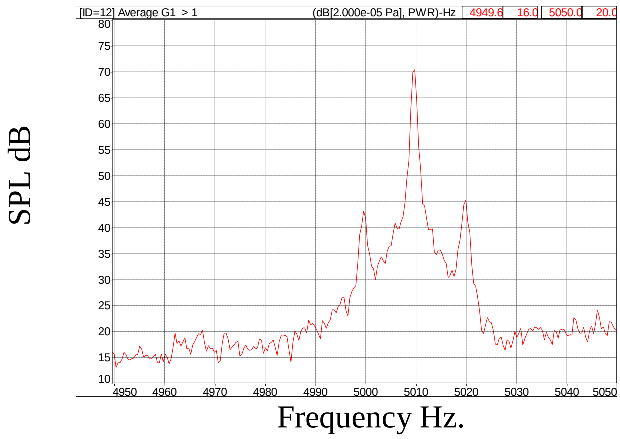


Figure 43 01dB 5th harmonic frequency spectrum

5.7.2 Method 2

The signal can be exported from 01dB as a WAV or text file. Using MATLAB the 01dB WAV file, or any other recorded WAV file, can be imported and the time domain, general frequency spectrum, together with a zoom spectrum centred on each harmonic can be displayed. The three plots are shown in Figures 44, 45 and 46.

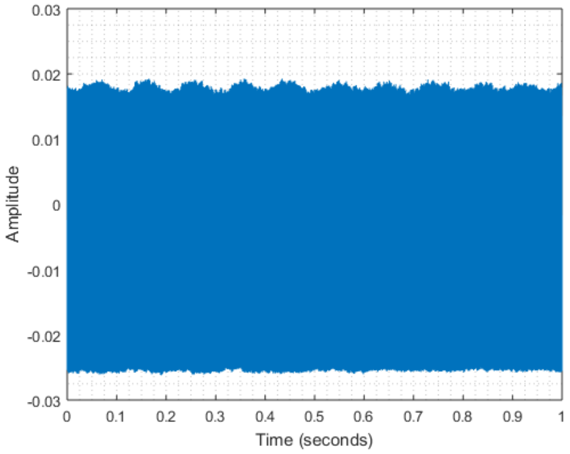


Figure 44 Time domain

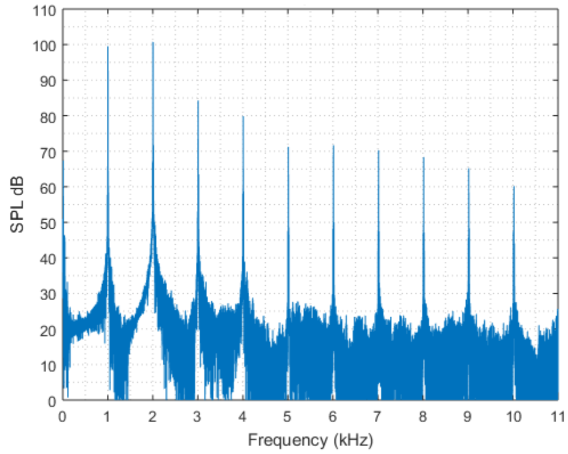


Figure 45 Frequency domain

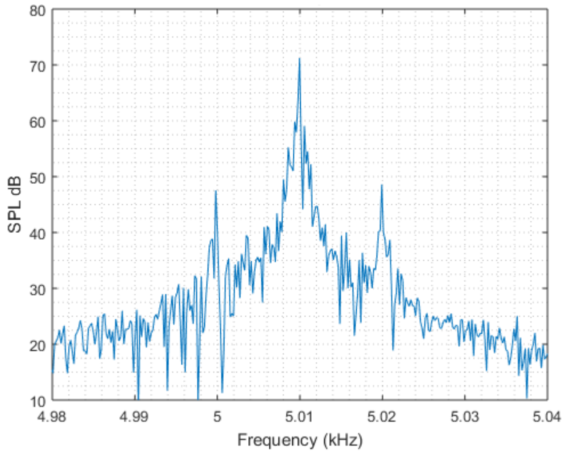


Figure 46 5th harmonic frequency spectrum

5.7.3 Method 3

Similar to method 2 a recorded WAV file can be imported into Adobe Audition and the frequency spectrum tool used. Figures 47, 48 & 49 show the time domain, frequency spectrum, and the 5th harmonic frequency spectrum showing the fundamental and side-bands levels.

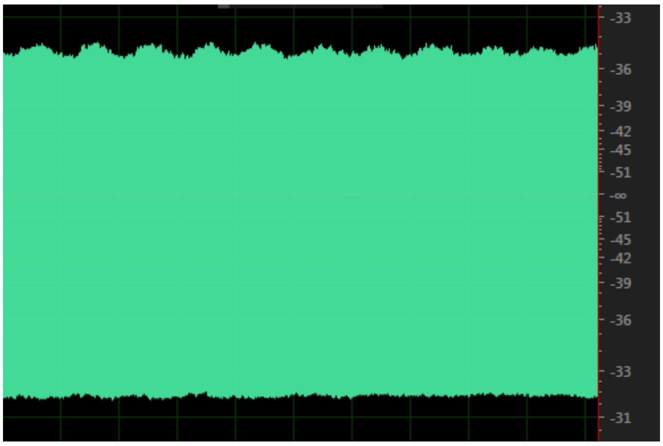


Figure 47 Adobe Audition time domain

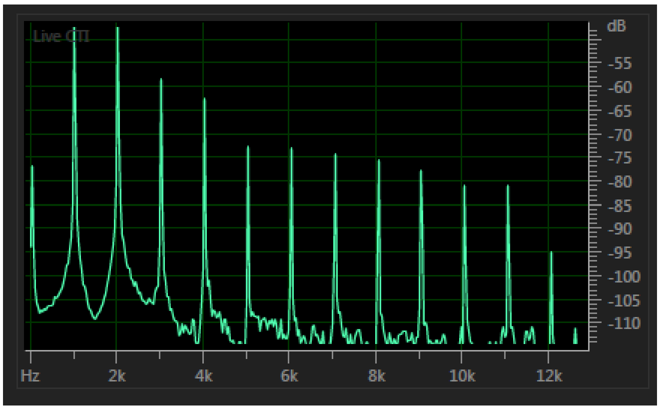


Figure 48 Adobe Audition frequency spectrum

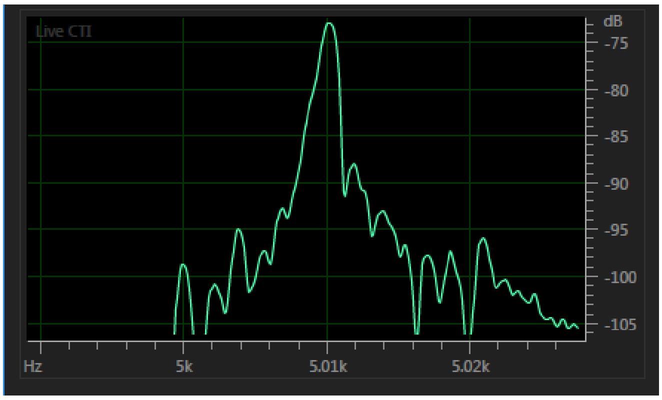


Figure 49 Adobe Audition 5th harmonic frequency spectrum.

5.8 Roughness

At very low modulation frequencies, the loudness changes slowly and reaches a maximum at approximately 4Hz and then starts to decrease, as shown in Figure 36.

At approximately 15Hz a second type of sensation starts to take over and reaches a maximum at a modulation frequency of approximately 70Hz and then decreases as the modulation frequency increases. This is shown in Figure 50 Zwicker [7]. This second type of sensation is termed “Roughness” and can be considered to cover the amplitude modulation frequency range of 15 to 300Hz.

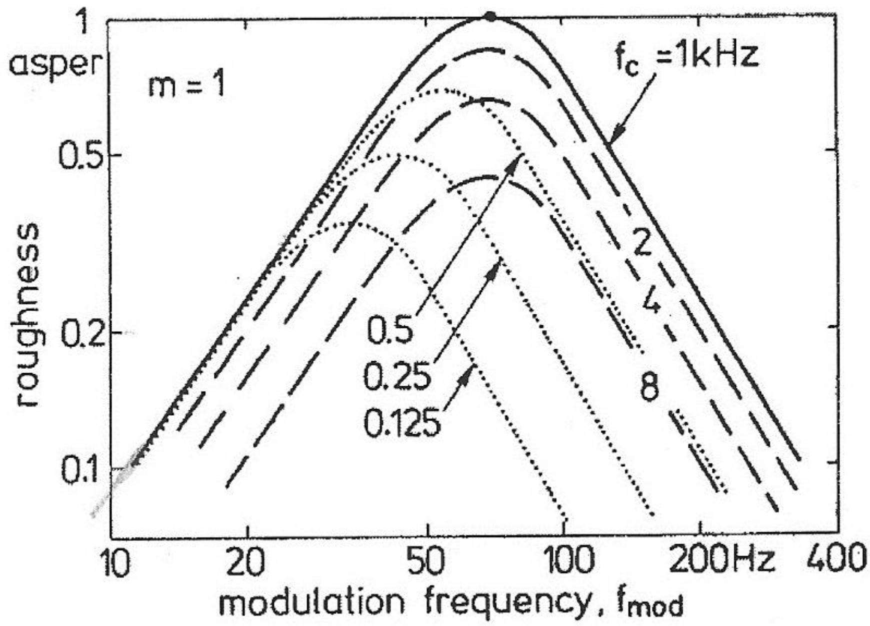


Figure 50 Roughness of 100% amplitude modulated tones for the given centre frequencies as a function of the frequency of modulation (After Zwicker [7])

It will be seen in Chapter 6 the effects of roughness are more relevant to sprung reservoir systems that have resonant frequencies at approximately 20Hz. However, with weighted reservoir systems, several harmonic frequencies below 100Hz are also present and may be at a sufficient level to modify the amplitude modulation of the sounding pipe.

5.9 Discussion

Most modulation research uses a single 1kHz carrier frequency which is far from the practical reality of a sounding organ pipe which may have over 20 harmonics. Also, sound pressure levels of approximately 100dB are experienced by the tuner at the soundboard.

To understand the effect of amplitude modulation, various single carrier frequencies, modulation frequencies, percentage modulation and modulation indexes were simulated. The presence and development of side-bands giving a good indication of the modulation frequency and level.

Modulation is used to good effect by some musicians to enhance their musical performance. However, in this organ building application the presence of any modulation on the sounding organ pipe is detrimental. The exception is the tremulant stop which is designed to impart extreme amplitude modulation on the sounding organ pipe.

Several methods are presented in this Chapter but only method 3 is available to organ builders.

The acoustic condition of “Roughness” is more relevant to sprung reservoirs which are examined in the next Chapter.

5.10 Conclusions

The use of post analysis software is a good starting point and will give a good indication of the extent and level of any organ pipe flutter.

Having established that the amplitude modulation of the sounding organ pipe is caused by reservoir vibrations, the use of an accelerometer fixed to the reservoir top is a convenient method to measure reservoir top vibration levels.

6 Blower and reservoir combinations

6.1 Preamble

Chapter 4 examined an electrically blown, weighted, single rise reservoir wind system with a 3000 rpm blower. This Chapter examines other possible electric blower speeds and reservoir combinations. The pipe organ survey shows that some pipe organs, Figure 17, have single rise reservoirs that are normally pressurised using springs. In addition to a weighted reservoir and 3000 rpm blower, other basic combinations are possible:

- 1500 rpm weighted reservoir
- 1500 rpm sprung reservoir
- 3000 rpm weighted reservoir
- 3000 rpm sprung reservoir.

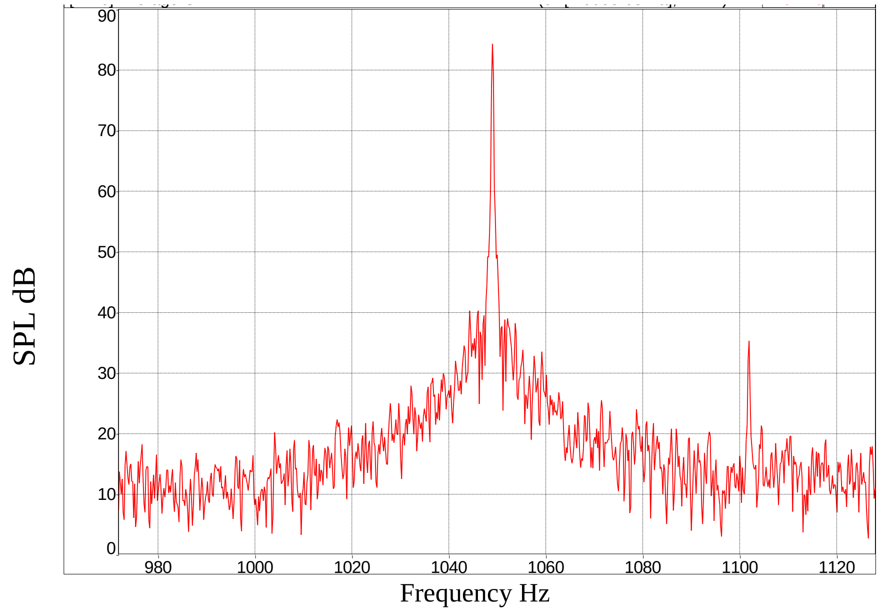
6.2 Methodology

Using the apparatus and measuring equipment described in Chapter 4, the 4 basic blower and reservoir combinations are examined. The 1500 rpm results use blower 1 and the 3000 rpm results use blower 6, detailed in Chapter 8.

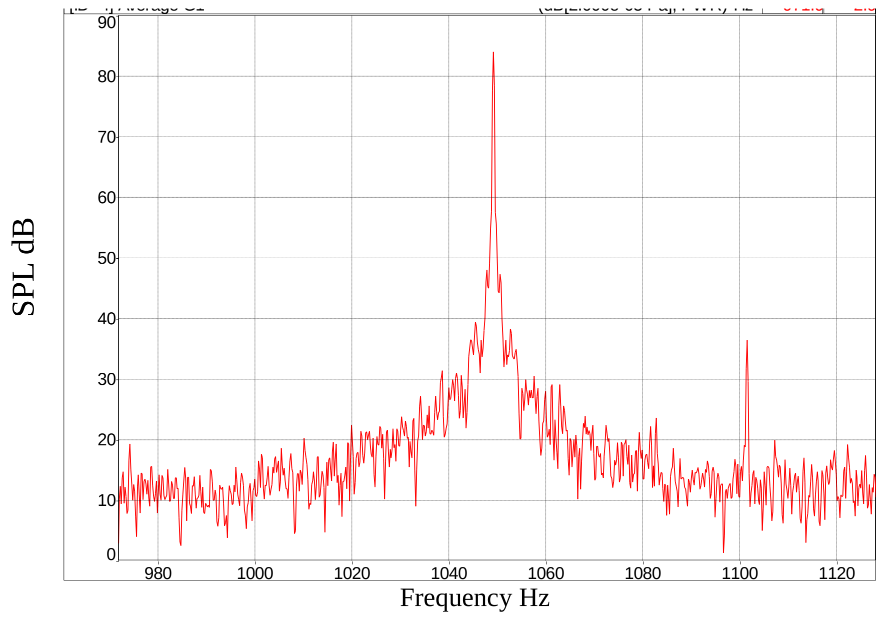
6.3 Results

6.3.1 Organ pipe microphone results

The results for the test organ pipe frequency spectra for the 4 combinations are shown in are shown in Figures 51 & 52.

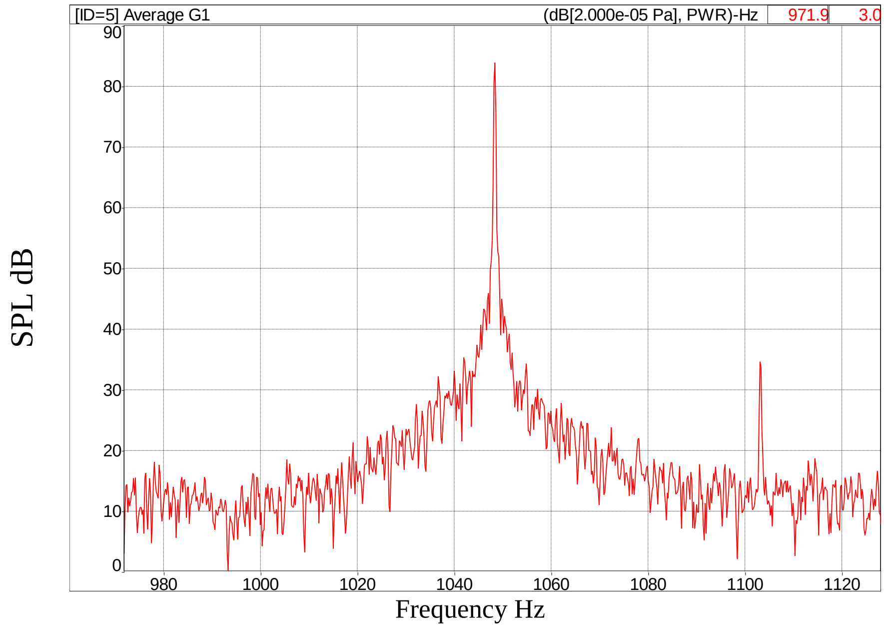


A 1500 rpm sprung

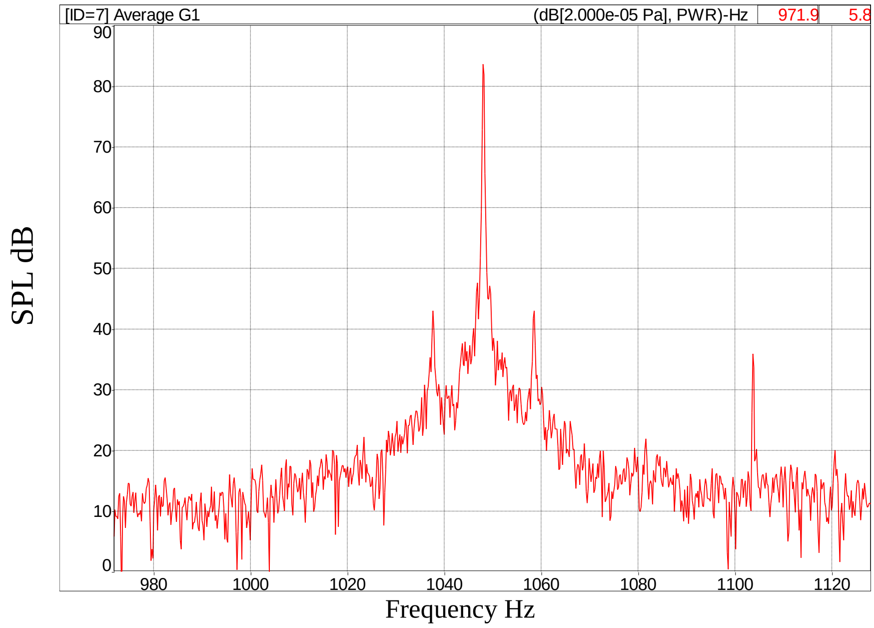


B 3000 rpm sprung

Figure 51 1500 & 3000 rpm sprung test pipe spectra



A 1500 rpm weighted

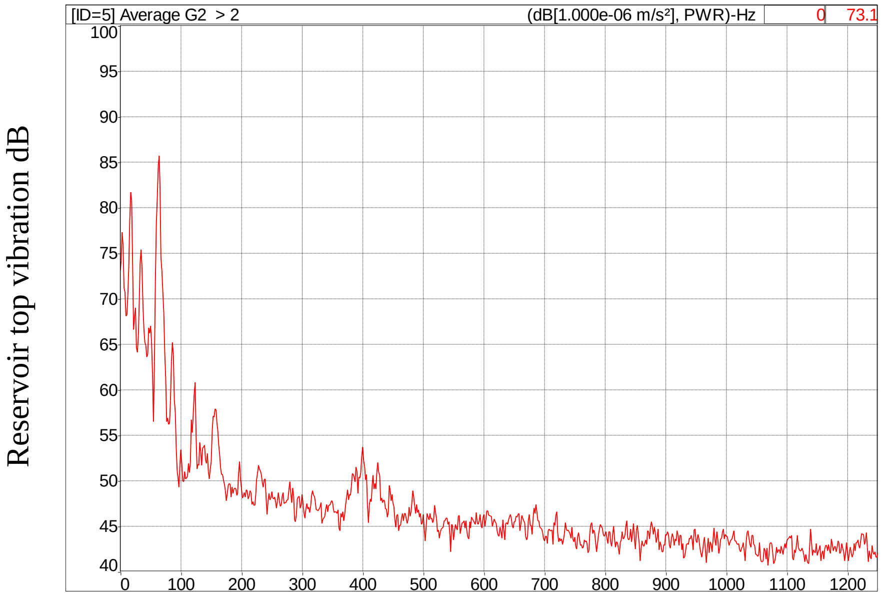


B 3000 rpm weighted

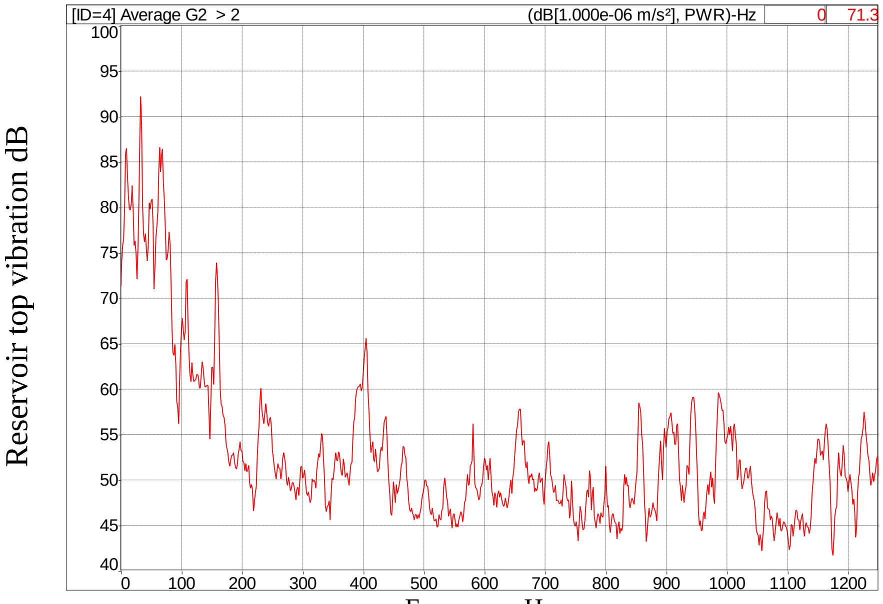
Figure 52 1500 & 3000 rpm weighted test pipe spectra

6.3.2 Reservoir top vibration accelerometer results

The results for reservoir top vibration for the 4 combinations are shown in Figures 53 & 54.

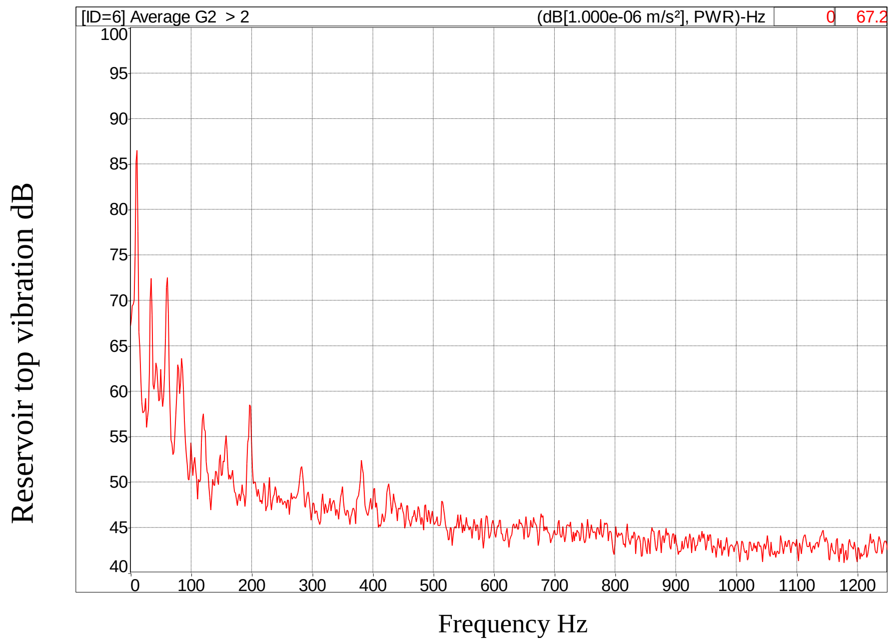


A 1500 rpm sprung

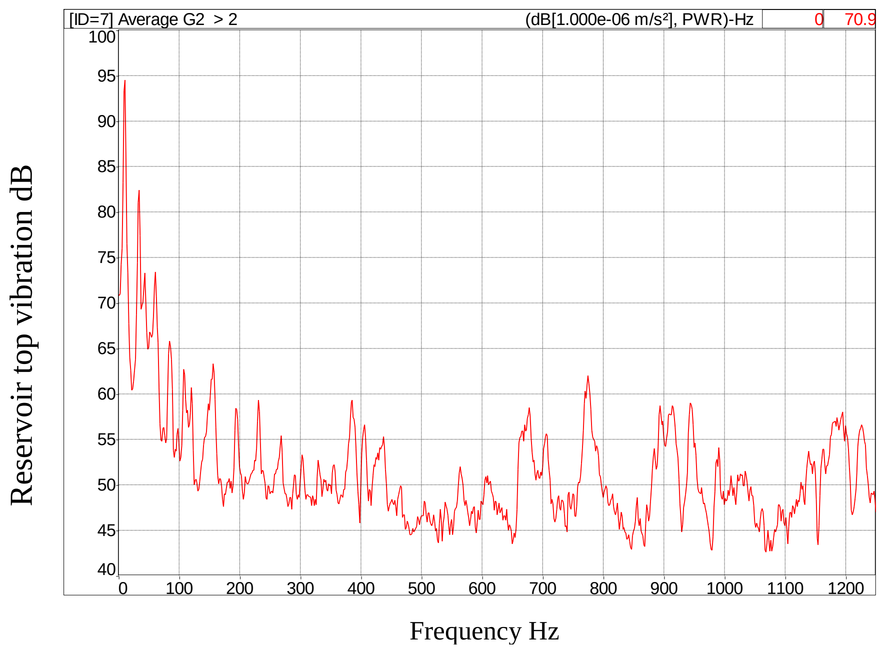


B 3000 rpm sprung

Figure 53 1500 & 3000 rpm sprung reservoir top vibration spectra



A 1500 rpm weighted



B 3000 rpm weighted

Figure 54 1500 & 3000 rpm weighted reservoir top vibration spectra

6.3.3 Reservoir top vibration accelerometer results 1/3 octave bands

An alternative method of analysing the vibration of the reservoir top is to use 1/3 octave frequency bands, as shown in Figures 55 & 56 for the 4 combinations.

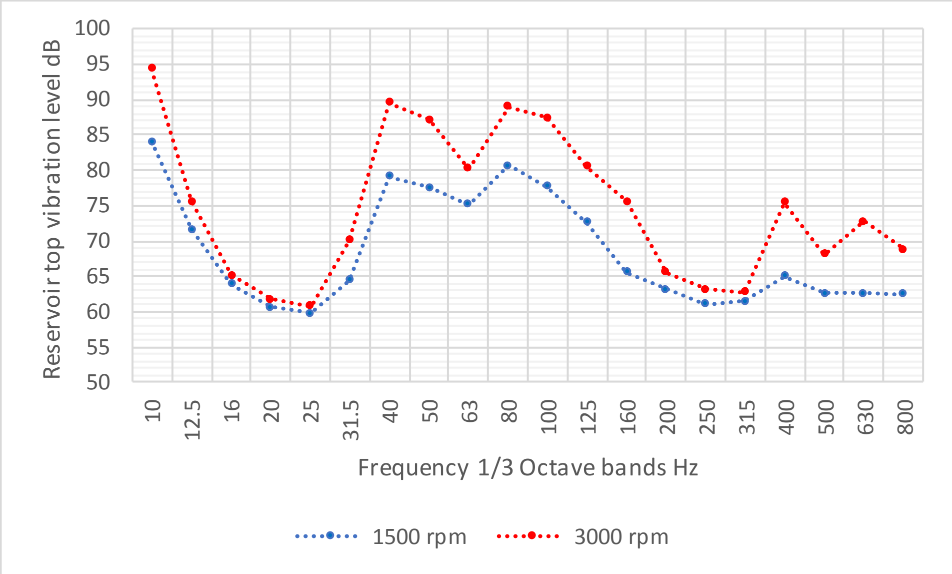


Figure 55 Weighted reservoir top vibration levels

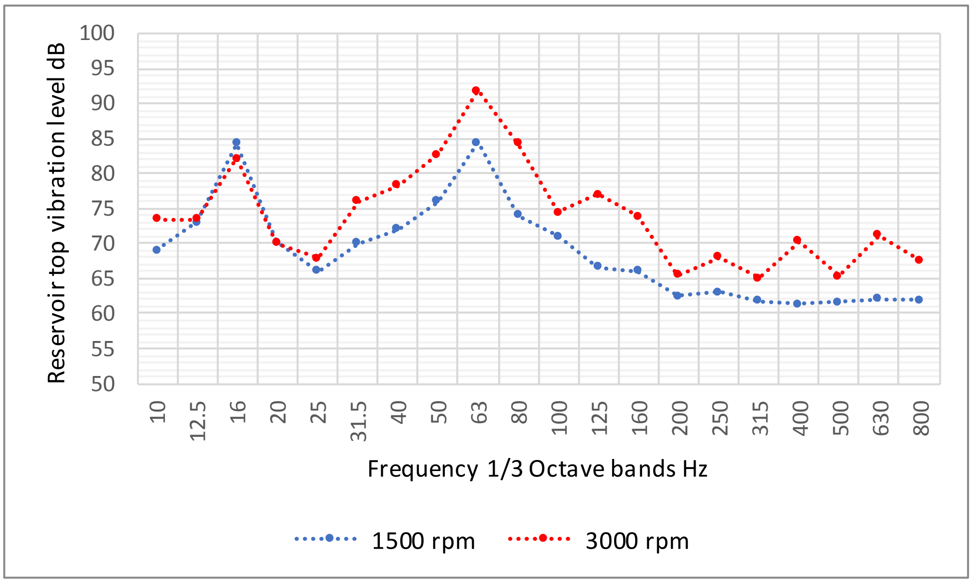


Figure 56 Sprung reservoir top vibration levels

6.4 Discussions

The frequency spectra for the test organ pipe, Figures 51 & 52, show the level of side-bands. The highest side-band level is produced from the weighted 3000 rpm blower combination, and the lowest, for the sprung 1500 rpm blower combination. However, there is little difference between the weighted 1500 rpm and the sprung 3000 rpm combination.

The corresponding results of reservoir top vibration, Figures 53 & 54, also show the maximum reservoir top vibration level (94dB) occurs with the weighted 3000 rpm blower combination. The minimum reservoir top vibration level (82dB) occurs with the sprung 1500 rpm blower combination. The other two combinations give very similar reservoir top vibration levels (85dB).

Figures 55 & 56, using 1/3 octave bands, give a clearer picture of reservoir top vibration and the difference between the 4 combinations is more apparent.

6.5 Conclusion

These findings support the views held by many organ builders that slow speed blowers produce less reservoir disturbance than high speed blowers.

7 Reservoir Resonance

7.1 Preamble

Chapter 6 outlined two basic ways of pressurising the reservoir. The first method was to apply cast iron bellows weights to the reservoir top. The second method was to fix tension springs to each corner of the reservoir. In this Chapter the resonant frequencies of each type of reservoir loading are investigated.

7.1.1 Weighted reservoir

The reservoir can be modelled as a mass spring system, and the resonant frequency of a weighted reservoir is determined by the following expression Barron [37]:

$$fres = \frac{1}{2\pi} \cdot \frac{1}{\sqrt{Ma.Ca}} \qquad (4)$$

Where Ma = Mechanical Mass (kg) / Area of the reservoir (m^2)

Ca = Volume of retained air (m^3) / density of air x (speed of sound)^2

Example: The mechanical mass of the test reservoir is approximately 75kgs

Area = 1.2 x 1.2 = 1.44m^2

Volume = 1.2 x 1.2 x .4 = 0.576m^3

Ma = 75/1.44 = 52.1kgs/m^2

Ca = 0.576/(1.2 x $(343)^2$) = 4.07x10^{-6} m^5/N

$fres$ = (1/2π) x 1/(52.1 x 4.07x10^{-6})$^{1/2}$ = 10.93Hz

The acoustic mass = 0.7kg (This represents only 1% of the mechanical mass)

The acoustics $fres$ = 95Hz

A more convenient equation, expressed in organ building terms of wind pressure and reservoir height, was outlined by Vellecote [24].

$$fres = \frac{1882}{\sqrt{p.h}} \qquad (5)$$

Where p = wind pressure (mm) water gauge h = reservoir height (mm)

In the above example p = 75mm h = 400mm

$fres$ = 1882/(75 x 400)$^{1/2}$ = 10.86Hz

Figure 57 shows both methods for wind pressures up to 200mm water gauge and a reservoir height of 400mm.

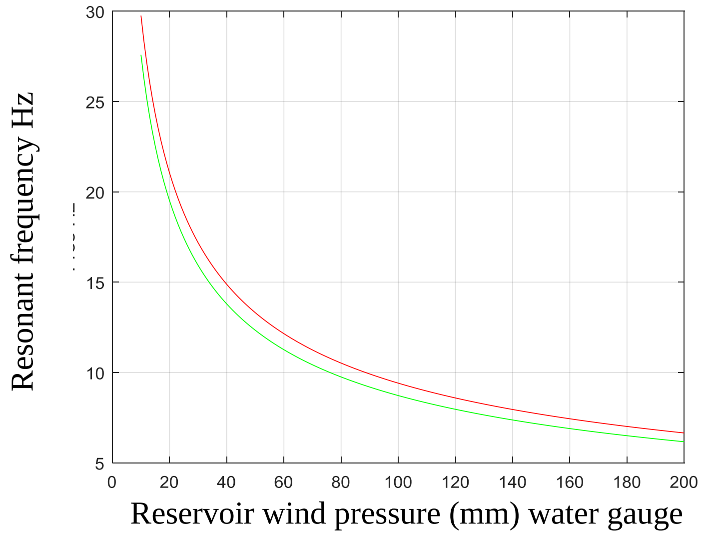


Figure 57 Comparison of reservoir resonant frequency

Equation 4 green & Equation 5 red.

Figure 58 shows reservoir resonant frequencies for a weighted reservoir and reservoir heights of 200, 300 & 400mm.

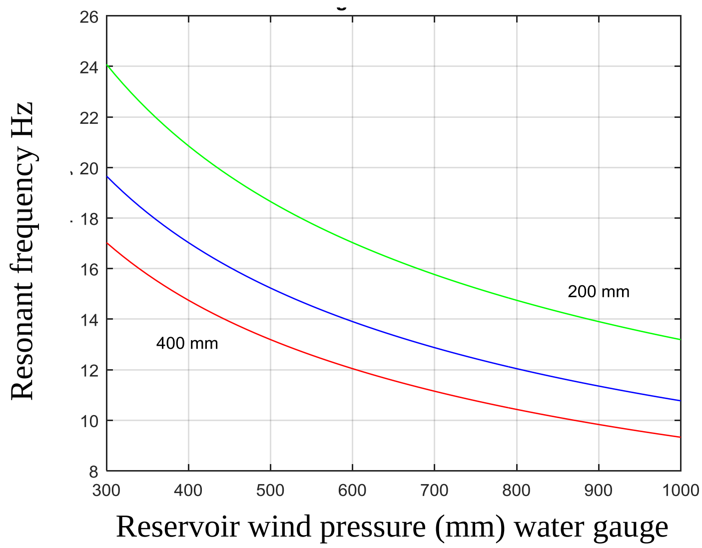


Figure 58 Weighted reservoir resonant frequency

7.1.2 Sprung reservoir

The resonant frequency is a function of the reservoir height and the mass of the reservoir top, and is determined in organ building terms, by the following equation Vellecote [24]:

$$fres = \frac{1882}{\sqrt{Mt.h}} \tag{6}$$

Where Mt = mass of the reservoir top (kg)

h = reservoir height (mm)

note: *fres* is not a function of the reservoir wind pressure.

Example: Mt = 25kg h = 400mm

$fres = 1882 / (25 \times 400)^{1/2} = 19\text{Hz}$

Using equation (4) $fres$ = 18.9Hz

Figure 59 shows the resonant frequency for a sprung reservoir for various reservoir top masses and reservoir heights.

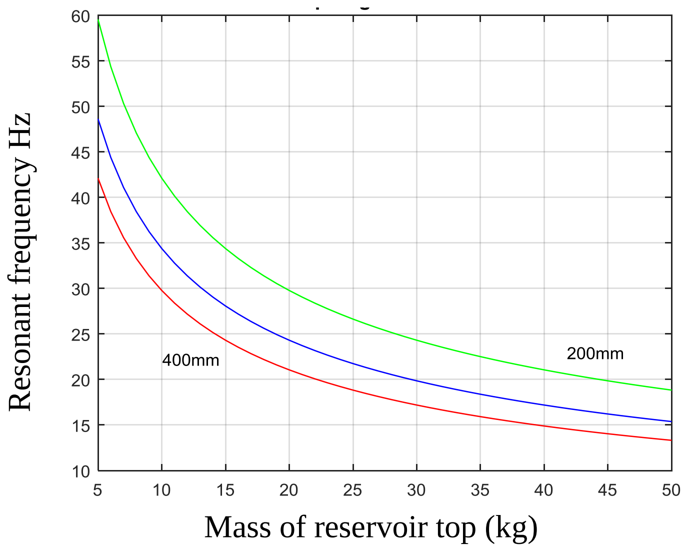


Figure 59 Sprung reservoir resonant frequency

7.2 Methodology

Using the apparatus and measuring equipment described in Chapter 4, weighted and sprung reservoir combinations are examined.

7.3 Results

7.3.1 Weighted Reservoir

7.3.1.1 Comparison of calculated and actual reservoir frequencies

Calculated and actual reservoir frequency plots are shown in Figure 60.

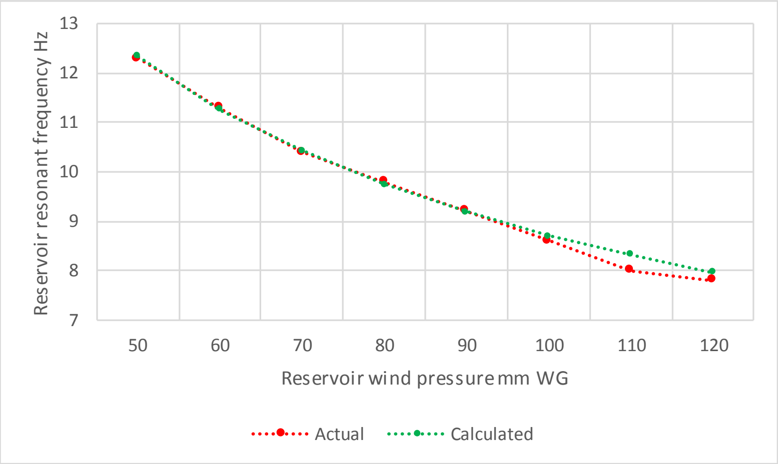


Figure 60 Comparison of actual and calculated reservoir vibration

7.3.1.2 The effects of reservoir height on reservoir wind pressure

Figure 61 shows the relationship between reservoir height and reservoir wind pressure. The reservoir has a single set of inward folding ribs and a height of 400mm when full.

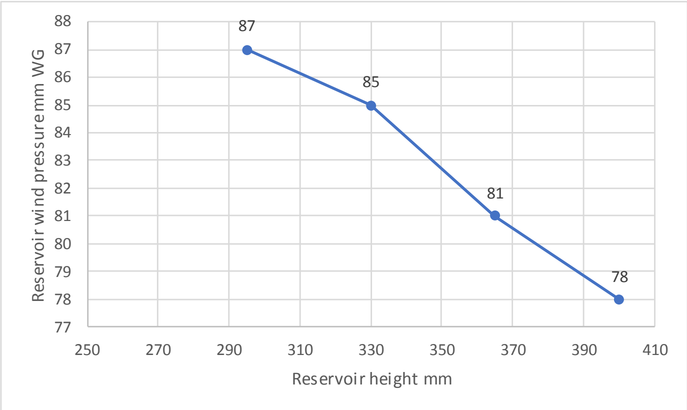


Figure 61The relationship between reservoir height and reservoir wind pressure

7.3.1.3 Variation in reservoir top vibration levels for various rib angles

As the height of the reservoir changes so does the angle that the ribs make with each other. At 400mm, the rib angle is 83 degrees and corresponds with a reservoir wind pressure of 78mm water gauge. The other rib angles are 57, 38 & 20 degrees for corresponding heights of 365, 330 & 295mm. Figure 62 shows the effect that changing the rib angle has on the reservoir top vibration levels.

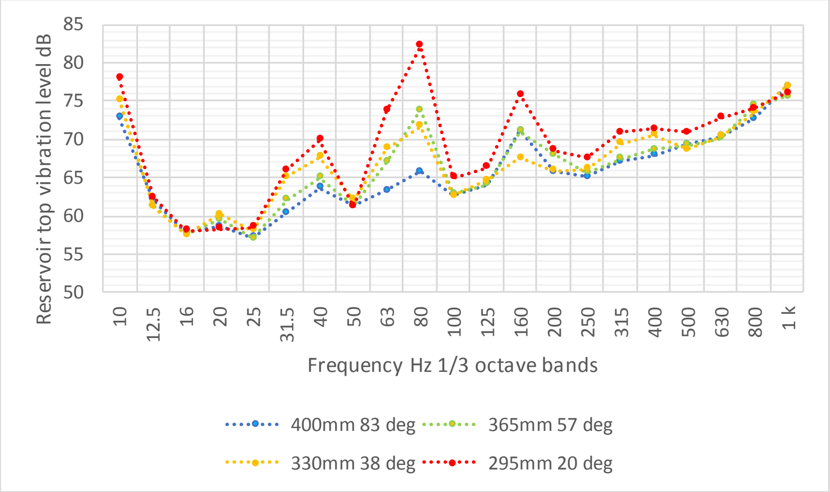


Figure 62 The relationship between reservoir top vibration levels, reservoir height and rib angles

7.3.1.4 Reservoir top vibration levels for various reservoir wind pressures

The reservoir top vibration levels, together with the organ pipe spectra, were measured for wind pressures of 50mm to 120mm WG, and a reservoir height of 400mm. The results are shown in Figure 63 together with the corresponding frequency spectra, Figures 64 to 71, showing the fundamental frequency and side-band levels.

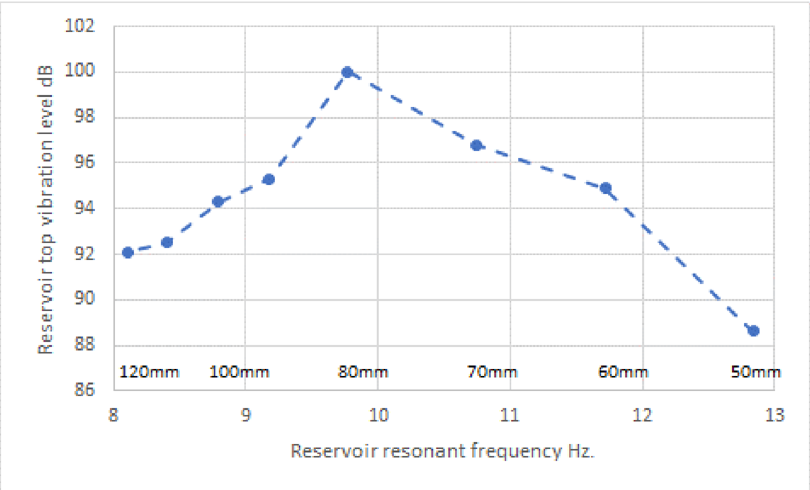


Figure 63 Variation in reservoir vibration level for different wind pressures

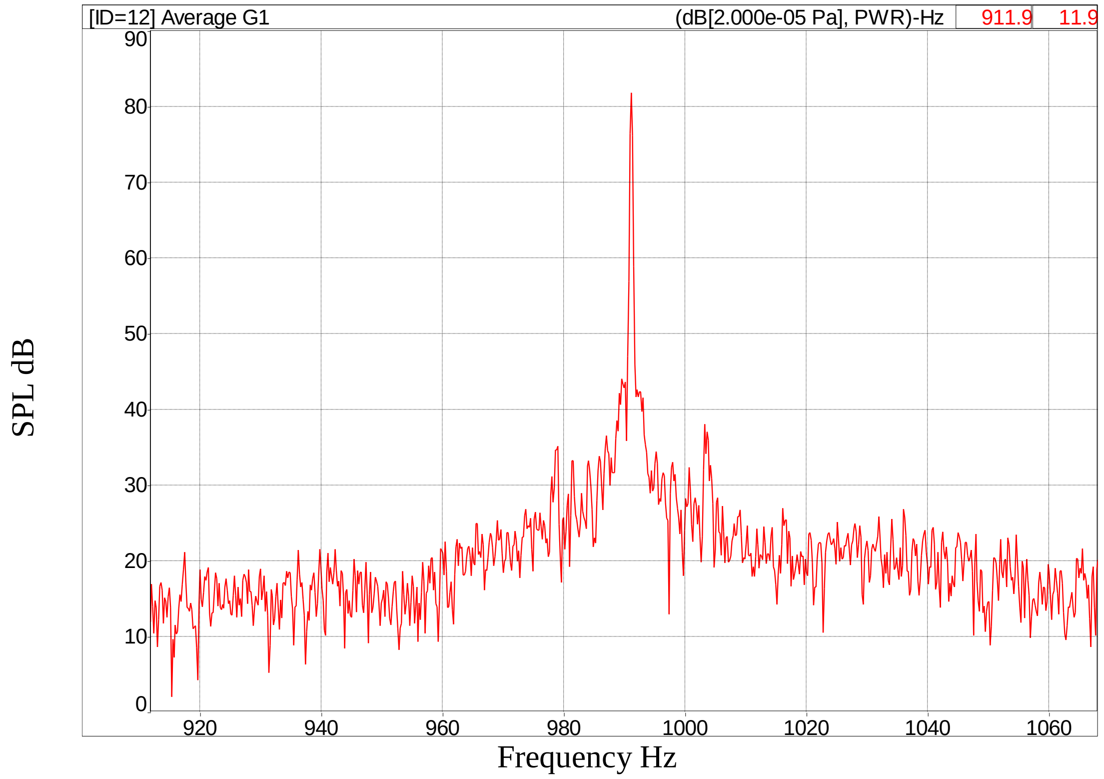


Figure 64 50mm reservoir wind pressure

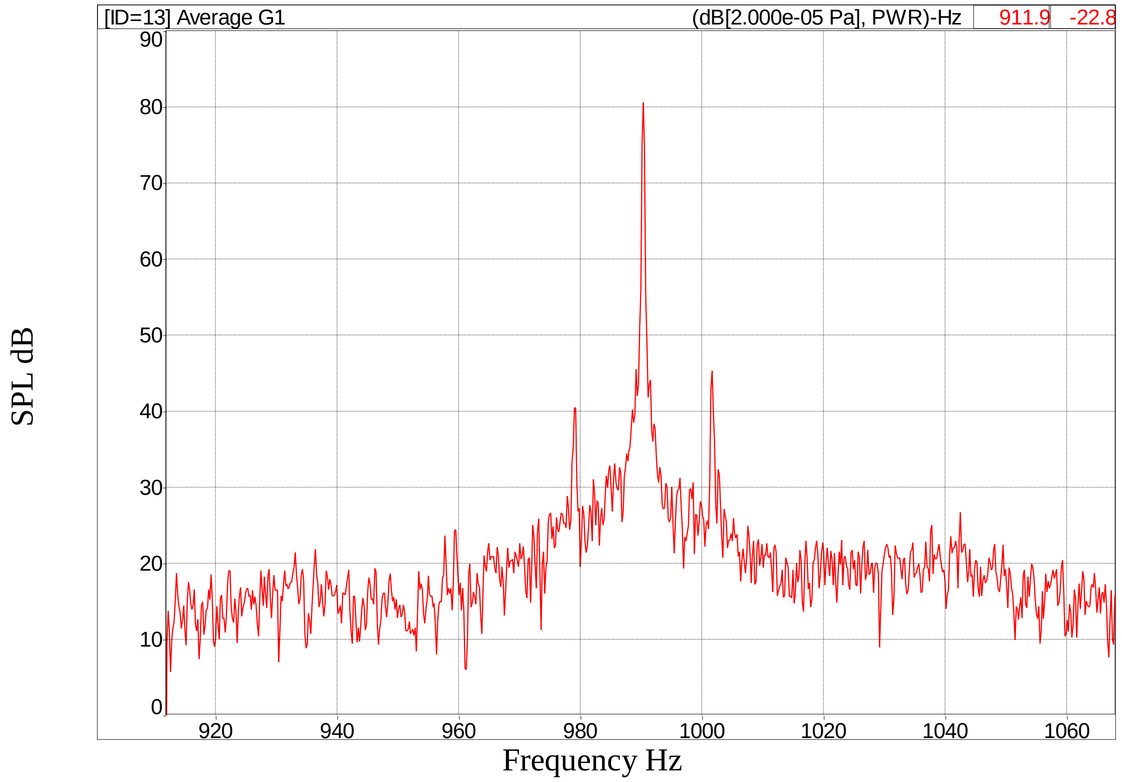


Figure 65 60mm reservoir wind pressure

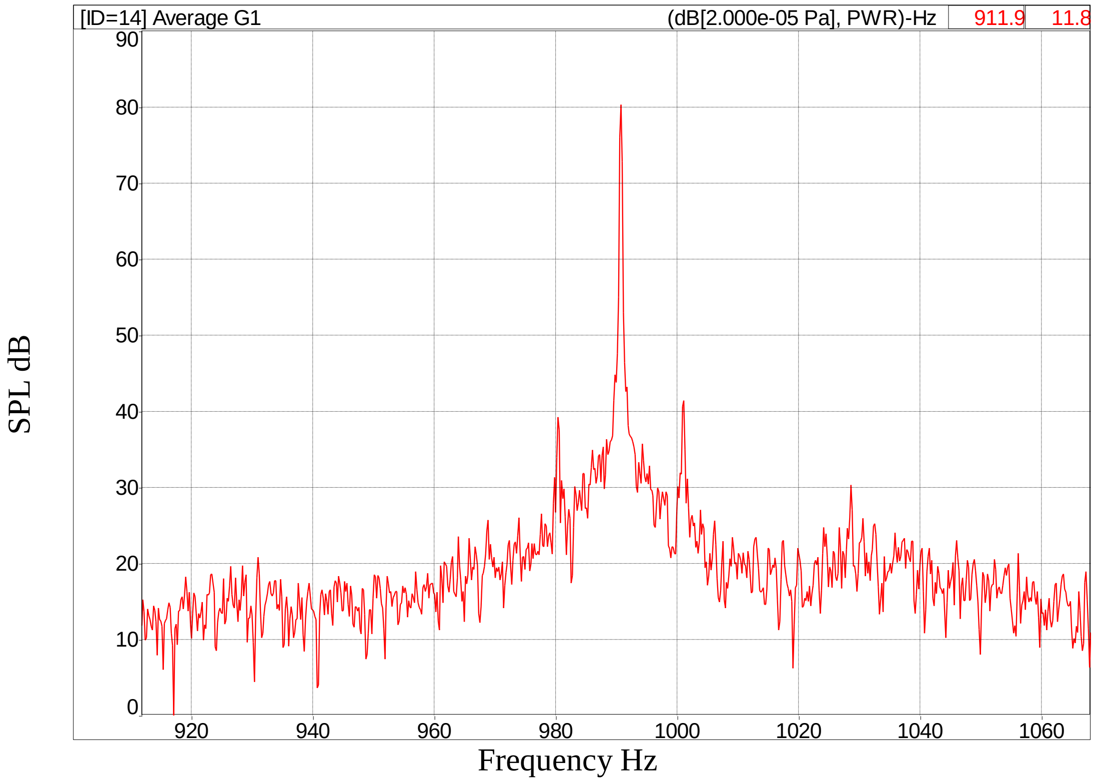


Figure 66 70mm reservoir wind pressure

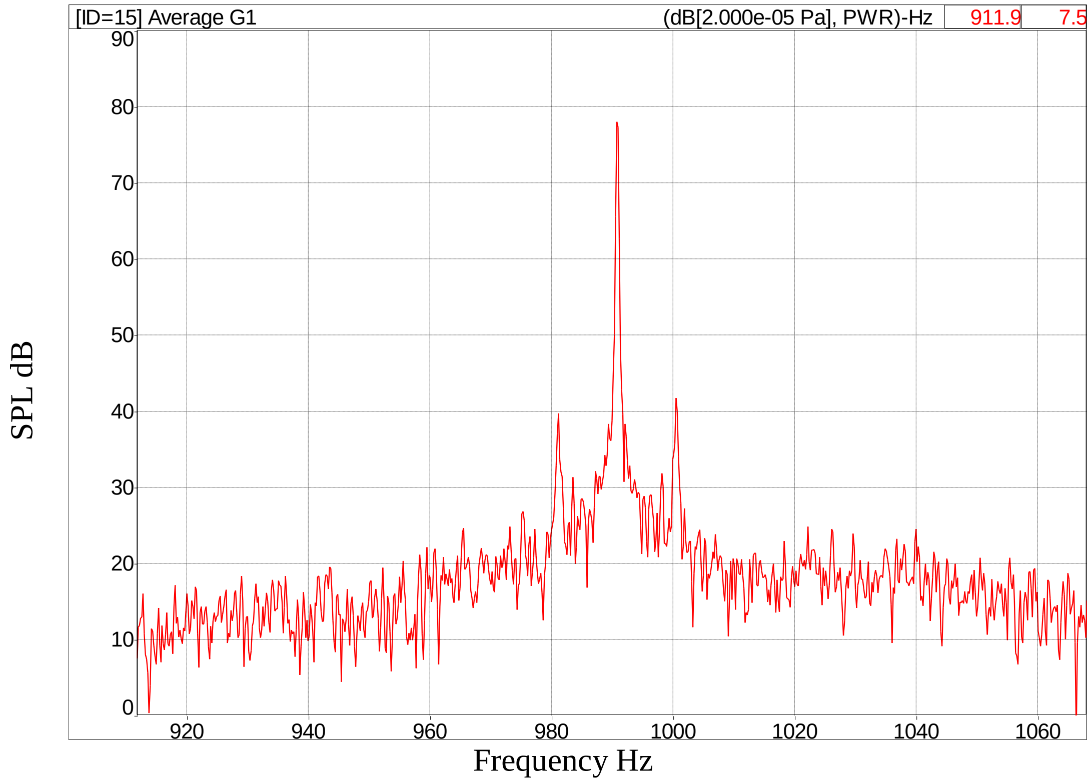


Figure 67 80mm reservoir wind pressure

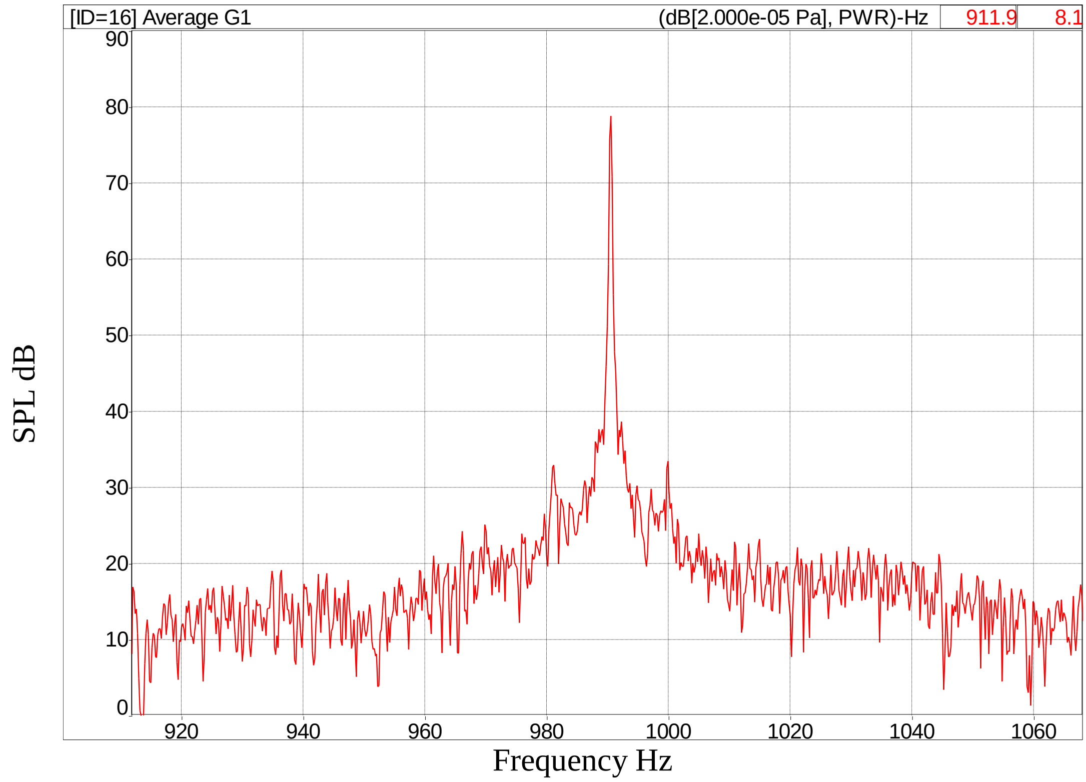


Figure 68 90mm reservoir wind pressure

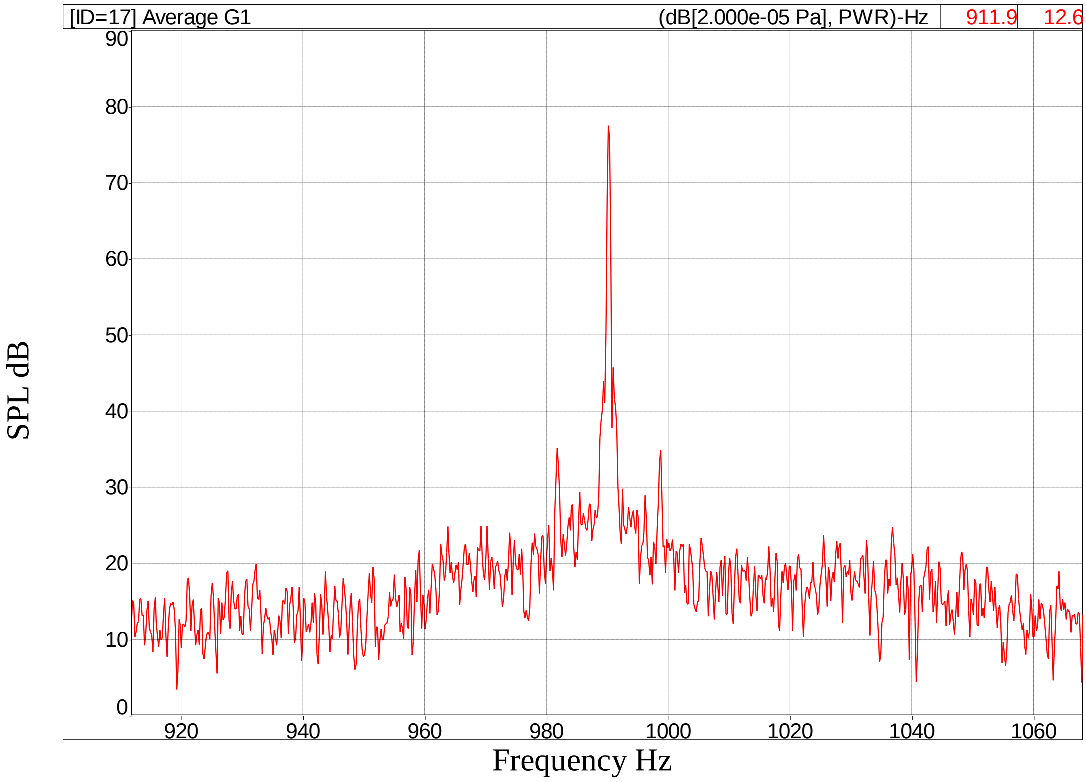


Figure 69 100mm reservoir wind pressure

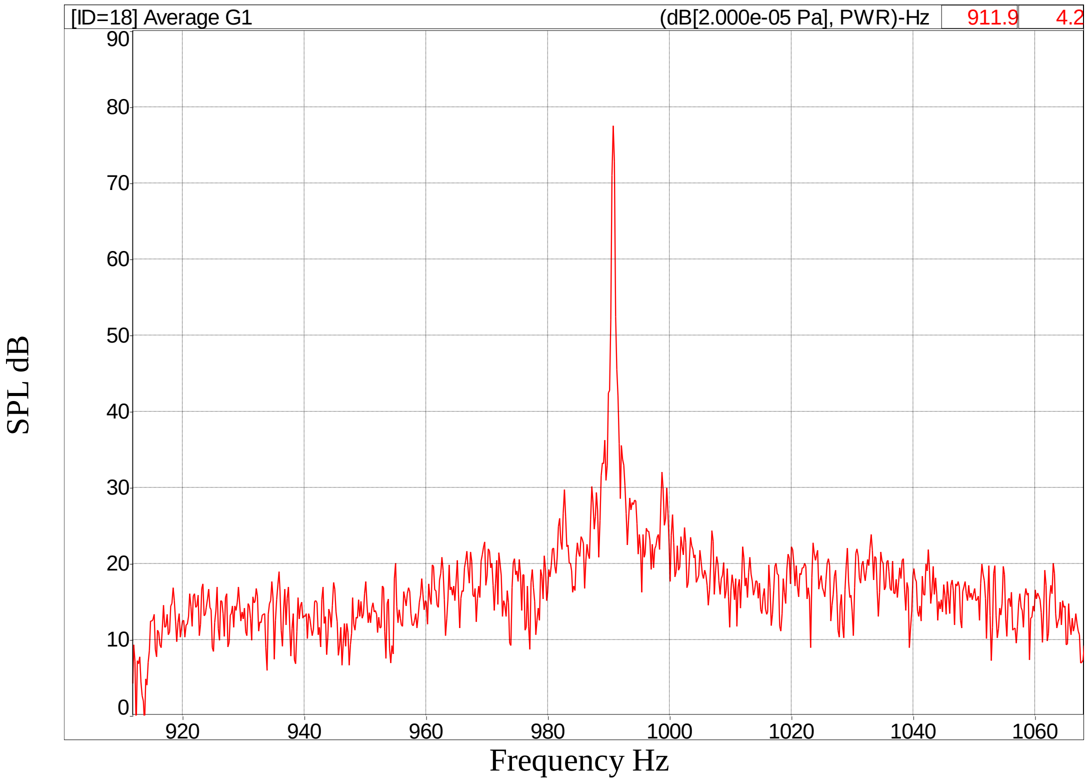


Figure 70 110mm reservoir wind pressure

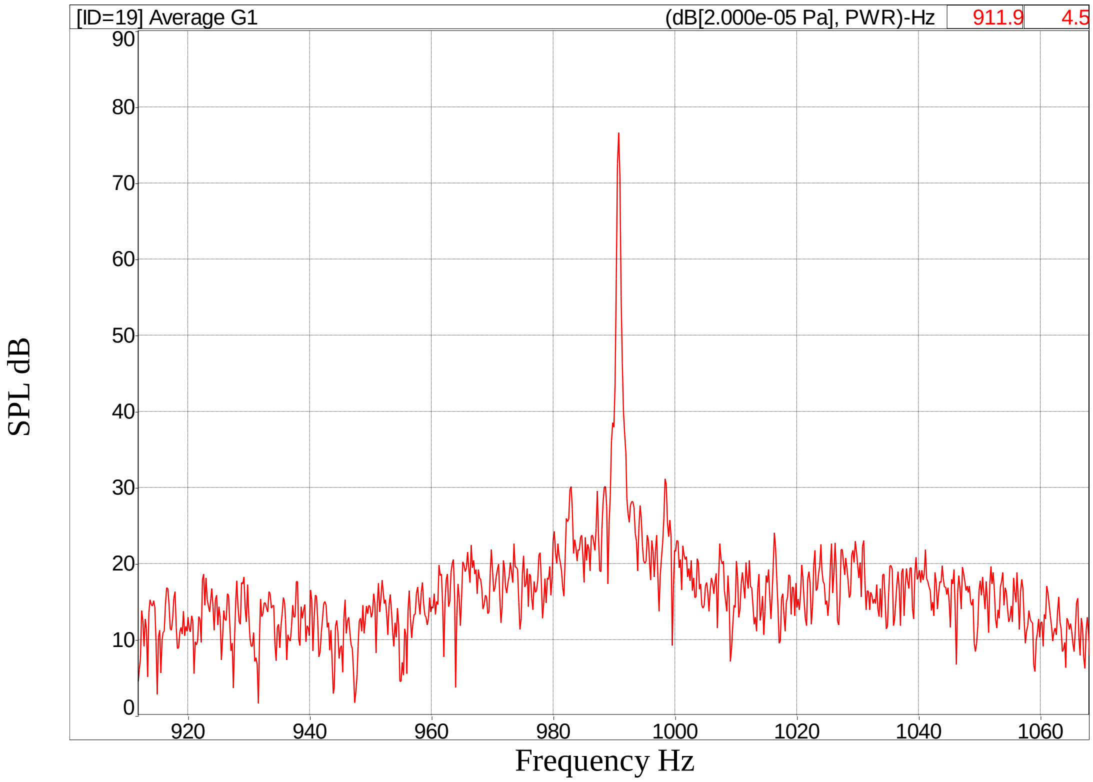


Figure 71 120mm reservoir wind pressure

7.3.2 Sprung Reservoir

The weights were removed, and the reservoir pressurised using the four external springs.

7.3.2.1 Variation of reservoir top frequency with reservoir wind pressure

The relationship between the reservoir top vibration levels and the three wind pressures (50, 57.7 & 62.5mm WG) is shown in Figure 72.

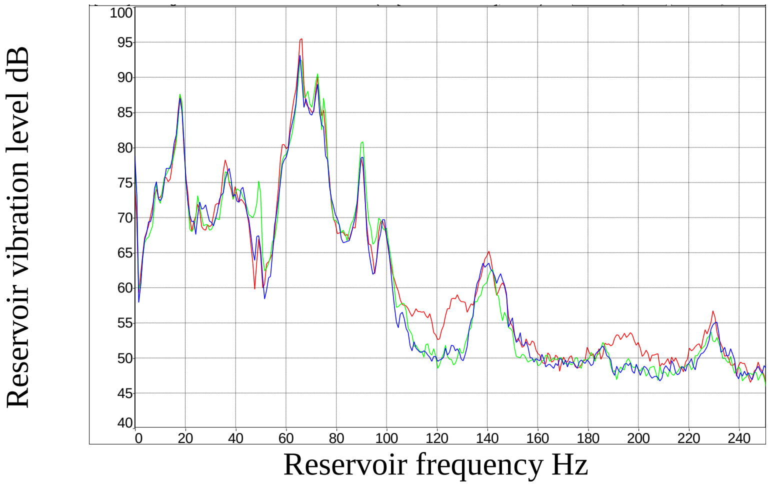


Figure 72 Fundamental frequency of a sprung reservoir for
50mm (red) 57.5mm (green) & 62.5mm (blue) wind pressures

7.4 Tuning a weighted single rise reservoir

Results for varying the weight spring ratio are shown in Figure 72A.

Figure 72A Tuning a weighted single rise reservoir.

7.5 Discussion

Figure 17 shows that 85% of the organs surveyed in the pipe organ survey had weighted double rise reservoirs. For weighted reservoirs the mechanical mass will always be dominant. The survey also shows that the most common reservoir height was 300mm and the most common wind pressure was 3 inches or 75mm WG.

The resonant frequency of a weighted reservoir is a function of the reservoir height and reservoir wind pressure. Normally, wind pressures vary between 50 and 120mm WG producing a resonant frequency of approximately 7.5 to 13Hz. Comparison between the theoretical and actual resonant frequency of a weighted reservoir shown in Figure 60 is good. The theoretical reservoir resonant frequency curves, shown in Figures 58 & 59, for weighted and sprung reservoirs for various reservoir heights, will be a valuable tool for many organ builders.

The frequency spectra, Figures 64 to 71, clearly show the development of side-bands on each side of the fundamental as the wind pressure is varied, reaching a maximum at one wind pressure. The frequency spectra also show that the presence and magnitude of side-bands on each side of the fundamental is a good indication of the amount of amplitude modulation or organ pipe flutter.

Figure 62 shows that changing the geometry of the reservoir ribs by adjusting the reservoir height, has a dramatic effect on the vibration levels of the reservoir top.

An important property of a sprung reservoir, shown in Figure 72, is that adding more spring force to increase the wind pressure, does not change the resonant frequency.

7.6 Conclusion

Figure 72A shows that removing weights from the reservoir top, and adding springs to maintain the reservoir wind pressure, results in an optimum condition being reached for reservoir top vibration levels.

Reservoir dimensions and geometry, together with reservoir wind pressure are important when considering reservoir resonance and organ pipe flutter. Armed with this information, pipe organ builders will be able to select reservoir dimensions and wind pressures that will give the most stable wind system with the minimum of pipe flutter.

8 Blower comparison tests

8.1 Preamble

From the initial 1500 & 3000 rpm blower experiments it is clear that the reservoir is excited by the organ blower. In this Chapter a more comprehensive series of blower experiments are conducted to determine if all 1500 & 3000 rpm blowers behave in a similar manner, and to determine if one element of the blower design is responsible for organ pipe flutter.

The blower comparison tests compare the effect of different blowers on the test single rise weighted wind system. Changing the blower should not produce different results in the wind system. By keeping the reservoir and connecting pipework constant and varying the blower, it should be possible to determine if one type of blower is more likely to induce unwanted reservoir top vibrations.

The various blowers represent a good cross-section of blowers used in many small and medium sized pipe organs. From Table 12 it can be seen that they represent several manufacturers with 1500 & 3000 rpm motor configurations, static wind pressures and blade configurations. For the sake of completeness, I have included a garden leaf blower, which is not specifically designed for organ building applications, but produces a high output pressure, thus, a comparison can be made with the organ blowers.

Blower reference	Make	Motor speed rpm	Mains voltage	Number of poles	Static Pressure mm WG	No of blades	Blade frequency Hz	Impeller diameter mm	Peripheral speed m/sec	Impeller width mm	Case width mm
1*	Discus	1500	415 ac	4	95	12 rad	300	533	40	20	114
2**	BOB TOT	1500	dc		95	8 BC 72	200 1800	533	42	22	89
3 ***	BOB Minor	1500	415 ac	6	86	8 BC 80	172 1720	558	38	28	89
4	Taylor	3000	240 ac	2	100	10 rad	500	260	41	15	80
5	4214-1	3000	240 ac	2	150	12BC 48	600 2400	312	49	35	125
6****	4222-3	3000	240 ac inv	2	180	12BC 48	600 2400	338	53	45	135
7	K C	3000	240 ac	2	140	6Rad 42	300 2100	317	47	27	70
8	Leaf blower		240 ac		350	5 FC		150	75		

Table 12 Blower details

* The blades of the impeller are set 25mm below the rim.

****** This blower is controlled by a 230 volt ac dc controller

*** This blower is belt driven and the motor runs at 1000 rpm. The pulley ratio gives a final drive speed of 1500 rpm

**** This blower is controlled by a 240 volt ac speed inverter drive.

8.2 Methodology

Using the apparatus and measuring equipment described in Chapter 4, with a reservoir wind pressure of 80mm WG, the eight blowers are examined.

8.3 Blower details

8.3.1 Blower 1

This old Discus unit is the largest and heaviest unit tested. The unit has a very heavy cast iron case, a 12-blade radial steel impeller and large ½ hp three phase motor.

Photograph 12 Blower 1

8.3.2 Blower 2

This blower was manufactured by British Organ Blowing Ltd. The unit has a steel plate fabricated case and steel impeller, and was originally fitted with a ¼ hp single phase motor running at 1500 rpm. The existing motor was time expired and replaced with a ½ hp dc motor and variable speed controller running at 1500 rpm.

Photograph 13 Blower 2

8.3.3 Blower 3

This blower was also manufactured by British Organ Blowing Ltd and designated as BOB Minor. The unit has a steel plate fabricated case and steel impeller. It was originally fitted with a ½ hp single phase motor running at 1500 rpm that was time expired. The existing motor was removed, and the impeller fitted to a shaft supported by two block bearings. The drive from the new 1000 rpm motor to the impeller is by two pulleys and a V belt. The ratio of the two pulleys is 1.5 to 1, giving a final drive speed of 1500 rpm.

Photograph 14 Blower 3

8.3.4 Blower 4

This blower, was made by Taylor's of Leicester and is the oldest and most primitive design. The unit is constructed from wood with a steel impeller and a single phase 3000 rpm ½ hp motor. The author has no connection with this company.

Photograph 15 Blower 4

8.3.5 Blower 5

This is a new blower with a cast aluminium casings and light aluminium impeller. The unit is fitted with a 3000 rpm ¾ hp single phase motor, fitted with sleeved bearings for exceptionally quiet running

Photograph 16 Blowers 5 & 6

8.3.6 Blower 6

This blower is similar to blower 5, but is rated at 22m^3 per minute, and is fitted with a 1hp single phase 3000 rpm motor.

8.3.7 Blower 7

The K C blower, manufactured by the Kinetic Company, is a small unit with a cast iron case and steel impeller, and is fitted with a single phase 3000 rpm 1/3hp motor. The Kinetic Company Lincoln was a subsidiary of the pipe organ builder J R Cousans, Lincoln. The patent for the design dates to 1903 and the first unit was installed in the pipe organ at Newland Congregational Church Lincoln to replace the failing hydraulic blower installed by Jardine & Co in 1891.

Photograph 17 Blower 7

8.3.8 Blower 8

The garden leaf blower has a 240 Volt single phase motor with an outlet velocity of 180 km/h. The 150mm diameter impeller is of open moulded plastic construction.

Photograph 18 Blower 8

8.4 Blower parameters

The various parameters for each blower are shown in Figures 73 to 76.

8.4.1 Blower static pressure

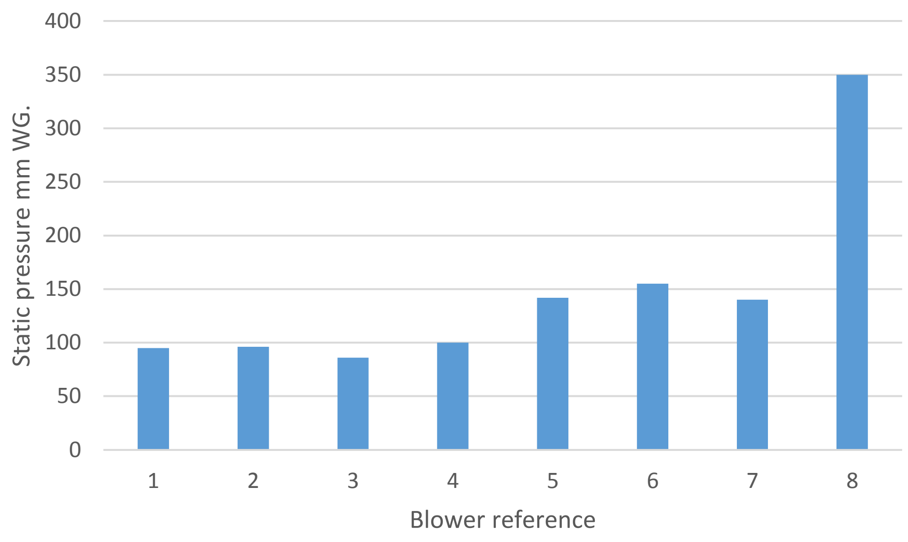


Figure 73 Static blower pressure

8.4.2 Number of blades

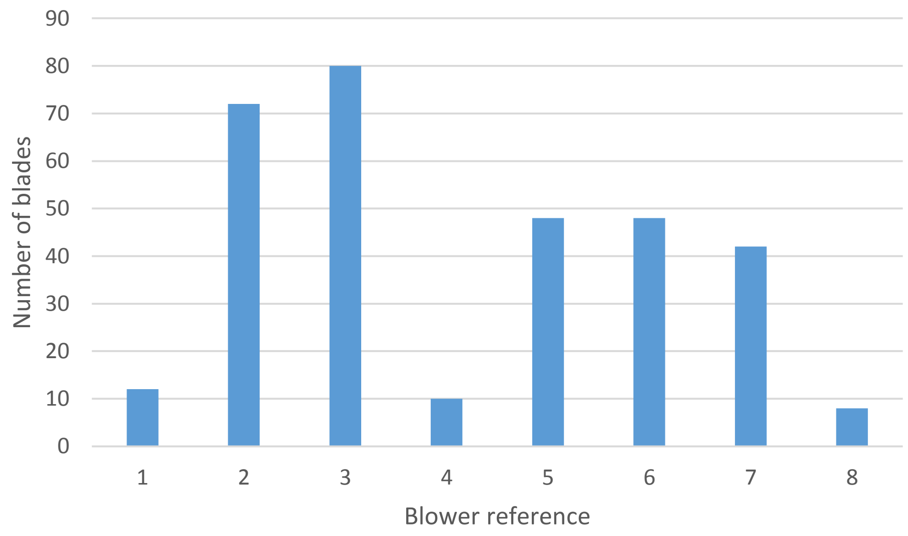


Figure 74 Number of blades

8.4.3 Peripheral tip speed

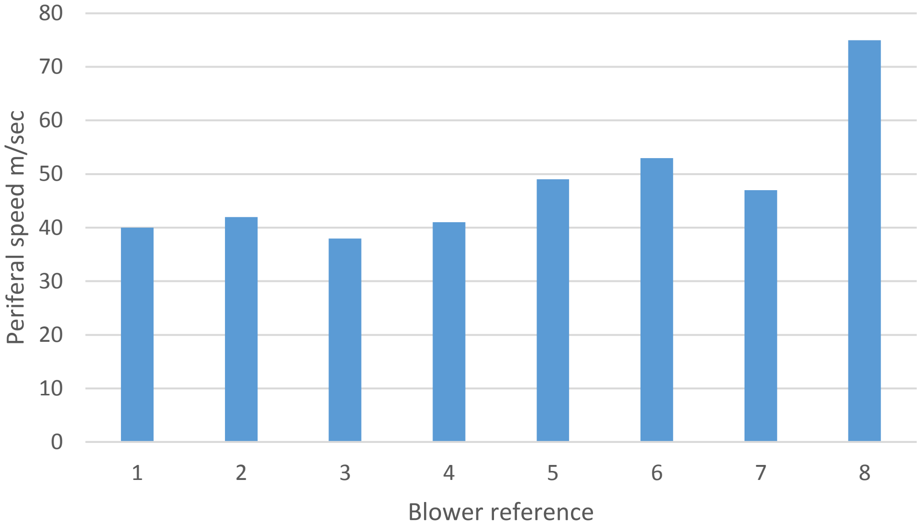


Figure 75 Peripheral speed

8.4.4 Impeller width

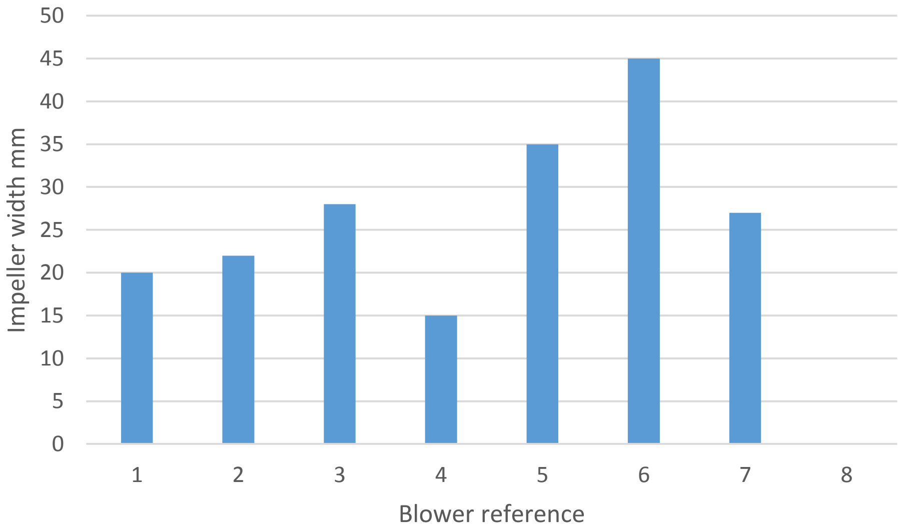


Figure 76 Impeller width

8.5 Results

8.5.1 Reservoir top vibration levels results.

Reservoir top vibration levels for the 8 blowers, are shown in Figures 77 & 81.

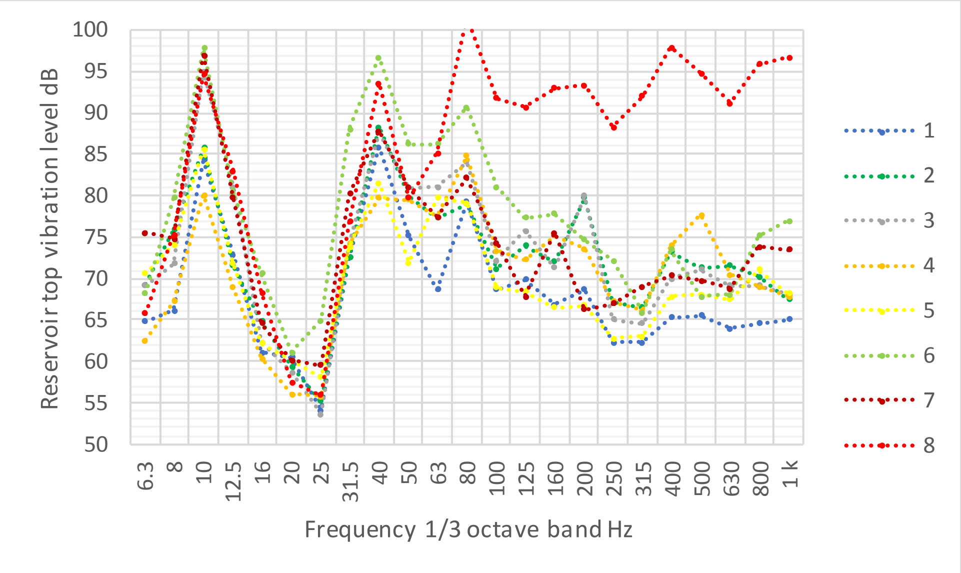


Figure 77 Reservoir top vibration levels for all 8 blowers 6.3Hz to 1kHz

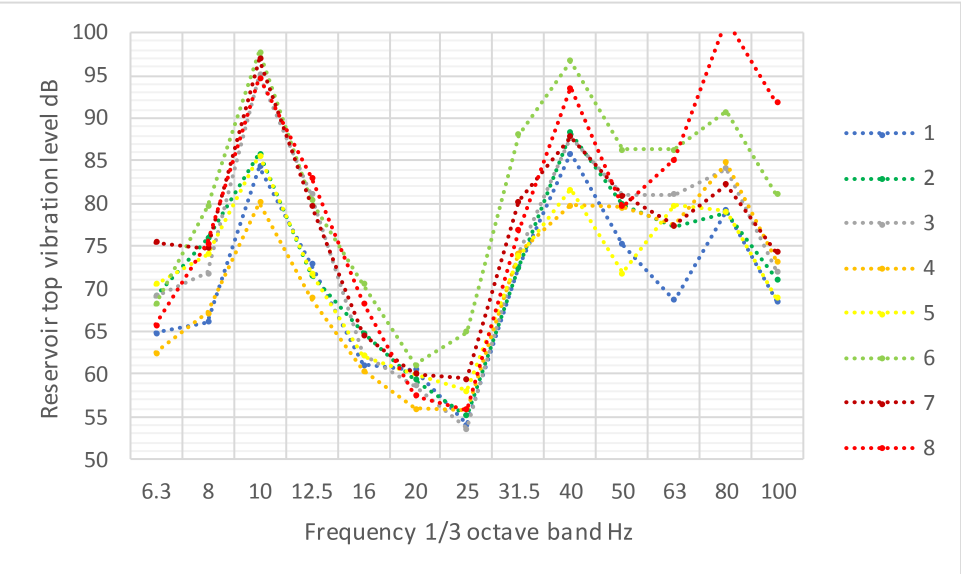


Figure 78 Reservoir top vibration levels for all 8 blowers 6.3Hz to 100Hz

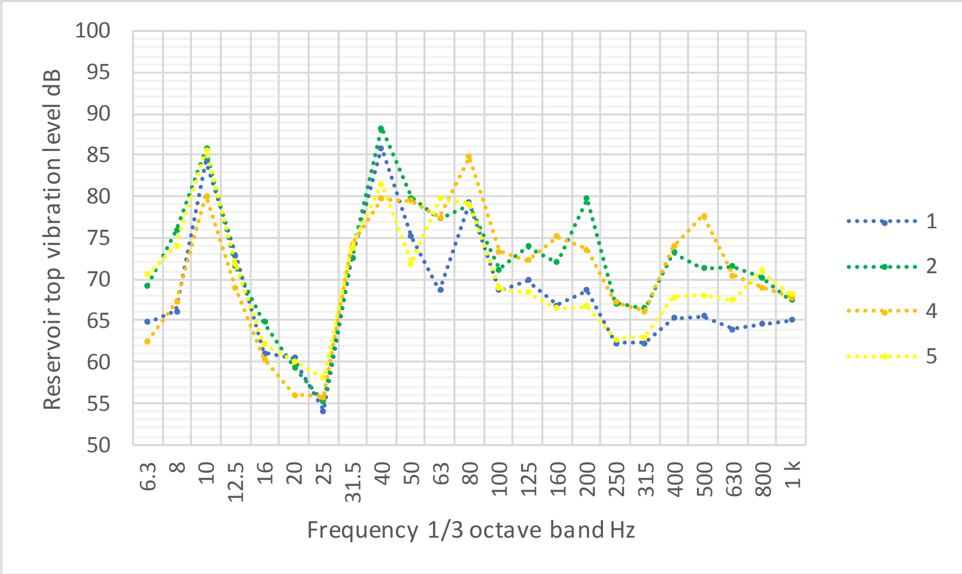


Figure 79 Reservoir top vibration levels for blowers 1, 2, 4, & 5 6.3Hz to 1kHz

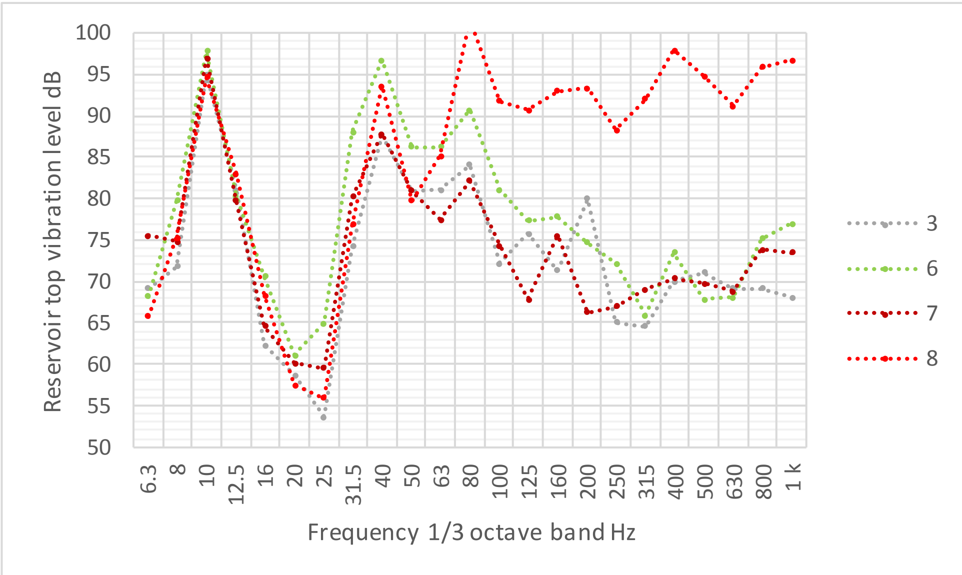


Figure 80 Reservoir top vibration levels for blowers 3, 6, 7, & 8 6.3Hz to 1kHz

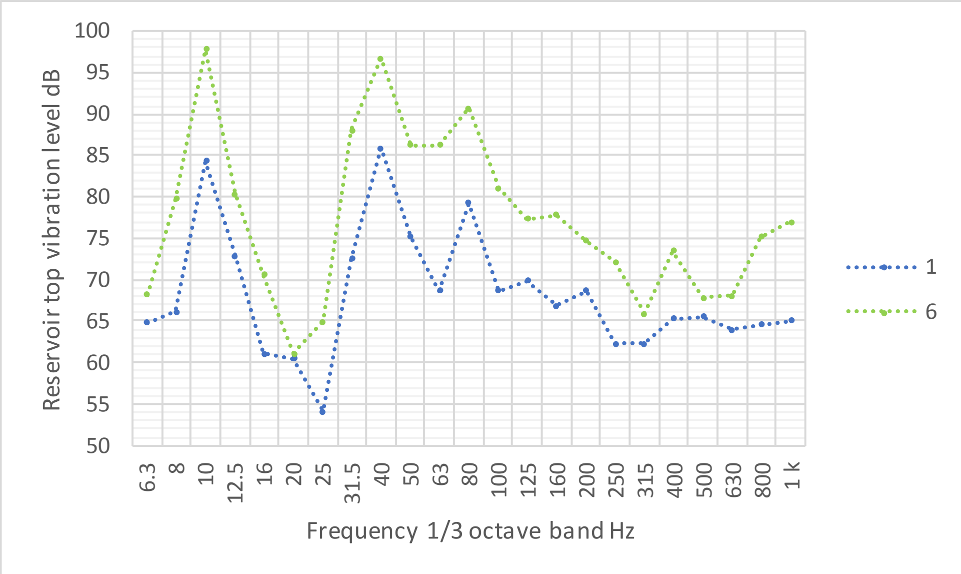


Figure 81 Reservoir top vibration levels for blowers 1 & 6 6.3Hz to 1kHz

8.5.2 Blower comparisons

The measurements of the eight blowers have been exported from 01dB as WAV files and imported into a MATLAB script (blowcompB) shown in Figure 82 & 83. By selecting the blower reference and a harmonic, it is possible to compare the effect that each blower has on the sounding pipe.

The first window displays the sounding pipe in the time domain to show the amount of amplitude modulation.

The second window shows the frequency spectrum of the sounding pipe for the first 10 harmonics.

The last window shows a zoom spectrum of a selected harmonic. It is this last window that is the most important and gives a good measure of the amount of amplitude modulation and organ pipe flutter. Blower 1 is shown in Figure 82 and blower 6 in Figure 83.

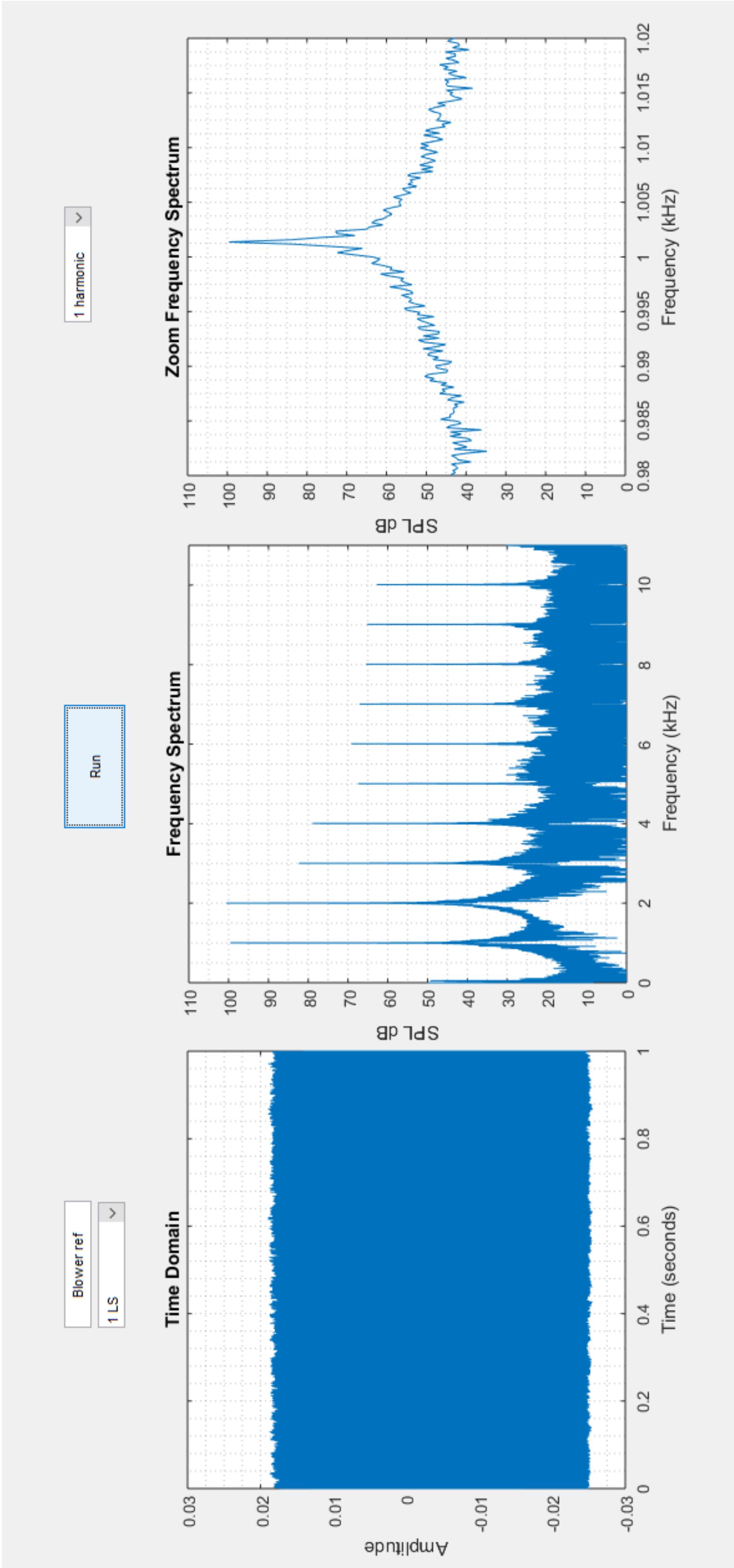


Figure 82 Blower 1 comparison

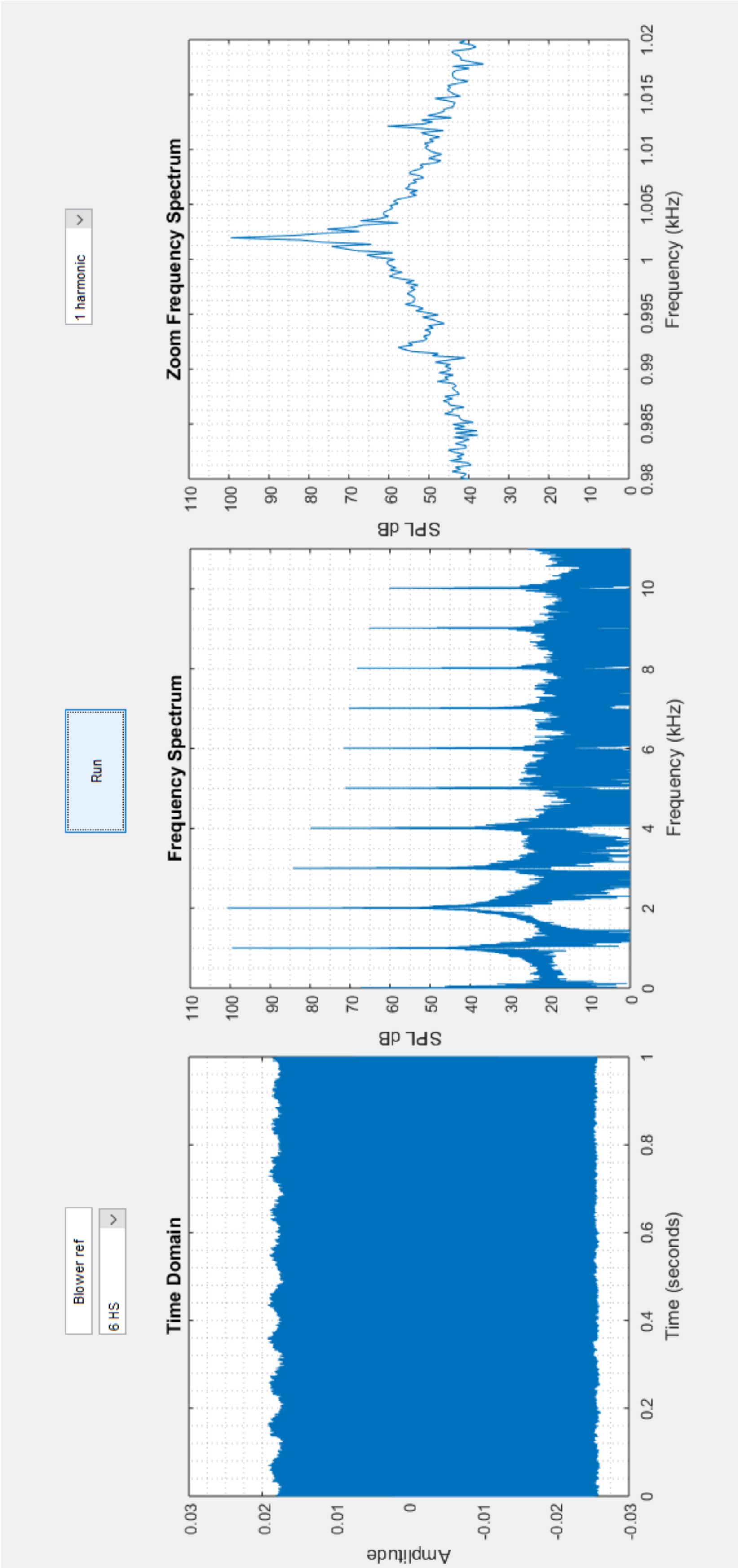


Figure 83 Blower 6 comparison

8.5.3 Percentage amplitude modulation results

Percentage amplitude modulation for each of the 8 blowers are shown in Figures 84 to 91.

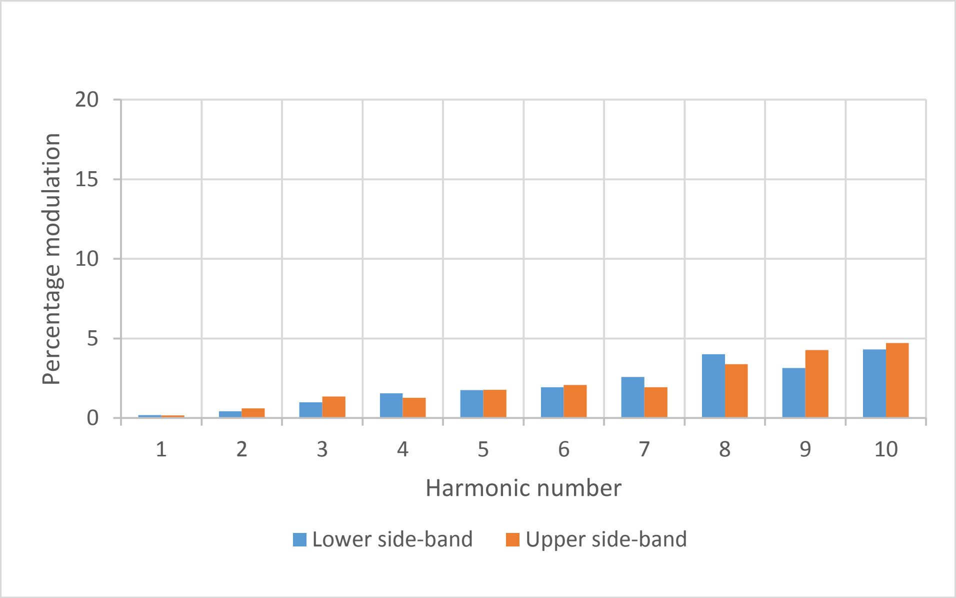


Figure 84 Blower 1 percentage amplitude modulation

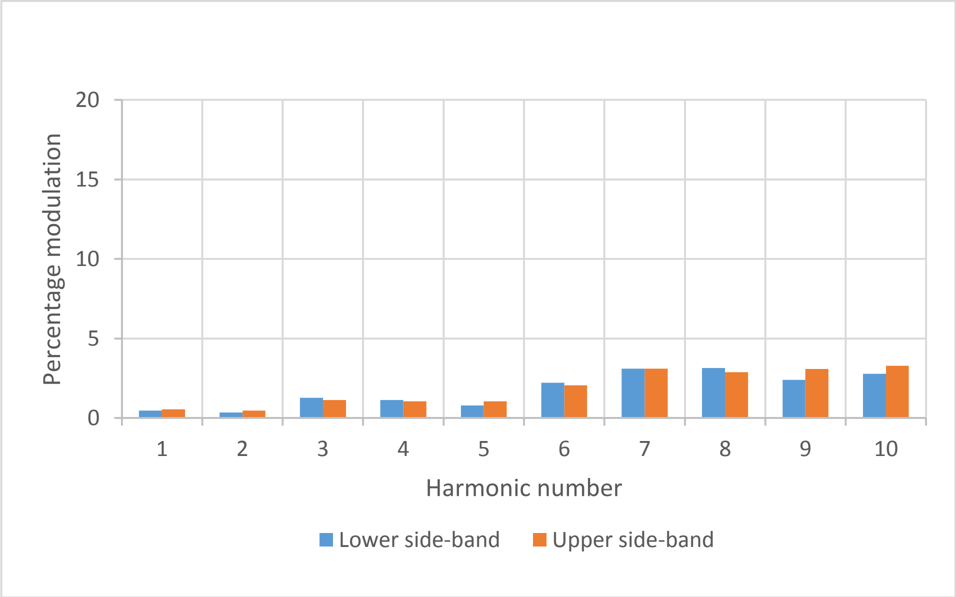


Figure 85 Blower 2 percentage amplitude modulation

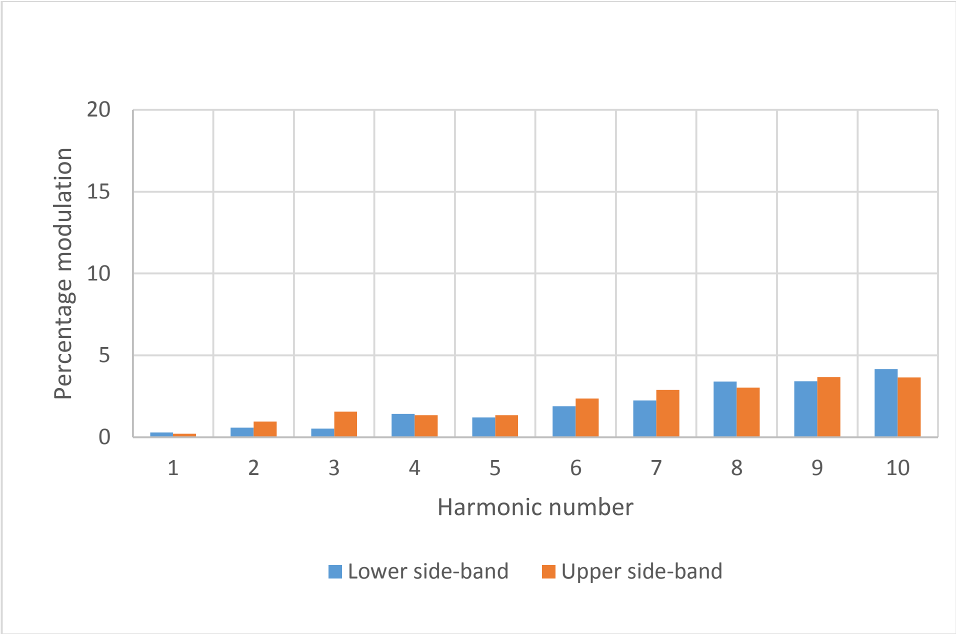


Figure 86 Blower 3 percentage amplitude modulation

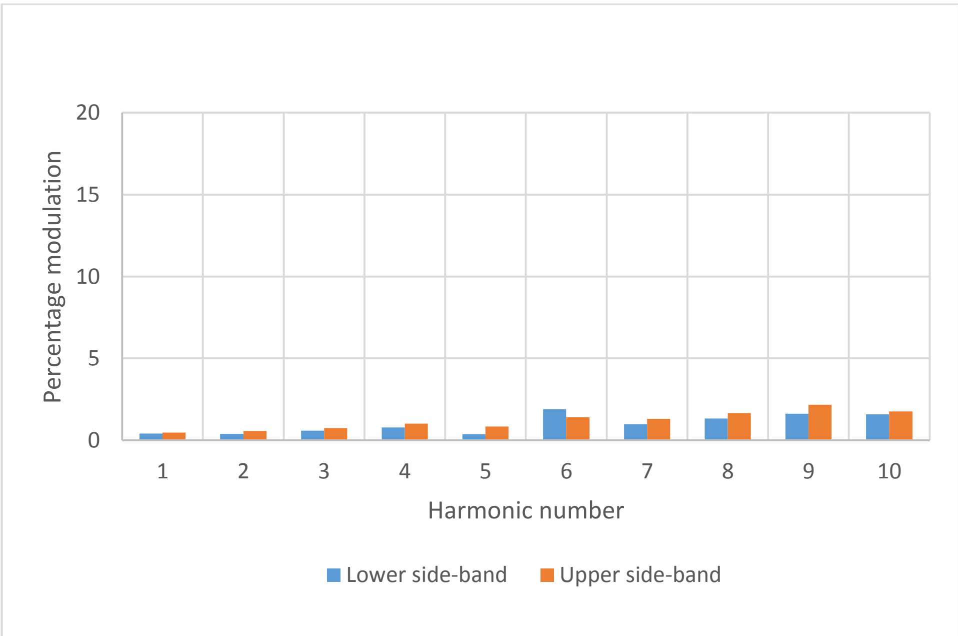


Figure 87 Blower 4 percentage amplitude modulation

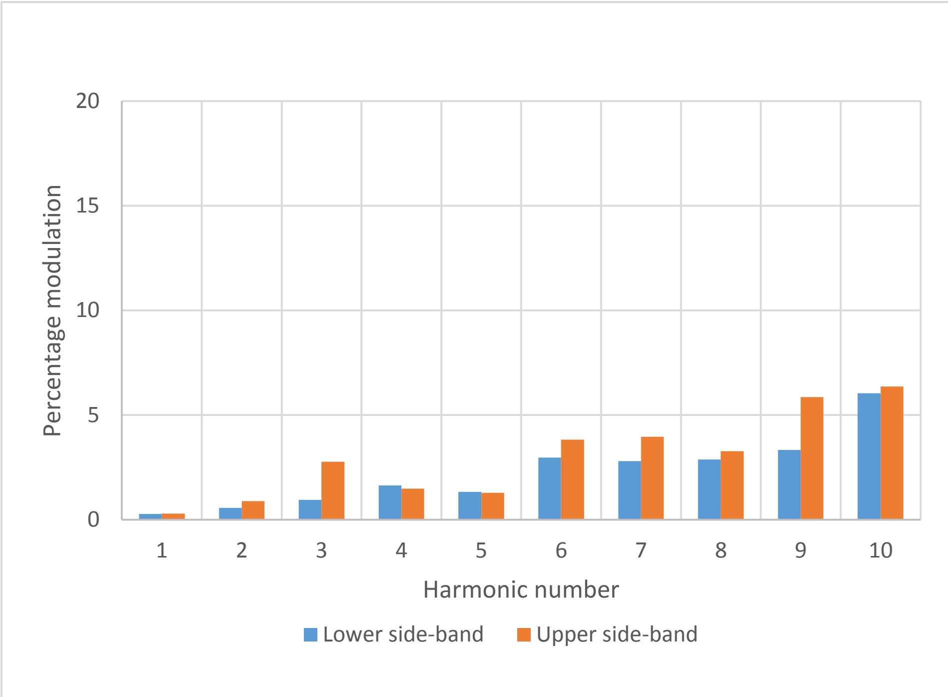


Figure 88 Blower 5 percentage amplitude modulation

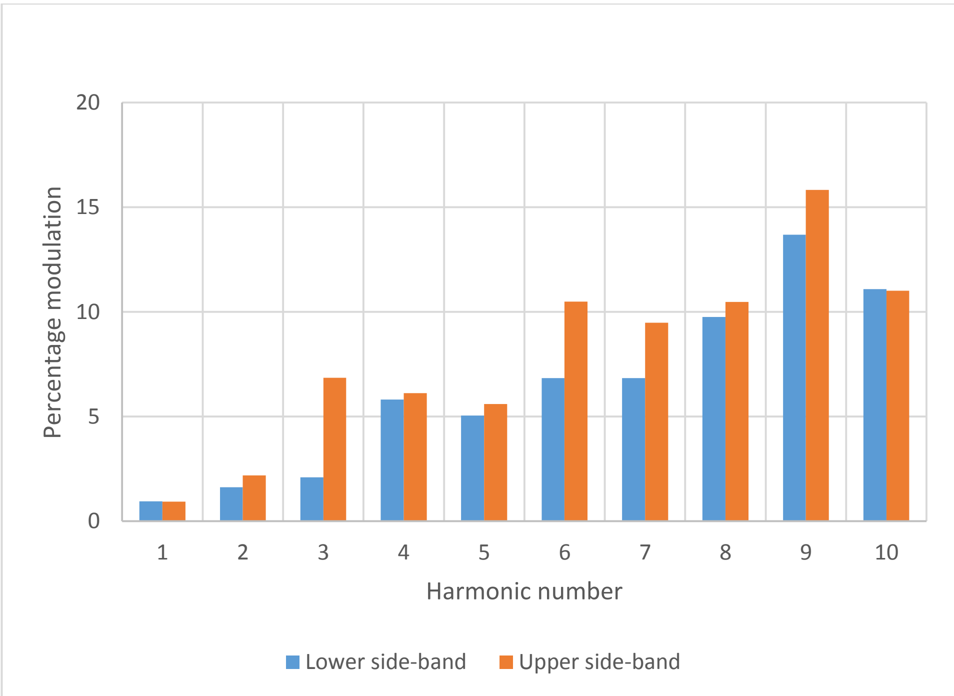


Figure 89 Blower 6 percentage amplitude modulation

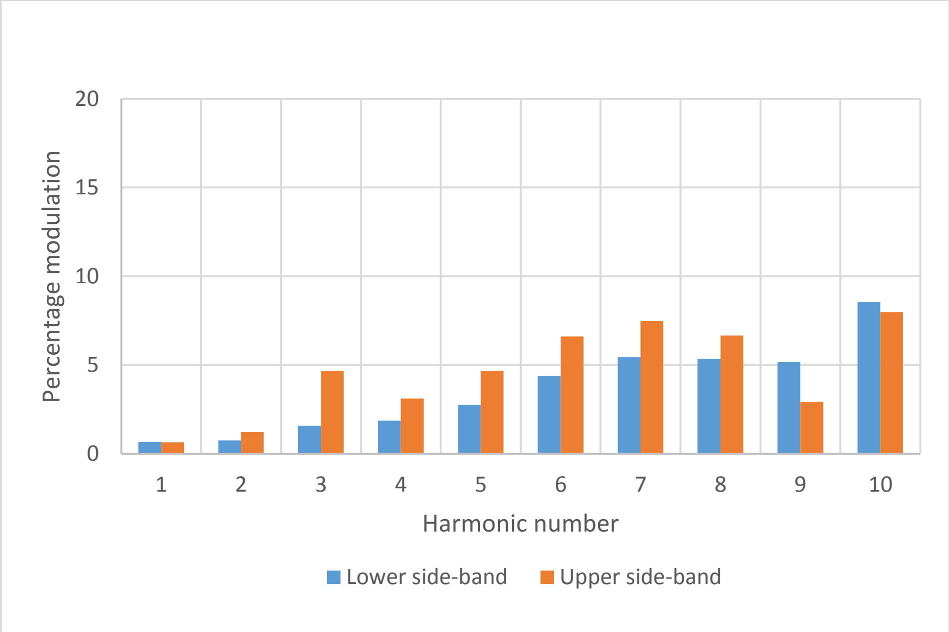


Figure 90 Blower 7 percentage amplitude modulation

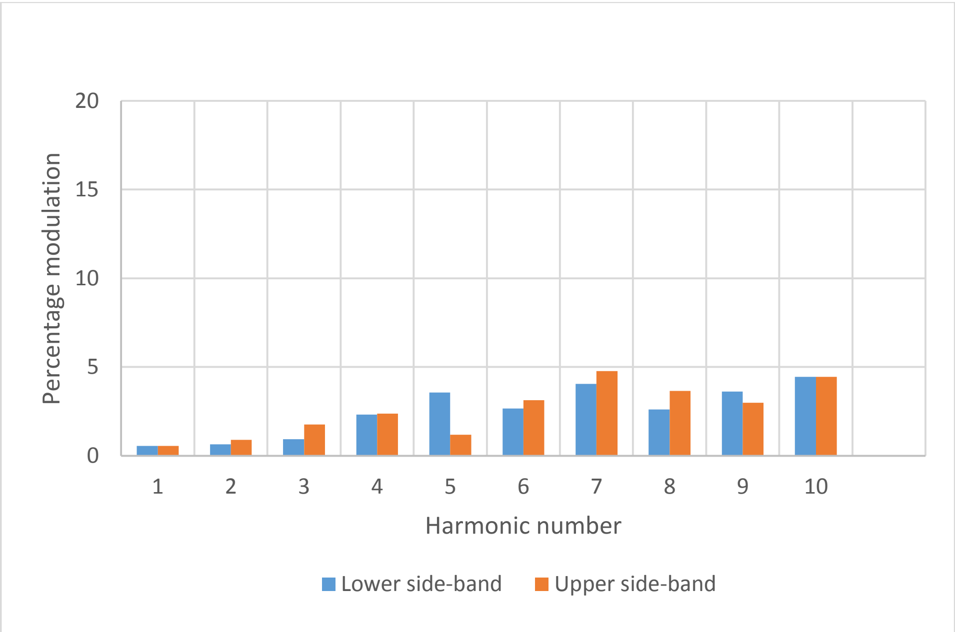


Figure 91 Blower 8 percentage amplitude modulation

8.6 Discussion

The pipe organ survey shows that all the organs had some form of electric blower with more having 3000 rpm (58%) than 1500 rpm blowers. Details of all the blowers are shown in Table 12 and various parameters are shown in Figures 73 to 76. Blower 6 produces the most organ pipe flutter which may be as a result of the peripheral speed, Figure 75, or the width of the impeller, Figure 76. For a given static pressure the peripheral speed is the same for low and high-speed units and is made possible by having different diameter impellers. Blowers 5 & 6 have the two widest impellers and produce the highest levels of amplitude modulation and organ pipe flutter. This may be coincidental and requires further investigation. Blower impeller design and cut-off details are investigated in the next Chapter.

The results for reservoir top vibration levels for all 8 blowers are shown in Figures 77 to 81. Generally, 1500 rpm units have lower levels of reservoir top vibration than 3000 rpm units. What is very surprising is the good performance of the high-speed blower 4 which has the most primitive and simplistic design of all the units tested and compares favourably with the 1500 rpm blower 1. This may be due to a $\Delta r/R$ ratio of .23 which is examined in the next Chapter.

Figure 81 compares two blowers that have been used in many of the experiments, blower 1 (1500 rpm) and blower 6 (3000 rpm). Clearly the 1500 rpm unit produces substantially less disturbance of the reservoir.

The comparison of each blower, Figures 82 & 83, shows that changing the blower changes the overall frequency spectrum, (window 2), to an extent that the change can be audibly detected. This script can also be used as a first step to determine the presence and level of any organ pipe flutter.

The percentage of amplitude modulation for each of the 8 blowers is shown in Figures 84 to 91. The Figures show that blower 1 and blower 4 produce the minimum amplitude modulation and blower 6, the maximum. Blowers 1 & 4 have very low levels of amplitude modulation up to the 5th harmonic, and blowers 6 & 7 are substantially higher from the 3rd harmonic.

Blower 8, which is not designed for organ building applications, gave some very unexpected results and produced less 10Hz reservoir vibration than some of the other specially designed pipe organ blowers. Figure 77 shows the high reservoir vibration and noise levels in the 1/3 octave bands above 63Hz which makes this blower totally unsuitable for organ building applications.

8.7 Conclusions

From the results, it appears that each blower has a unique acoustic finger print. Normally, for general industrial applications this is not a problem where capacity and pressure are the main design criteria.

It is important for the pipe organ builder to be aware that the move to smaller 3000 rpm blowers from 1500 rpm units, that have time expired, may increase reservoir top vibration levels and organ pipe flutter.

9 Blower impeller and cut-off tests

9.1 Preamble

In this Chapter, blower parameters that may excite reservoir resonance and organ pipe flutter, are investigated under “no flow” conditions, i.e. the only flow is due to any leakage from the reservoir and connecting duct. Under “no flow” conditions the mechanical efficiency of the fan is very low as shown in appendix A Figure 166

Chapter 8 has shown that each blower produces different levels of amplitude modulation and organ pipe flutter. Blowers 5 & 6 produce the highest levels of amplitude modulation and blowers 1 & 4 produce the lowest.

There are three basic sources of noise generated by centrifugal fans:

- Aerodynamic
- Electro-magnetic
- Mechanical

The electro-magnetic and mechanical noise have been addressed by pipe organ blower manufacturers by using special motors with sleeved bearings.

The aerodynamic noise is produced by the rotating impeller and vortex action and was first examined by Lighthill [25] [26]. The vortex component is due to the shedding of vortices from the trailing edge of the fan blades and has a broad band frequency spectrum.

There is also a rotational component that produces a series of impulses given to the air by the blades as they pass a given point in the fan housing. This noise consists of discrete frequencies at the fundamental blade frequency and its harmonics. The level of the blade frequency is mainly determined by the impeller cut-off clearance. This is the minimum distance between the impeller and the case at the blower outlet and is shown in Figure 94.

9.1.1 Blade tones

The blade frequency is defined as: impeller speed (revs per second) x number of blades.

For example, a 1500 rpm motor with a 12- blade impeller:

Blade frequency = 1500/60 x 12 = 300Hz

The corresponding blade frequency for a 3000 rpm motor is 600Hz.

Further information regarding fan types and designs is contained in Appendix A.

With the increasing use of air conditioning systems in private and public spaces, some method of determining the noise performance of fans used in these applications was necessary. This was initially addressed by Beranek et al [27] [28], who constructed an apparatus for predicting ventilation system noise. The apparatus used by Beranek, is shown in Figure 92. The main feature of the apparatus is that the test duct is terminated anechoically.

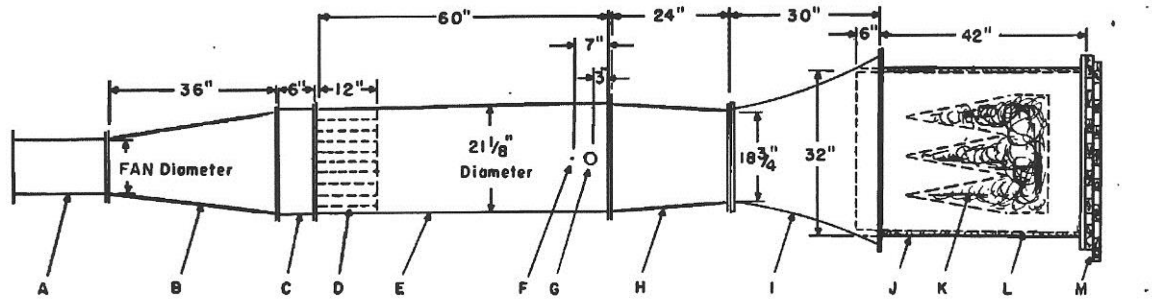


FIG. 1. Fan and duct system used in the experiments discussed in this paper. A—fan, B—conical adapter, C—canvas coupling, D—straightening vanes, E—measuring section, F—manometer fixture, G—microphone opening, H—adapter, I—exponential horn, J—acoustic termination, K—acoustic wedges 24 in.×24 in., L—fiberglas lining, M—back-pressure panels.

Figure 92 Details of the apparatus used by Beranek et al (After Beranek [27])

Figure 93 compares the sound power levels for axial flow and centrifugal fans. The centrifugal fan is characterised by the relatively large amounts of low frequency noise and the spectrum slopes off towards the high frequencies at approximately 5dB per octave. The axial flow fan has an almost flat frequency spectrum. The centrifugal fan produces more noise in the low frequency bands, and the axial flow fan produces more noise in the high frequency bands.

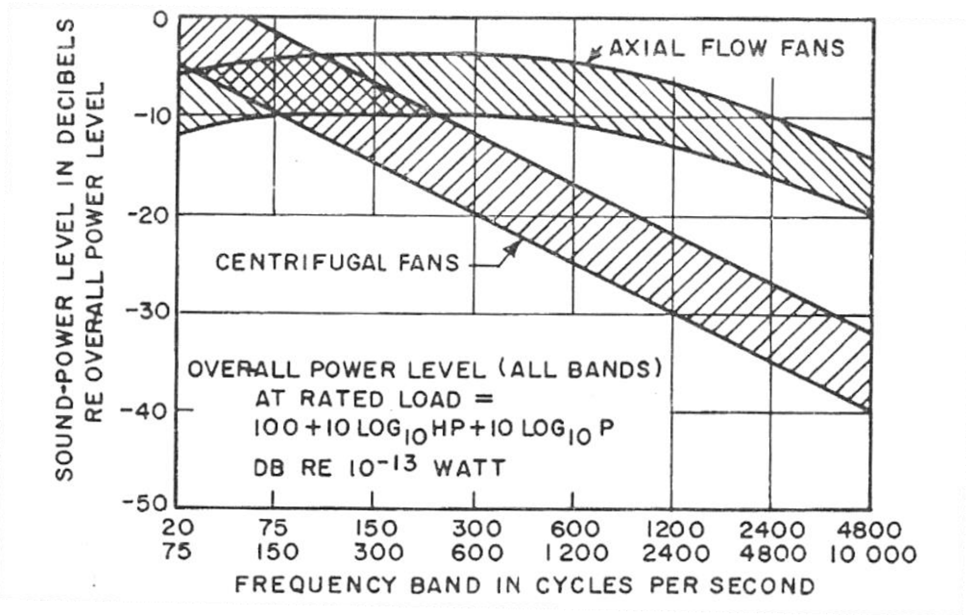


Figure 93 Comparison of axial and centrifugal fans (After Beranek [27])

Centrifugal fans with 3 or 4 blades have higher blade frequency levels than fans with 10 or more blades. It is therefore normal, for centrifugal fans to have between 10 and 16 blades. Axial flow fans are not able to develop the air pressure needed for pipe organ building applications, so the needs of the pipe organ builder must be met by centrifugal fans.

Embleton [29] and Neise [30] [31] showed that the cut-off clearance, shown in Figure 94, had a significant effect on the level of the passing blade frequency. The cut-off clearance should be made as large as practically possible without affecting the performance of the fan. The siren is an extreme case where the cut-off clearance is very small, so that an extremely high level of blade tone is produced.

Leidel [32] also showed that cut-off clearance strongly influenced the sound power level of the blade frequency. The influence of the cut-off radius on the level of the blade frequency was less significant than the cut-off clearance. The lowest sound power level was achieved for a cut-off clearance of $\Delta r/R = 0.25$ and a cut-off radius of.$r/R = 0.2$.

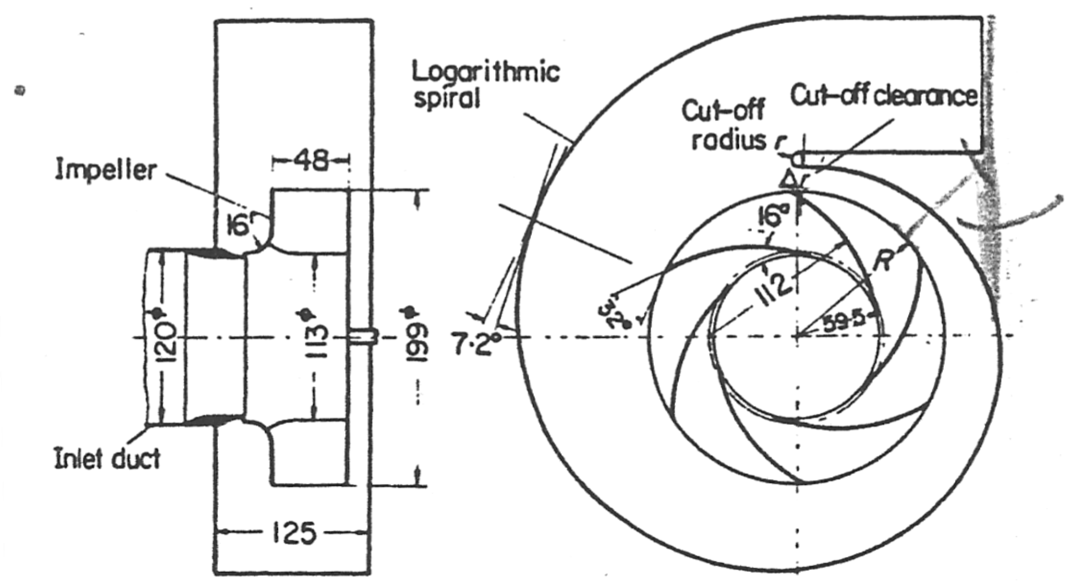


Figure 94 Schematic of fan showing cut-off clearance and cut-off radius
(After Leidel [32])

The cut-off clearance (Δr) for blowers 1 to 7 is shown in Table 13 together with ($\Delta r/R$). The leaf blower has been omitted because the impeller is not of the closed design.

Blower ref	Impeller radius R mm	Cut-off clearance Δr mm	Δr/R
1	533/2	115	0.43
2	533/2	45	0.17
3	558/2	50	0.18
4	260/2	30	0.23
5	312/2	65	0.42
6	338/2	100	0.6
7	317/2	50	0.31

Table 13 Blower cut-off comparison details

9.2 Methodology

Using the apparatus and measuring equipment described in Chapter 4, the presence and level of blade frequencies and their effect on reservoir top vibration under "no flow" conditions, is examined using the blowers listed in Tables 12 &13.

The microphone is placed inside the single rise reservoir and two conditions are examined and measurements taken.

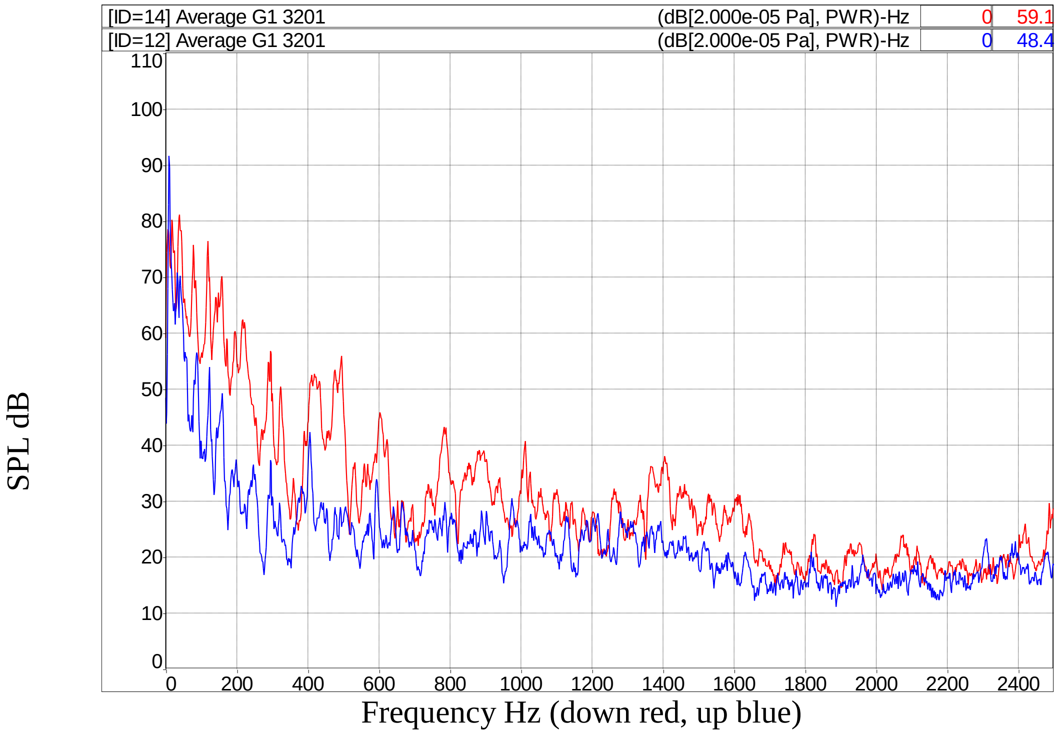
Condition 1. Reservoir down. In this position there is a clear path from the blower to the microphone.

Condition 2. Reservoir up. This is the normal working position and the internal control valve interrupts the direct path of the blower and the microphone.

The first measurements were taken using the fixed speed blowers 1, 2, 3, 4, 5, 6 & 7. A further set of measurements was taken using blower 6 fitted with a speed controller, which allows discrete motor speeds to be selected by varying the mains frequency. Mains frequencies of 35, 40 & 45Hz were selected with corresponding motor speeds of 2100, 2400 & 2700 rpm. Blower 6 has 12 main blades and four inter-blades producing blade frequencies of 420, 480 & 540Hz for mains frequencies of 35, 40 & 45Hz. The corresponding inter-blade frequencies are: 1680, 1920 & 2160Hz. Finally, a further set of measurements was taken using blower 6 with a motor speed of 50Hz, a blade frequency of 600Hz, but with the cut-off reduced from 100mm to 25mm, by inserting a wooden wedge.

9.3 Results

The microphone results for the fixed speed blowers 1, 2, 3, 4, 5, 6 & 7 with the reservoir "down" and "up" are shown in Figures 95 to 100.



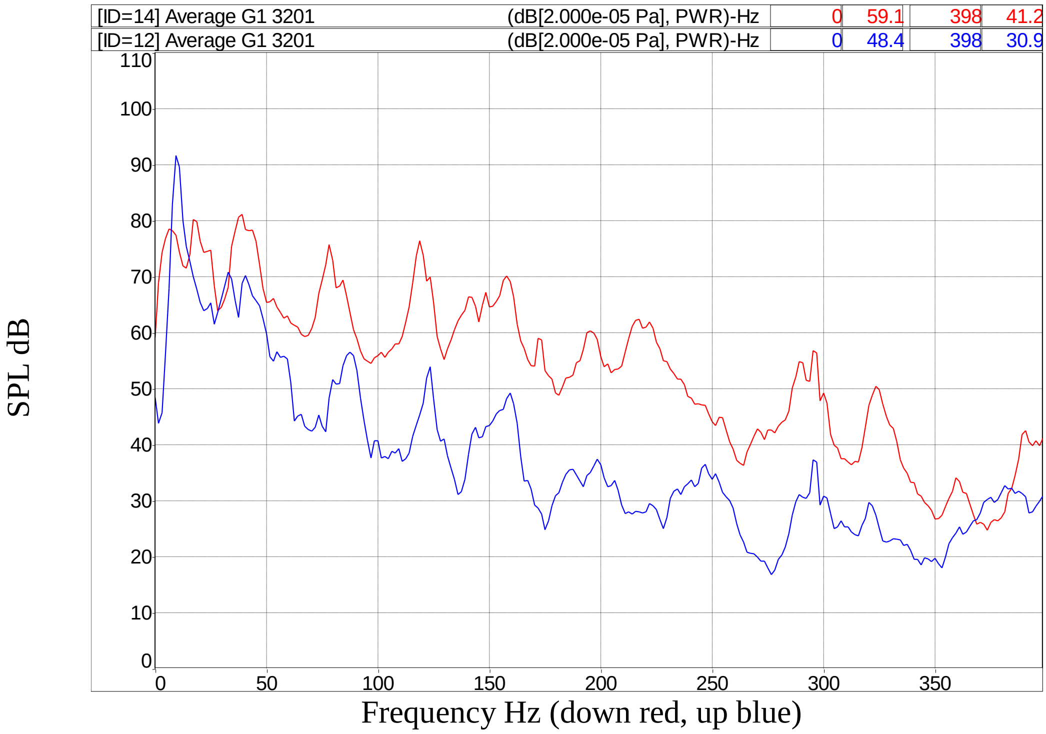
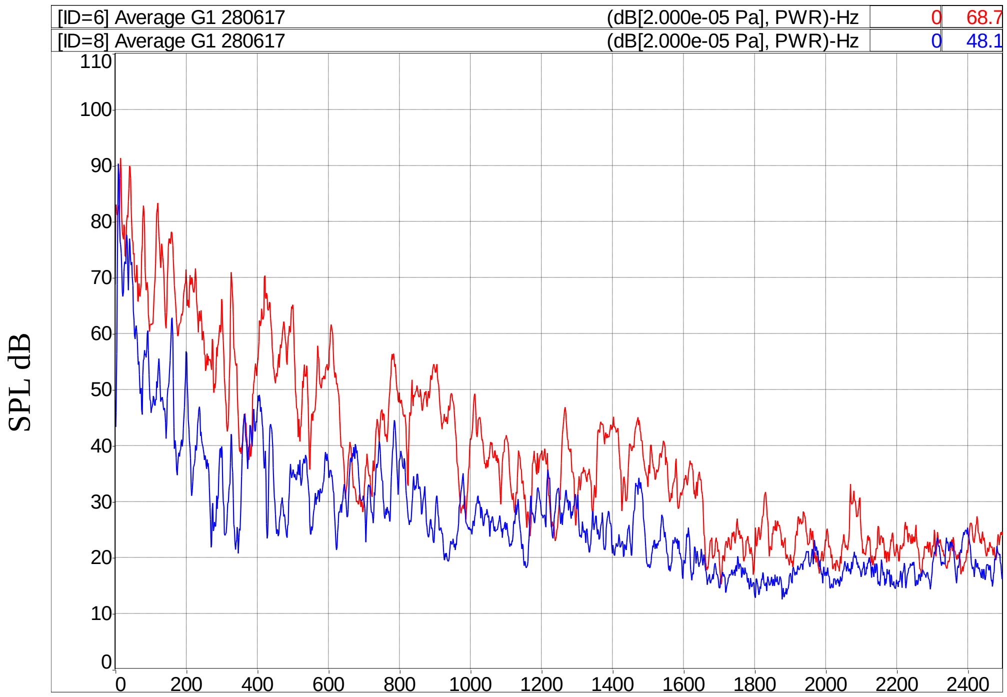
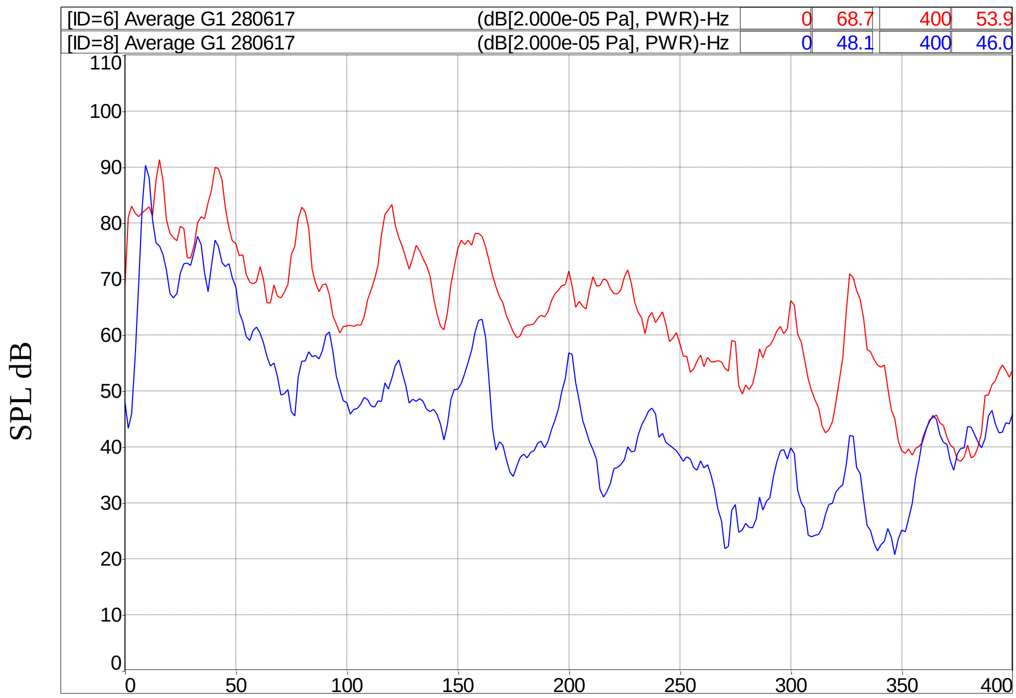


Figure 95 Blower 1 blade frequency 300Hz

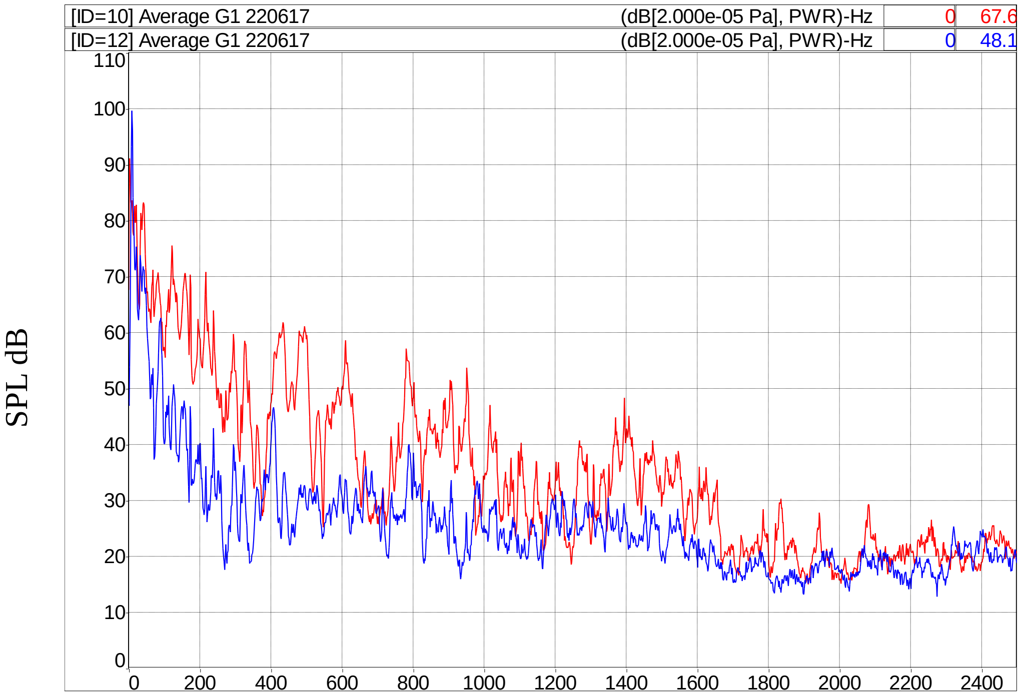


Frequency Hz (down red, up blue)

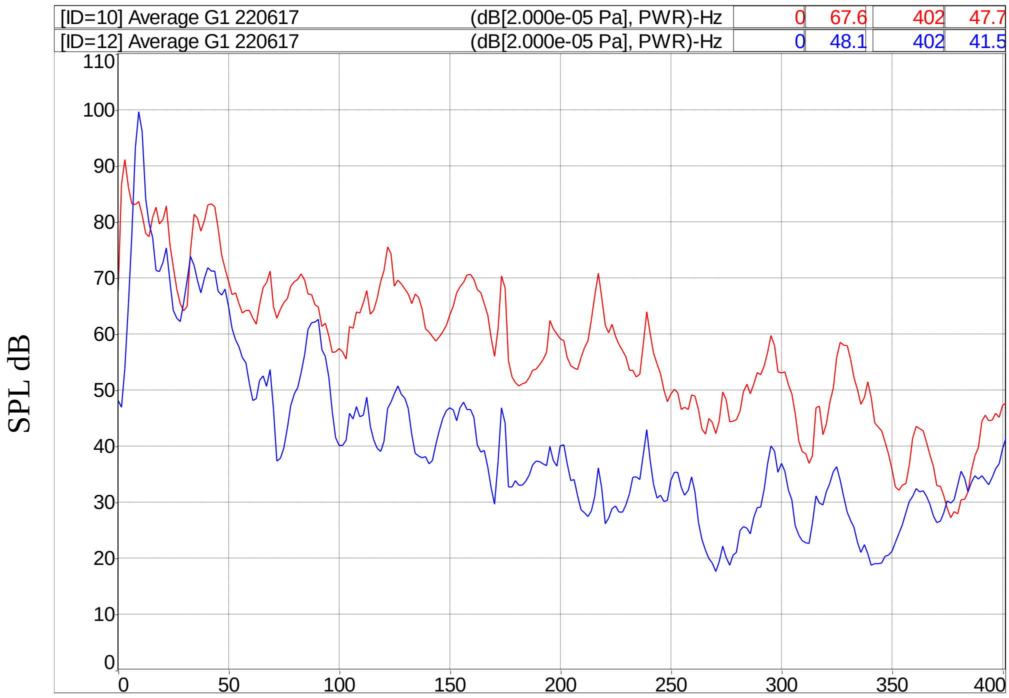


Frequency Hz (down red, up blue)

Figure 96 Blower 2 blade frequency 200Hz

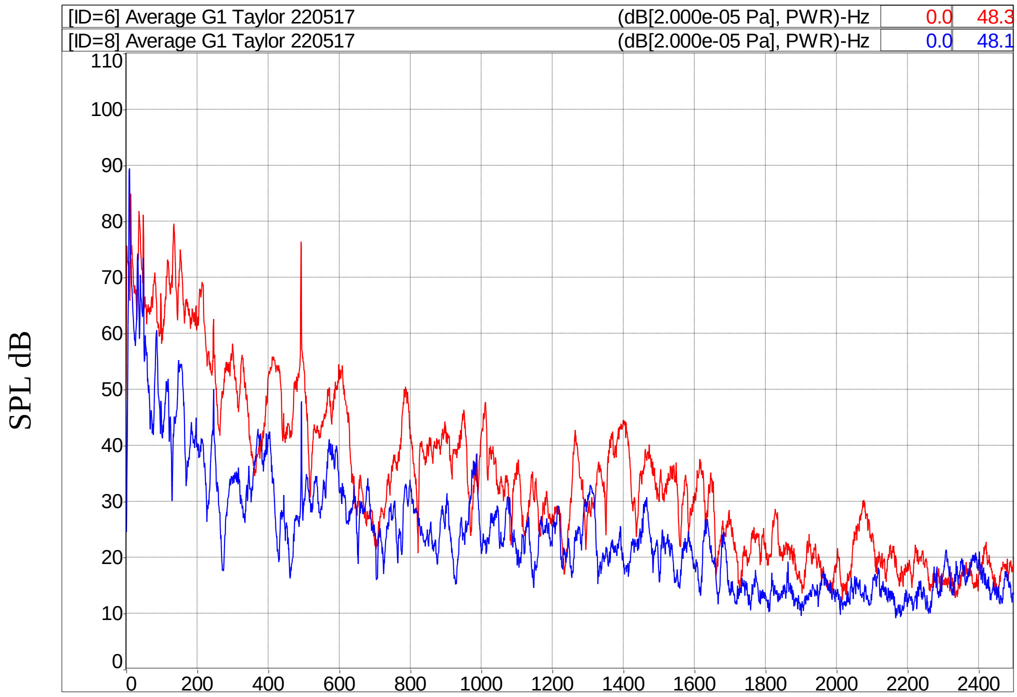


Frequency Hz (down red, up blue)

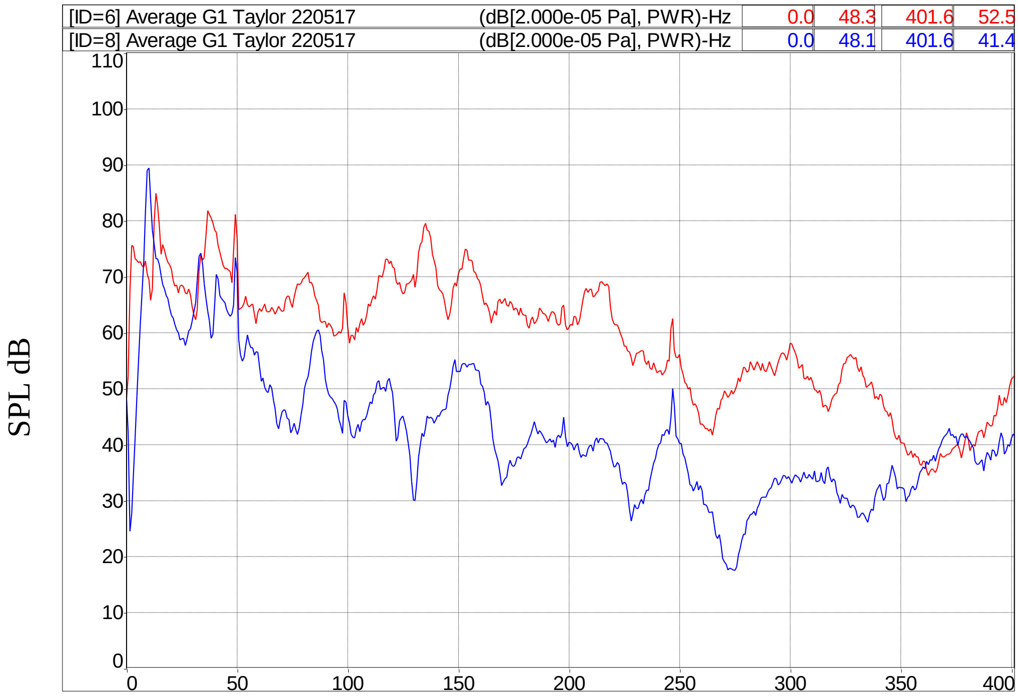


Frequency Hz (down red, up blue)

Figure 97 Blower 3 blade frequency 172Hz

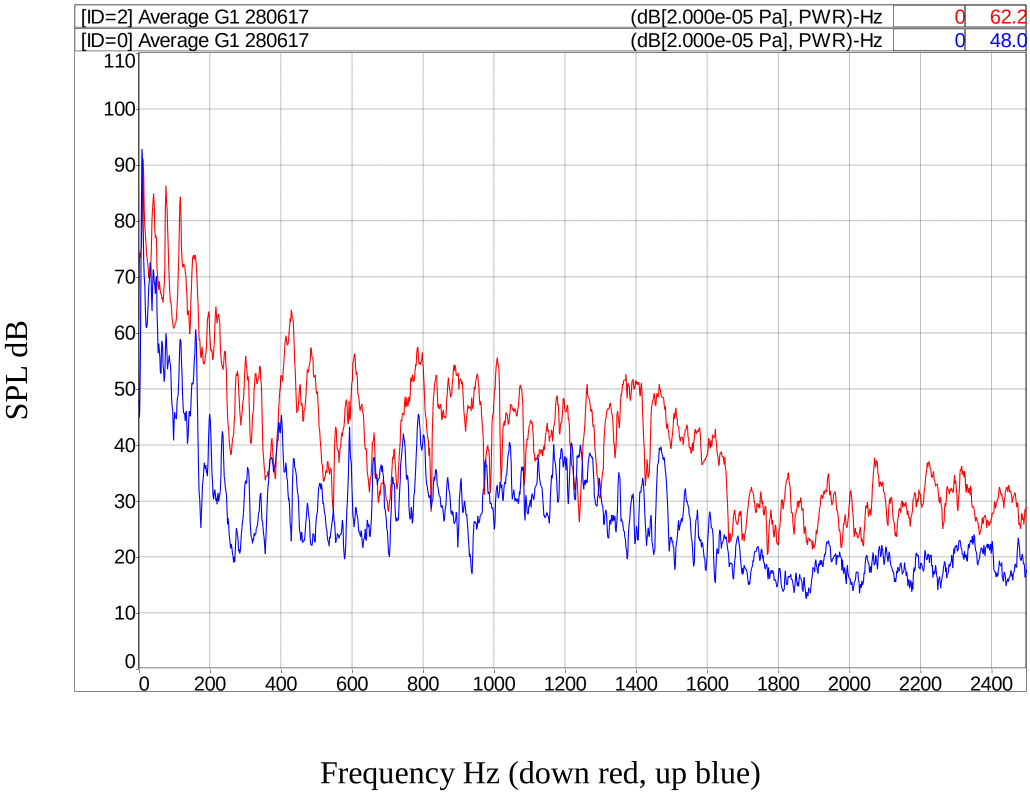


Frequency Hz (down red, up blue)



Frequency Hz (down red, up blue)

Figure 98 Blower 4 blade frequency 500Hz



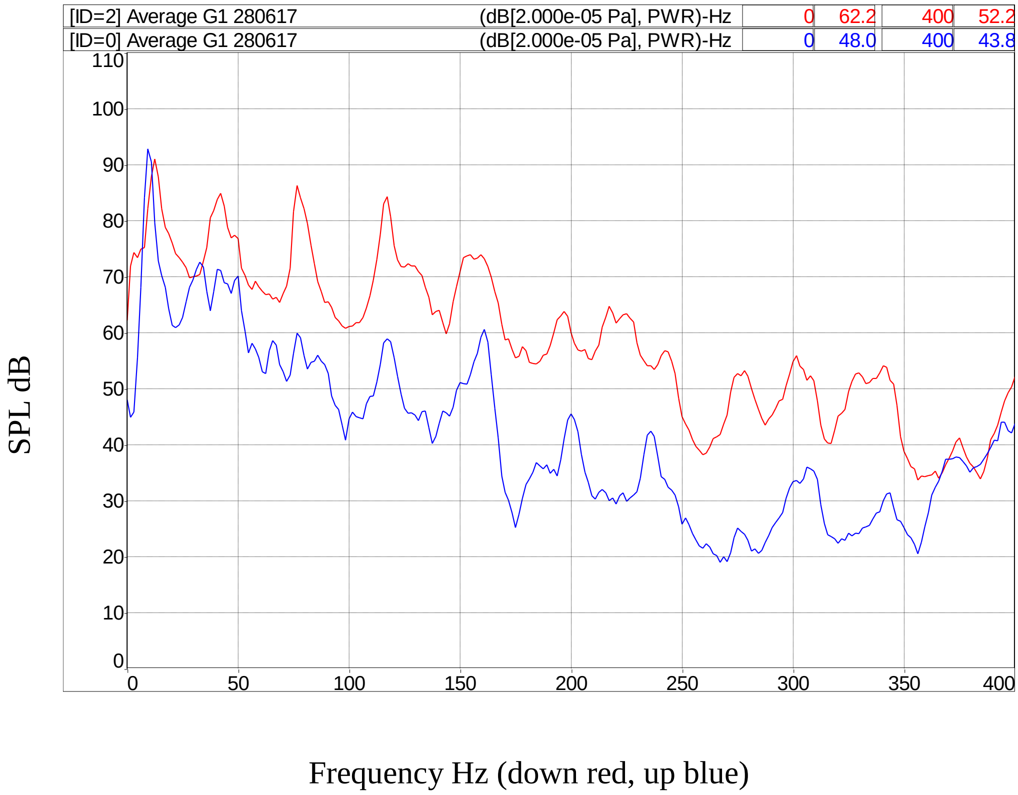
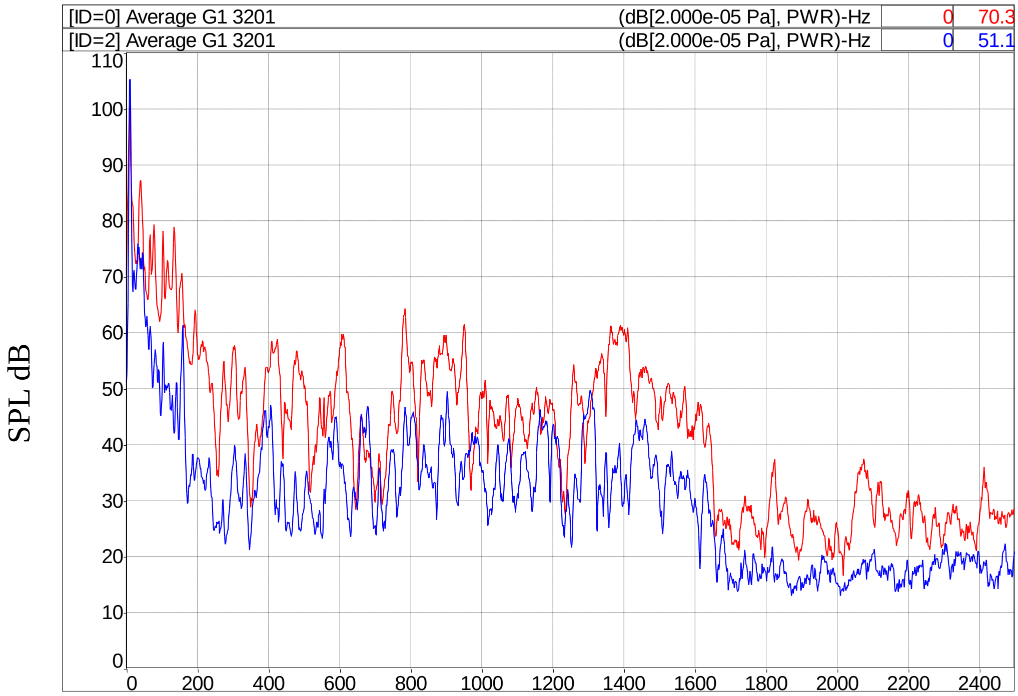
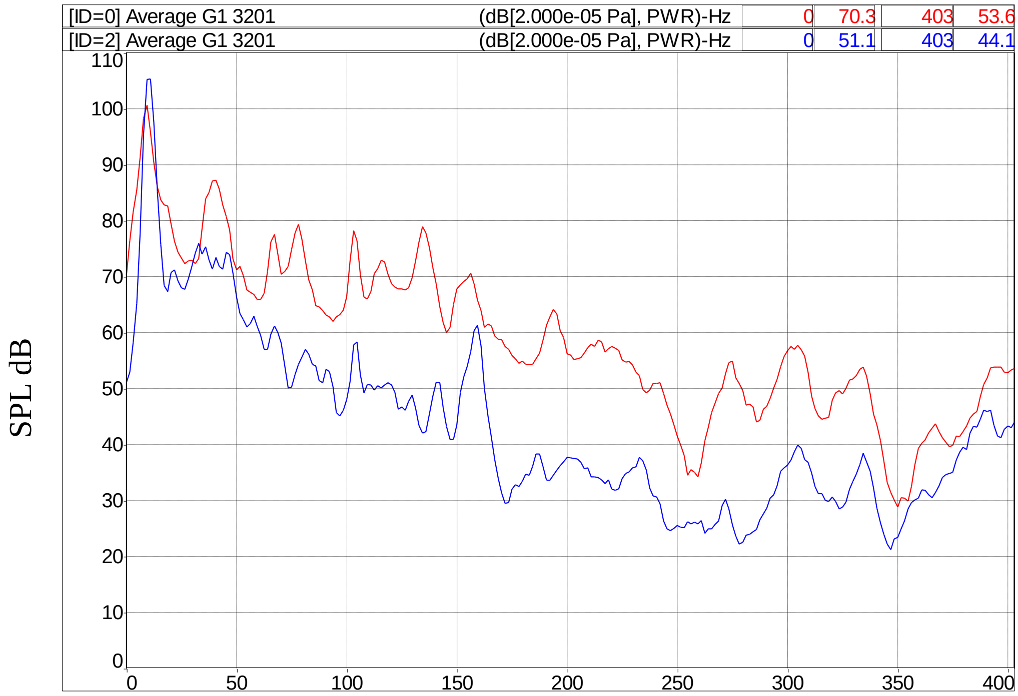


Figure 99 Blower 5 blade frequency 600Hz

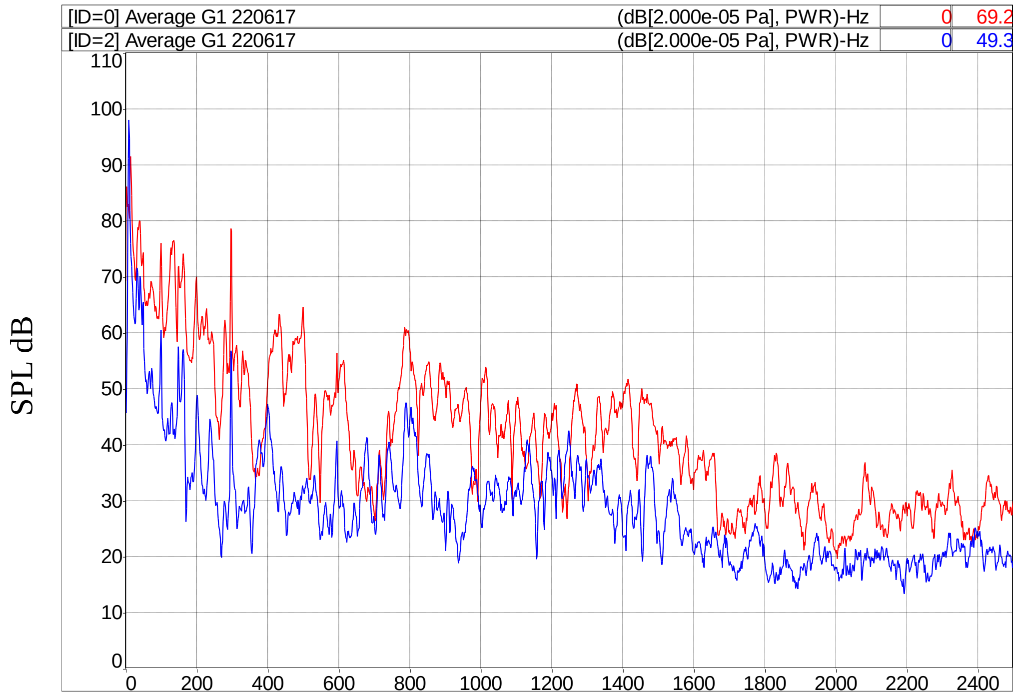


Frequency Hz (down red, up blue)

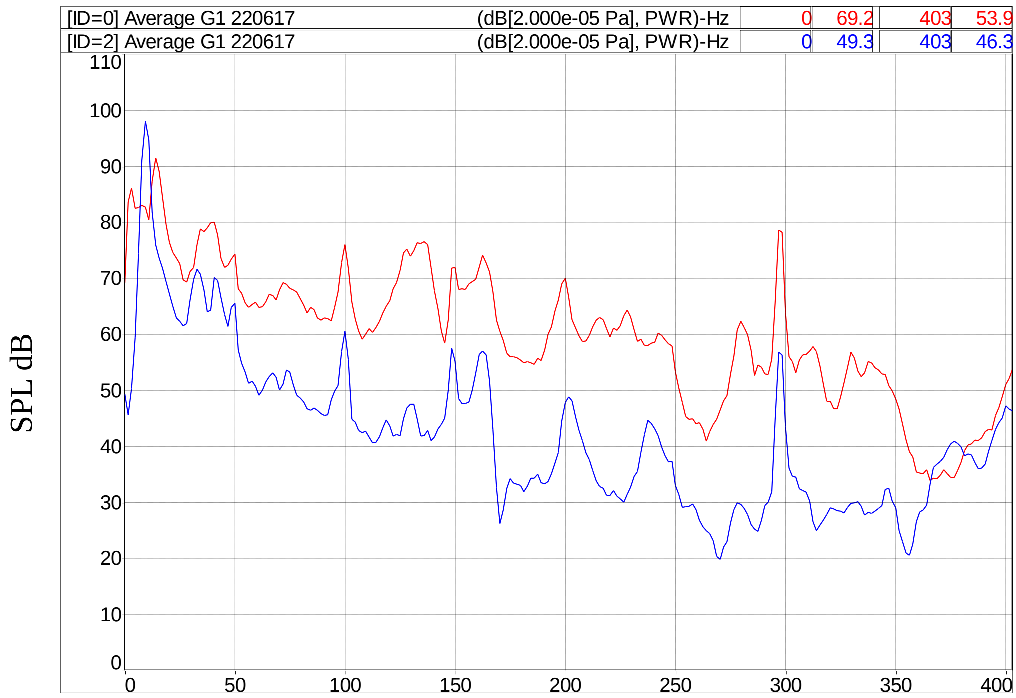


Frequency Hz (down red, up blue)

Figure 99A Blower 6 blade frequency 600Hz



Frequency Hz (down red, up blue)



Frequency Hz (down red, up blue)

Figure 100 Blower 7 blade frequency 300Hz

The microphone results for blower 6 and motor speeds with blade frequencies of 420, 480 & 540Hz, with the reservoir up and down, are shown in Figures 101 to 106.

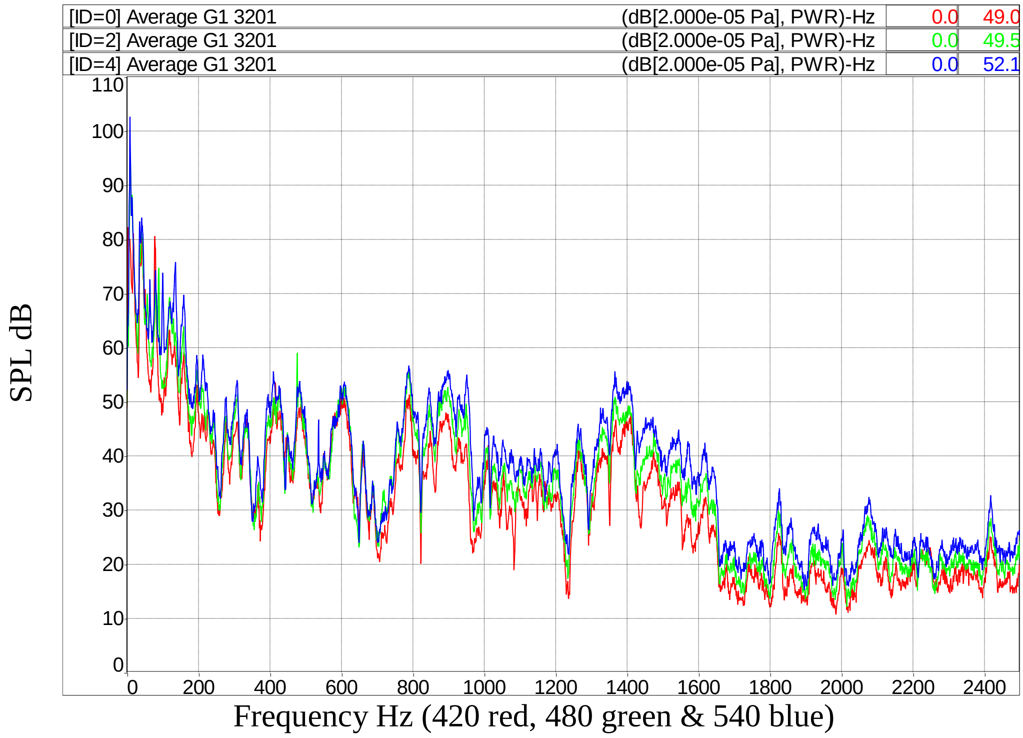


Figure 101 Blower 6 microphone frequency spectrum reservoir “down”

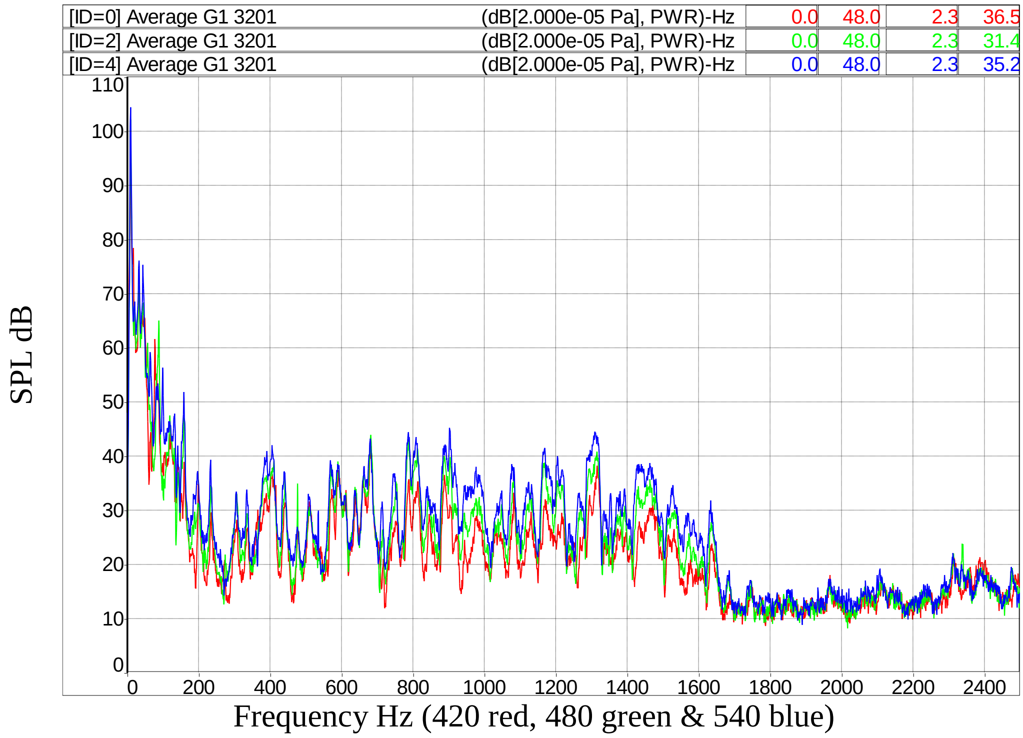
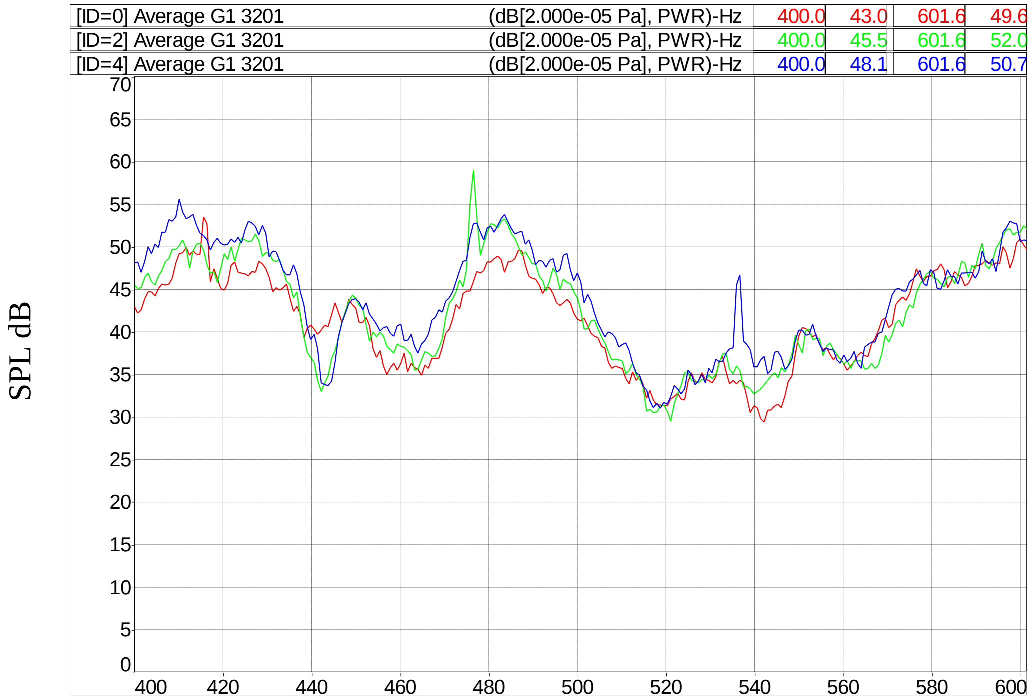
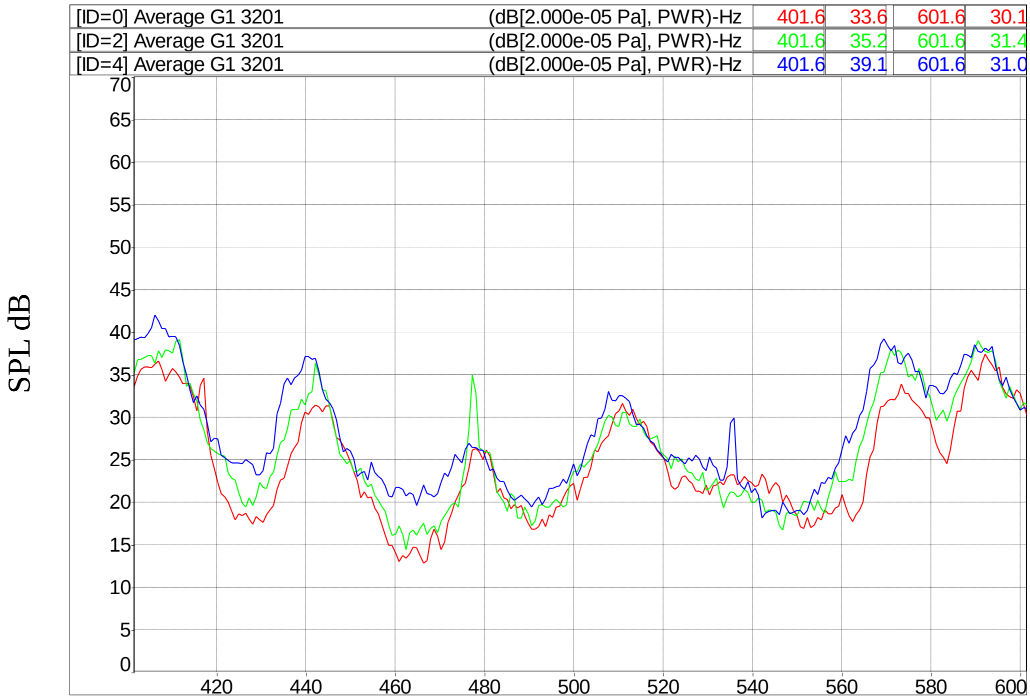


Figure 102 Blower 6 microphone frequency spectrum reservoir “up”



Frequency Hz (420 red, 480 green & 540 blue)

Figure 103 Blower 6 zoom frequency spectrum reservoir "down" 400 to 600Hz



Frequency Hz (420 red, 480 green & 540 blue)

Figure 104 Blower 6 zoom frequency spectrum reservoir "up" 400 to 600Hz

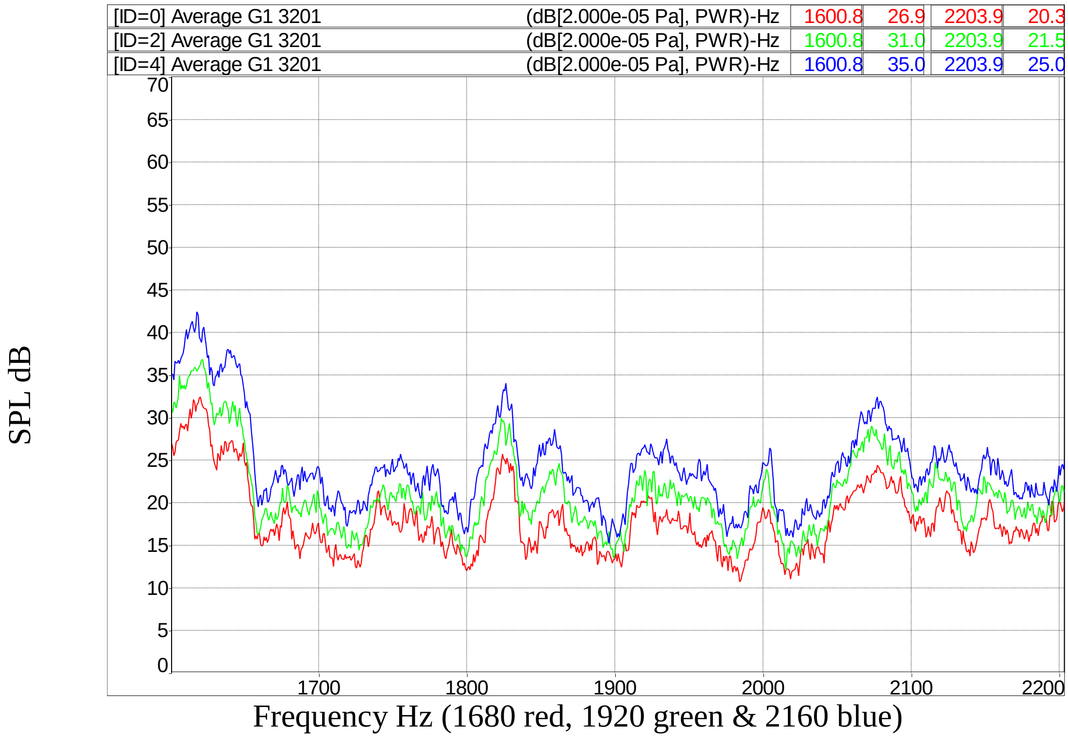


Figure 105 Blower 6 zoom frequency spectrum reservoir "down" 1600 to 2200Hz

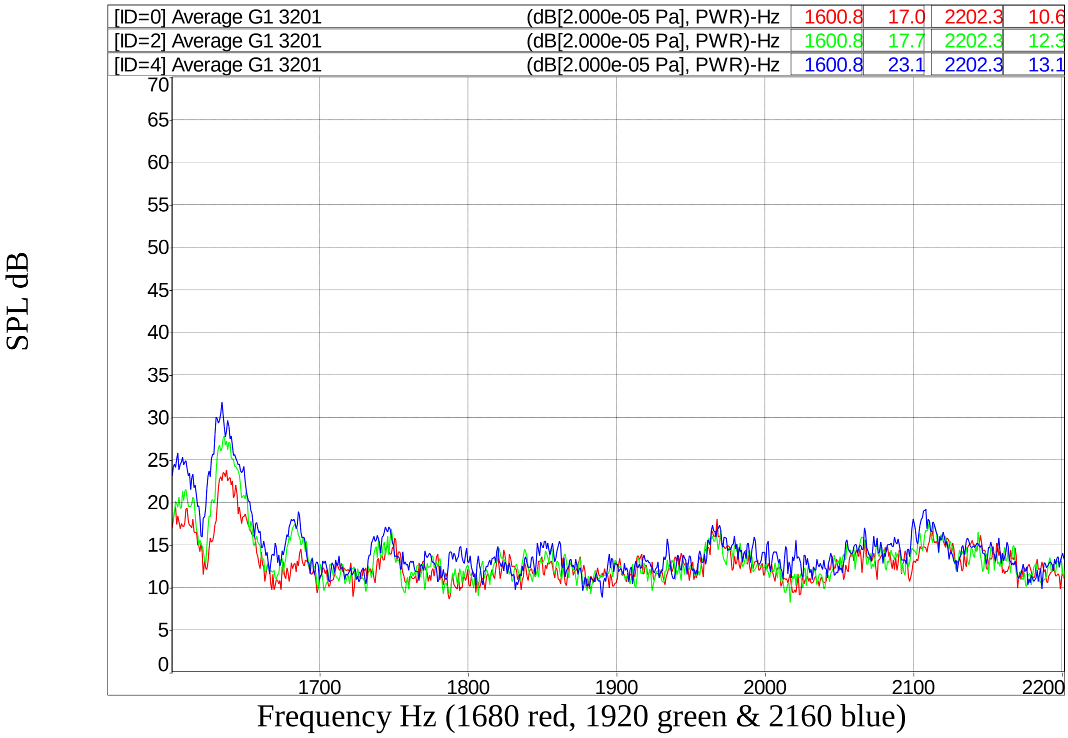


Figure 106 Blower 6 zoom frequency spectrum reservoir "up" 1600 to 2200Hz.

The corresponding reservoir top vibration levels for blower 6 are shown in Figures 107 & 108.

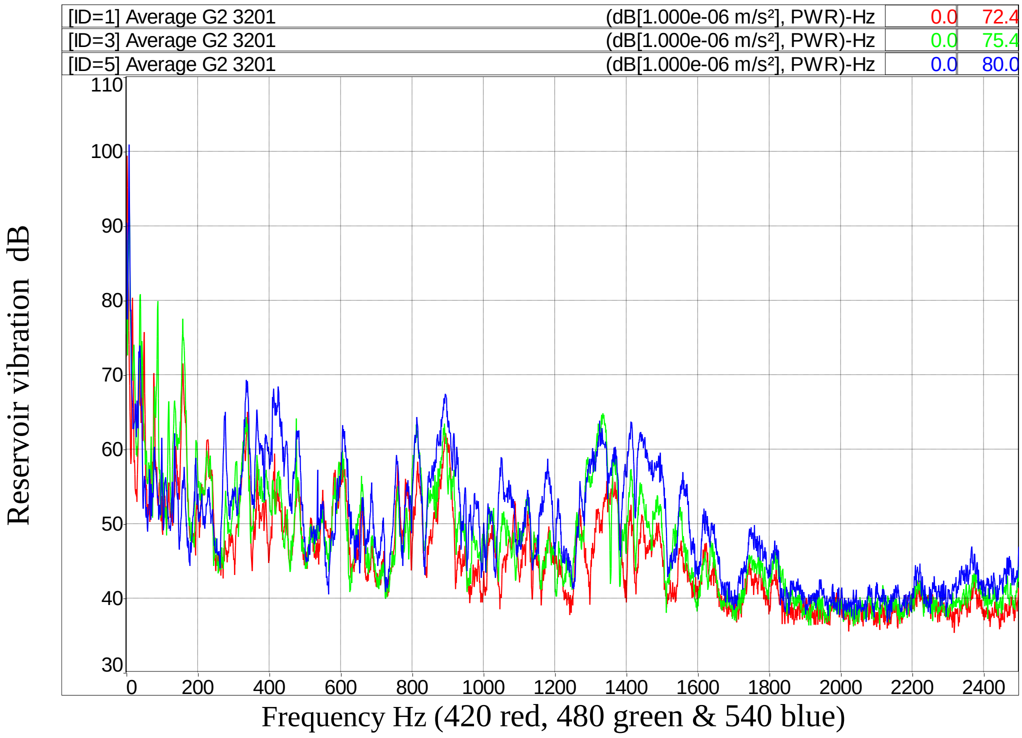


Figure 107 Blower 6 frequency spectrum of reservoir top vibration, reservoir “down”

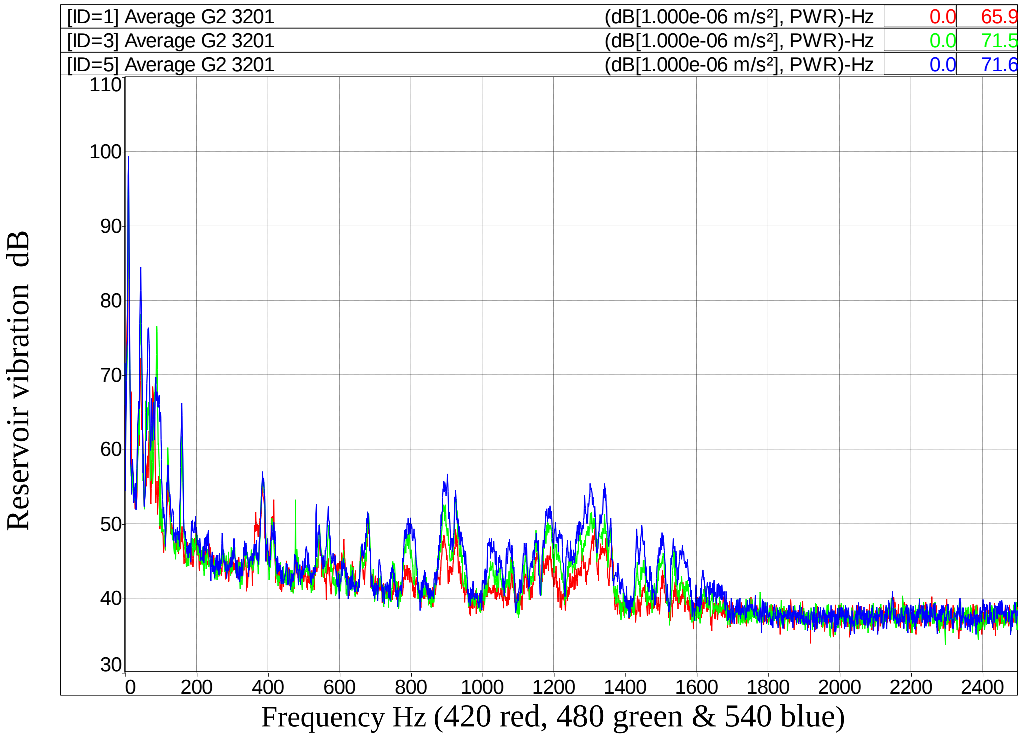
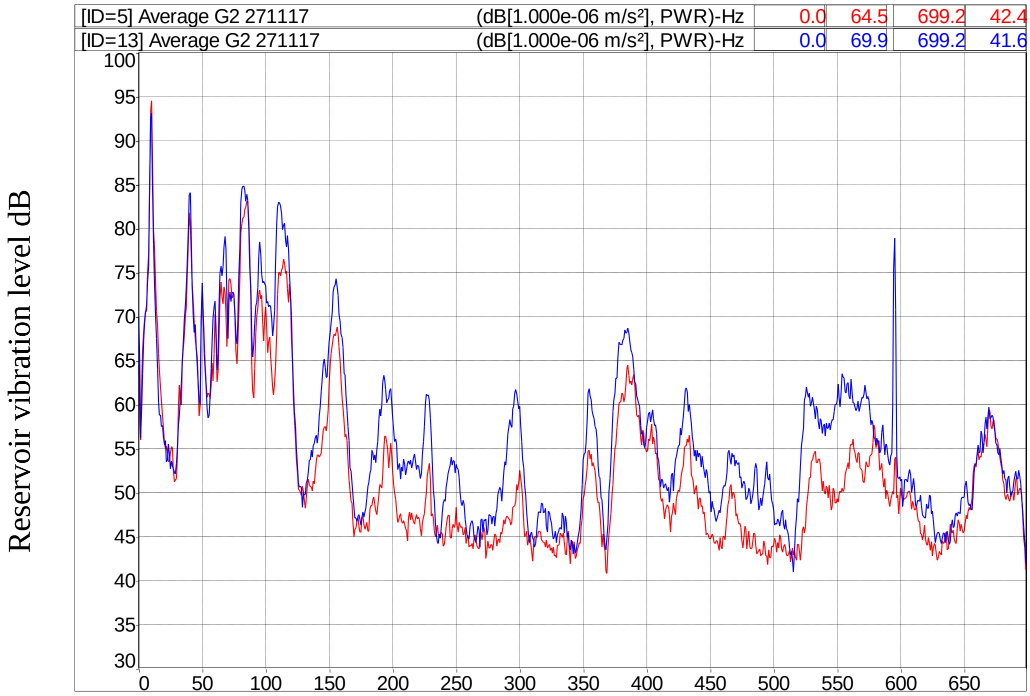
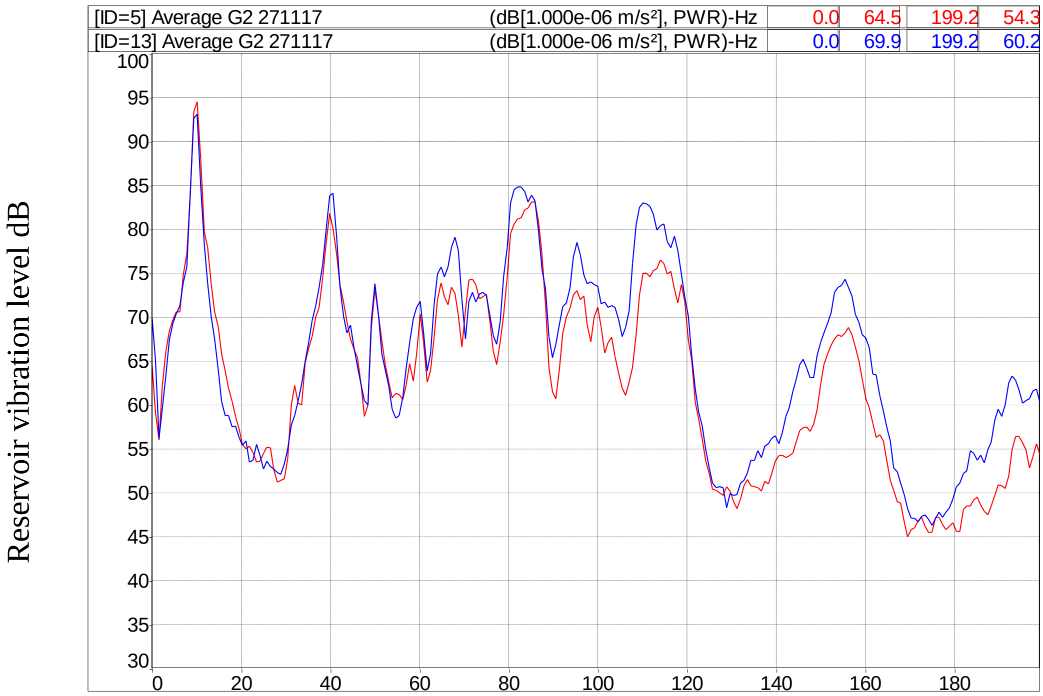


Figure 108 blower 6 frequency spectrum of reservoir top vibration, reservoir “up”

The reservoir top vibration and side-band levels for cut-off distances of 100 & 25mm are shown Figures 109 &110

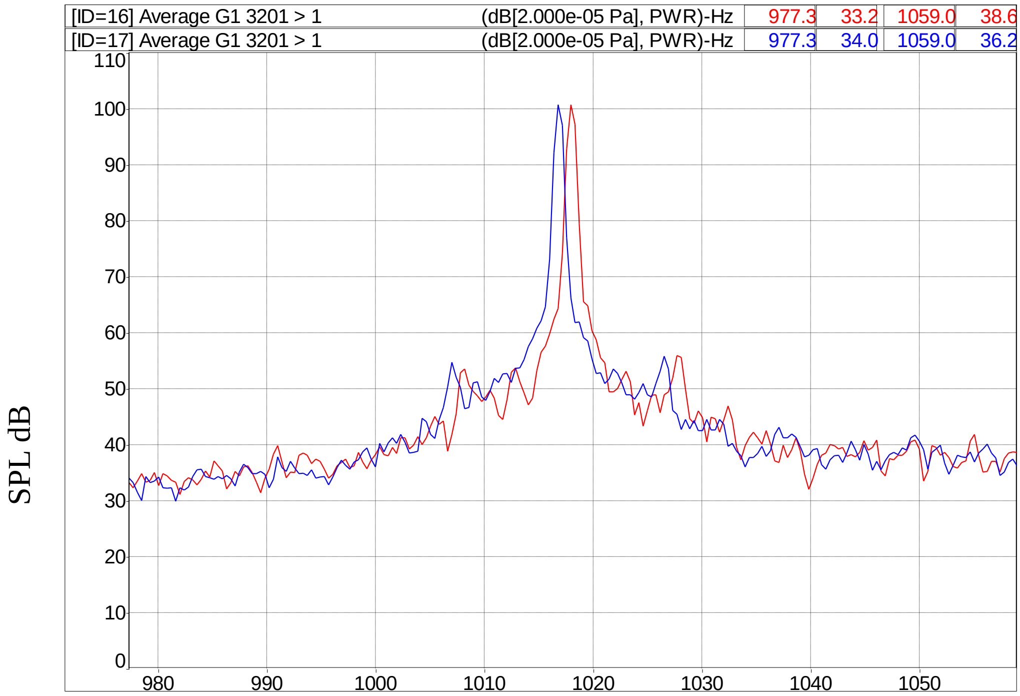


Frequency Hz 100mm red 25mm blue cut-off



Frequency Hz 100mm red 25mm blue cut-off

Figure 109 Blower 6 reservoir top vibration levels for 100 & 25mm cut-off



Frequency Hz 100mm red 25mm blue cut-off

Figure 110 Blower 6 side-band levels for 100 & 25mm cut-off

The passing blade SPL results for blowers 1 to 7 are shown in Table 14, together with the comparison for Δr/R in Figure 111.

Blower ref	Δr/R	SPL dB res down	SPL dB res up	Blade Freq Hz	Figure
1	0.43	54	32.5	300	95
2	0.17	70	56.5	200	96
3	0.18	70	47	172	97
4	0.23	76	48	500	98
5	0.42	56	42	600	99
6	0.6	60	45	600	99A
6a (35 Hz mains freq)	0.6	53.5	35	420	101- 106
6b (40 Hz mains freq)	0.6	58.5	35	480	101-106
6c (45 Hz mains freq)	0.6	46	30	540	101-106
7	0.31	78.5	57	300	100

Table 14 Blade tone levels

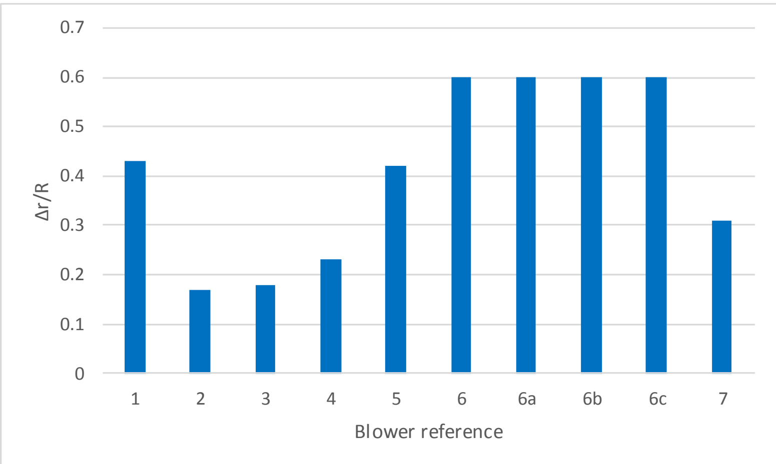


Figure 111 Δr/R for various blowers

The results for blowers 1 to 7 blade frequency levels are shown in Figure 112.

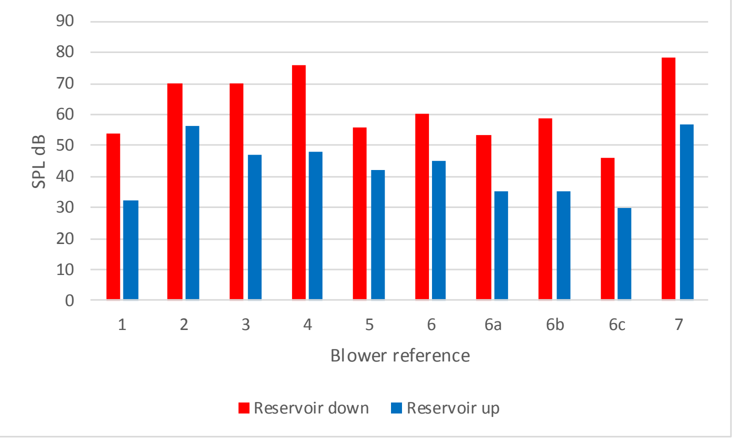


Figure 112 Blade frequency results for reservoir down and up.

The results for blowers 1 to 7 reservoir resonant frequency levels are shown in Table 15 and Figure 112A.

Ref	F down Hz	SPL down	F up Hz	SPL up
1	17	80.2	10	91.6
2	16	91.3	10	90.3
3	3	91.1	10	99.8
4	13.3	84.9	10	89.4
5	13	91	10	92.8
6	9.5	100.6	10	106.2
7	14	91.5	10	98

Table 15 Reservoir resonant frequency SPL

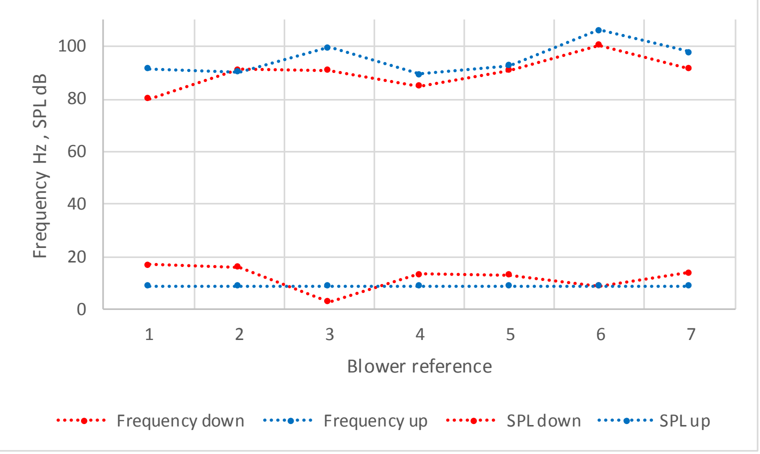


Figure 112A Blowers 1 to 7 resonant frequencies and SPL

9.4 Discussion

The use of a standard test arrangement, as described by Beranek et al, assumes that the outlet of the fan is terminated anechoically. Clearly this is not the case for organ blowing applications where the outlet is under pressure and the blower may be subjected to reflections from the reservoir control valve and other parts of the wind system.

Osborne[33] and his team, found that when the fan exit was pressurised, there appeared to be an absence of the passing blade frequency. Figures 103 & 104 show, that blade frequencies exist in a pressurised pipe organ wind system, but the levels are low due to the cut-off distance. The shift in blade frequency for the three motor speeds are shown in Figures 103 & 104.

From Table 14 the cut-off clearance varies from 45mm ($\Delta r/R = 0.17$) blower 2, to 100mm ($\Delta r/R = 0.6$) blower 6. Blower 4 has a ($\Delta r/R = 0.23$) which is very close to the optimum value of 0.25.

The level of blade frequencies for the 7 blowers is shown in Figure 112 for both reservoir "down" and "up" positions. In the active "up" position, the levels are attenuated by approximately 20dB. The maximum SPL (78.5dB) occurs with blower 7. Figures 98 & 100 show that blowers 4 & 7 produce very noticeable blade frequencies at 500Hz & 300Hz. with the reservoir "down" and "up". The results of adding a wooden wedge to reduce the cut-off distance of blower 6 from 100 to 25mm, are shown in Figure 109. Adding the wedge, increases the reservoir top vibration levels at 600Hz by 25dB and the blade frequency can now be heard. Figure 110 shows that there is no difference in the level of side-bands on the sounding test pipe, indicating that blade frequencies are not responsible for the amplitude modulation. This is supported by blowers 4, which has a cut-off distance of 30mm and produces a very strong blade frequency and one of the least amounts of amplitude modulation and organ pipe flutter.

Blower 6 has the largest cut-off clearance of 100mm ($\Delta r/R = 0.6$) and produces very low blade frequency levels, (60dB down) & (45dB up). Despite these very low levels of blade frequencies, this blower produces the greatest amount of amplitude modulation and organ pipe flutter. Figure 76 shows that blowers 5 & 6 have the widest impellers and are made by the same blower manufacturer with different outputs and pressure characteristics. This suggests, that impeller width may be a factor in the development of organ pipe flutter and needs further investigation. This can best be determined by the blower manufacturer using

CFD, followed by testing. Fans are normally made to geometrical similar designs Francis [34] Massey [35] and the Fan Laws are used to calculate volume flow and pressure.

Results showing the reservoir resonant frequency levels of blowers 1 to 7 are shown in Table 15 and Figure 112A. For blower 6, the frequency of the blower and connecting duct exactly match the resonant frequency of the reservoir and produce the maximum reservoir top vibration. Effectively, the system is tuned to produce the maximum amplitude modulation of the test apparatus.

Figures 105 & 106 show no blade frequencies at the inter-blade frequencies of 1680, 1920 & 2160Hz. The effects of inter-blades on organ pipe flutter are examined in the following Chapter.

9.5 Conclusion

The system can be de-tuned by changing the wind pressure (Figure 63) or by removing weights and adding springs (Figure 72A). Alternatively, the reservoir height can be changed (Figure 58).

The presence of blade frequencies has little effect on both the amplitude modulation of the organ pipe and organ pipe flutter. However, further research is needed to understand how impeller geometry can be improved to limit reservoir top vibration levels under “no flow” conditions.

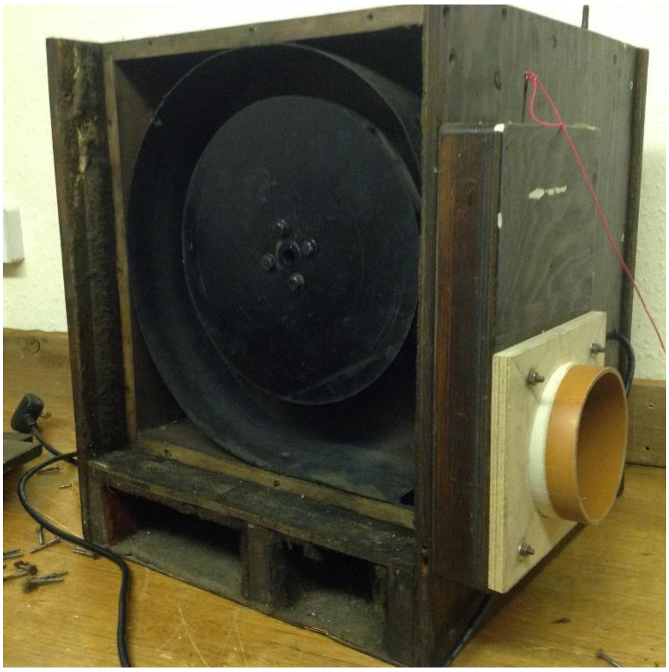
10 Impeller Inter-blades

10.1 Preamble

In Chapter 8 the number of main blades and inter-blades for each blower is shown in Table 12 and Figure 74. The number of main blades range from 6 to 12 and the total number of blades including inter-blades varies from 10 to 80.

The design of the impeller is closely guarded by fan manufacturers and there are few firms that make fans specifically for pipe organ blowing applications. Furthermore, there is very little technical information to support their designs. The inclusion of inter-blades is common in most modern organ blowers but rarely found in older units. The practice of adding inter-blades to the impellers of organ blowers seems to have started with the Swiss firm, Meidinger AG, and then copied by other manufacturers.

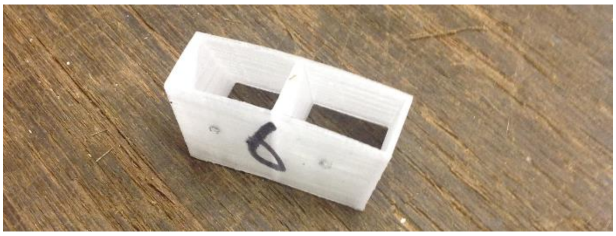
Fortunately, blower 4 has 10 main blades and no inter-blades. Its simple construction allows the inclusion of inter-blades between the 10 main blades. The blower housing and impeller are shown in Photograph 19. Photograph 20 shows the blower with the impeller removed.



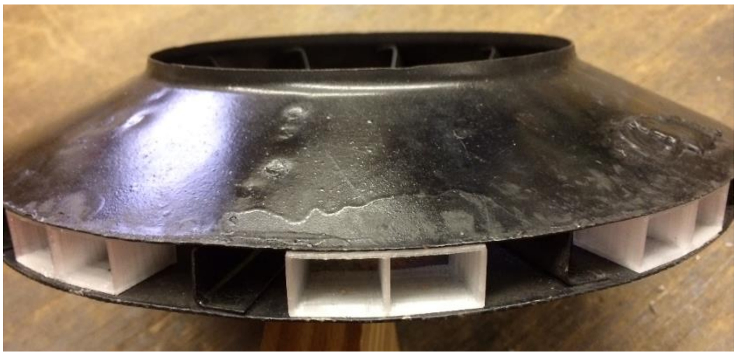
Photograph 19 Blower 4 with side panel removed showing the impeller

Photograph 20 Blower 4 with the impeller removed

The additional inter-blades, shown in Photograph 21, were 3D printed and held in place with two 12 BA screws. The impeller with the inter-blades fitted, making the total number of blades 40, is shown in Photograph22.



Photograph 21 New 3D printed inter-blade



Photograph 22 Existing impeller with the new inter-blades

10.2 Methodology

Using Blower 4, with and without inter-blades fitted, and the apparatus and measuring equipment described in Chapter 4, a comparison can be made, and the organ pipe amplitude modulation calculated in accordance with Chapter 5.

10.3 Results

The percentage amplitude modulation results for blower 4 without inter-blades is shown in Figure 113. The results for blower 4 with the new inter-blades is shown in Figure 114.

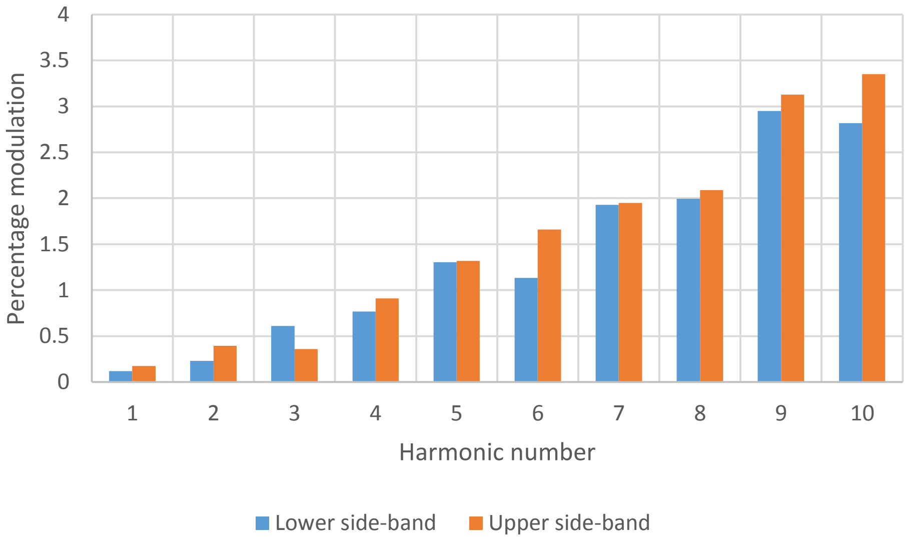


Figure 113 Percentage amplitude modulation without inter-blades

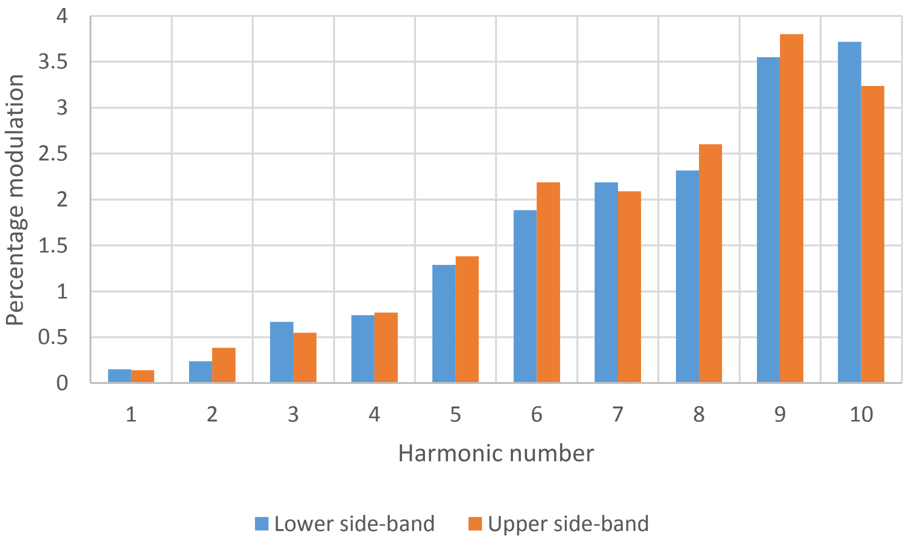


Figure 114 Percentage amplitude modulation with inter-blades

10.4 Discussion.

The percentage modulation for various harmonics shown in Figures 113 & 114 reveals that the addition of inter-blades has a slightly detrimental effect on the higher harmonics.

In main-stream fan manufacture, inter-blades are normally only used to increase the output of the fan. In this application, the air flow and pressure characteristics are of primary importance. No attempt has been made to determine if the inclusion of inter-blades has changed the air flow performance of the blower in accordance with BS848 using a standard air way.

10.5 Conclusion

Results show that the addition of inter-blades is found to be counter-productive. Further research using CFD is required to determine how blade geometry, number of blades, general impeller and case design can be improved to produce blowers that produce minimal organ pipe flutter, under "no flow" conditions.

11 Control valves

11.1.1 Preamble

The purpose of this Chapter is to determine if the type of control valve has any influence on the development and level of organ pipe flutter. Also, the air supply from the blower is selectively reduced, to simulate the reservoir top vibration levels that would be achieved using a feeder system of air generation.

Chapter 2 describes how a system of feeders was the only method of producing sufficient air for the organ, until the introduction of the centrifugal blower. The use of a centrifugal blower necessitates the use of some form of control valve to shut off the air supply from the blower, once the reservoir reaches its working height.

Figure 8 shows a multi-stage Kinetic blowing plant with low and high-pressure reservoirs. A guillotine valve is used to control the air supply to the low-pressure reservoir and a roller blind valve controls the air supply to the high-pressure reservoir.

The pipe organ survey, Figure 20, shows that the most popular control valve was the simple guillotine valve (53%) followed by the roller blind (36%) and the internal valve (11%).

Tables 5 & 6 show 36 organs surveyed have no flutter. Tables 16 & 17 show the distribution for the three valves for organs with no flutter (NF) - guillotine 12, roller blind 17 and internal 2.

These results indicate that the type of control valve has a significant role to play in the generation of organ pipe flutter. The measurements and analysis of this Chapter examine this hypothesis.

Tables 16 &17 show details of the control valves shown in Figure 20.

Ref	Guillotine	Internal	Roller B
1	x		
2	x		
3	x		
4	x		
5			
6	x		
7	x		
8	x		
9		x	
10	x		
11	x		
12		x	
13			x
14		x	
15	x		
16	x		
17			x
18	x		
19	x		
20	-		
21	x		
22	x		
23			
24	x		
25	x		
26	x		
27	x		
28	x		
29	x		
30			x
31			x
32			x
33			x
34		x	
35	NF		
36	NF		
37			NF
38			NF
39	NF		
40	NF		

NF indicates the organs with No Flutter.

Table 16 Control valves with no flutter organs 1 to 40

Ref	Guillotine	Internal	Roller B
41		x	NF
42			NF
43	NF		
44		NF	
45			
46	NF		
47		NF	
48			NF
49			NF
50	x		
51			
52			NF
53			
54	NF		
55			
56			
57	NF		
58			NF
59	x		
60	NF		
61		x	
62			NF
63	NF		
64	NF		
65	x		
66			x
67			NF
68	x		
69			NF
70			NF
71			x
72			x
73			NF
74			NF
75			NF
76	x		
77	x		
78	NF		
79			
80			NF
81			
82			NF
83			x

NF indicates the organs with No Flutter.

Table 17 Control valves with no flutter organs 41 to 83

11.2 Methodology

Using the test apparatus described in Chapter 4, the air supply from the blower is controlled by the internal control valve. The internal valve is then replaced by an external guillotine valve, and then an external roller blind valve. Finally, the air from the blower is restricted by two 22mm gate valves.

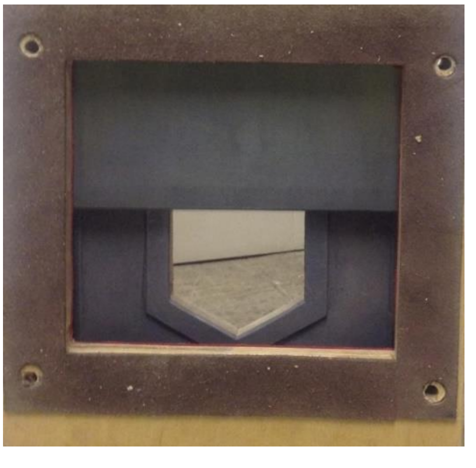
11.2.1 Internal control valve

The single rise test reservoir internal control valve is shown in Photograph 5.

11.2.2 Guillotine valve

The external guillotine control valve is shown in Photograph 23 and the profile of the valve opening is shown in Photograph 24.

Photograph 23External guillotine valve



Photograph 24 Guillotine valve opening profile

11.2.3 Roller blind valve

The external roller-blind control valve is shown in Photograph 25 and the profile of the valve opening is shown in Photograph 26.

Photograph 25 External Roller-blind valve

Photograph 26 Roller-blind valve opening profile

11.2.4 Gate vales

Two, 22mm isolating gate valves, shown in Photograph 27, replace the control valve and control the flow of air from the blower. Measurements were taken for two valves open, one valve open and one valve partially open to balance any air leakage.

Photograph 27 Two isolating gate valves

11.3 Results

11.3.1 Guillotine, roller-blind and internal valve results

The reservoir top vibration levels for the three control valves are shown in Figure 115.

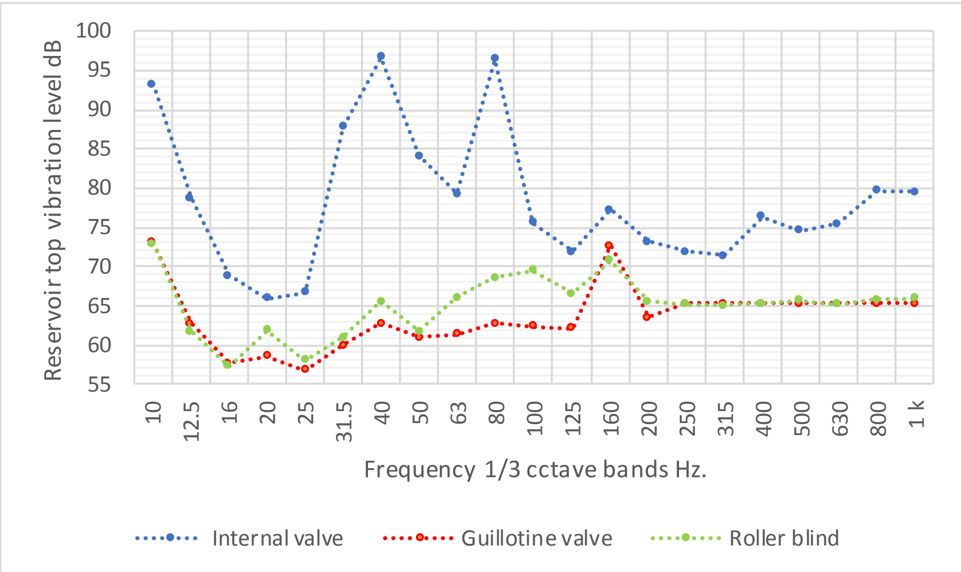


Figure 115 Reservoir top vibration results for the 3 control valves

11.3.2 Gate valve results

11.3.2.1 Organ pipe fundamental side-band results

Frequency spectra for the fundamental of the sounding test pipe are shown in Figures 117 to 120 for the various gate valve openings.

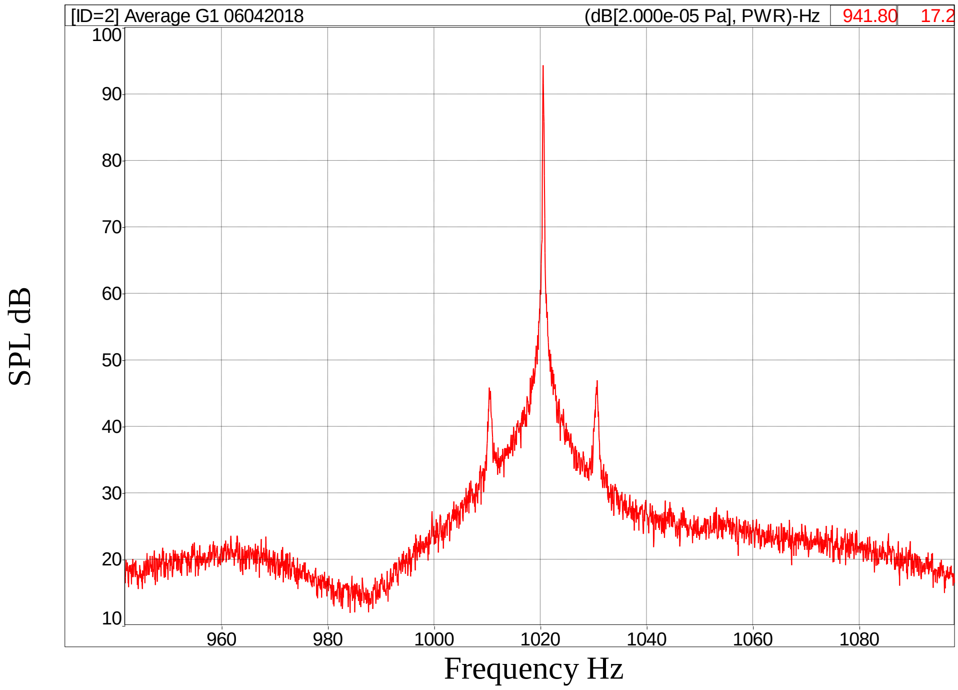


Figure 116 Internal control valve

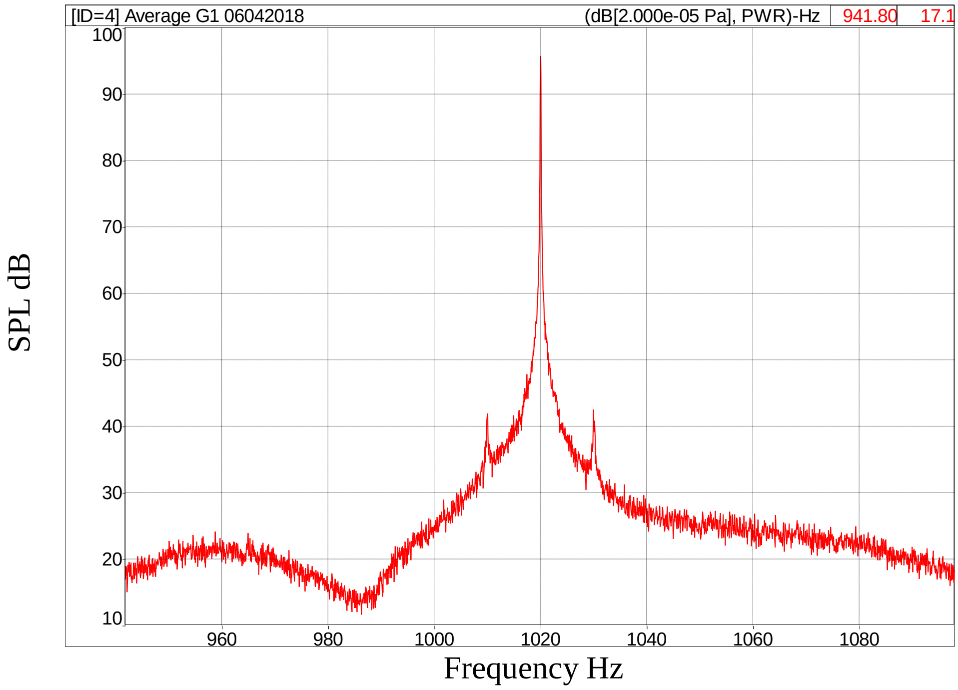


Figure 117 Both gate valves open

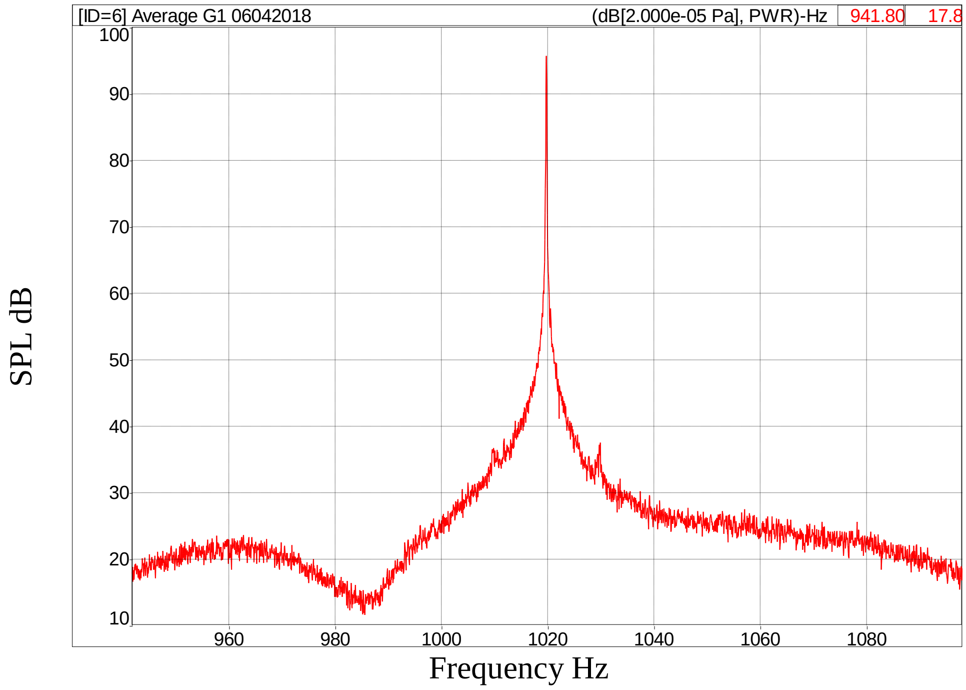


Figure 118 One gate valve open

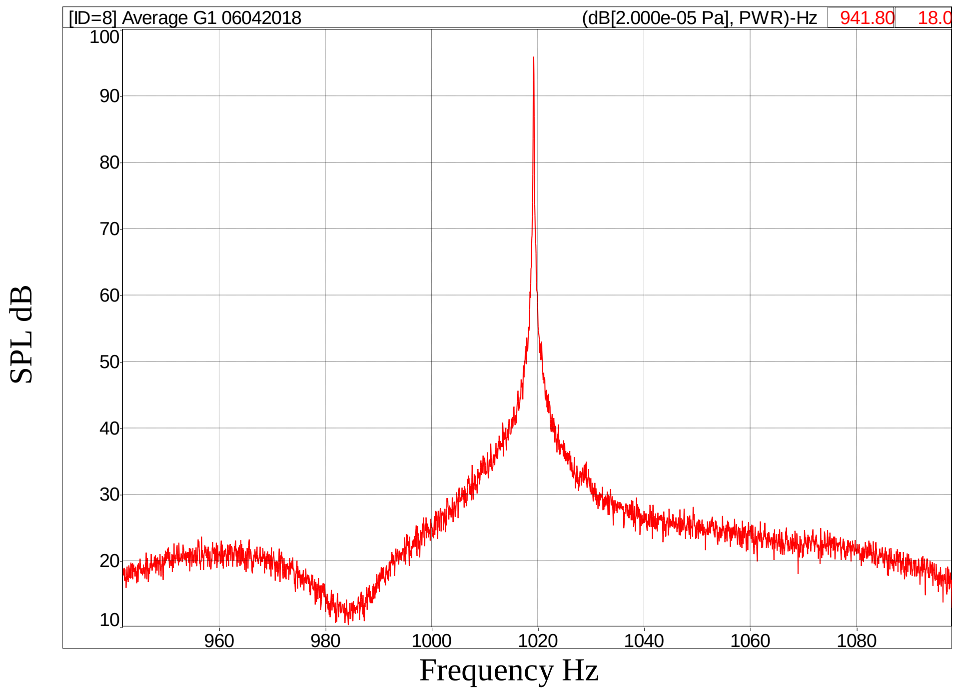


Figure 119 One gate valve closed and one gate valve partially open

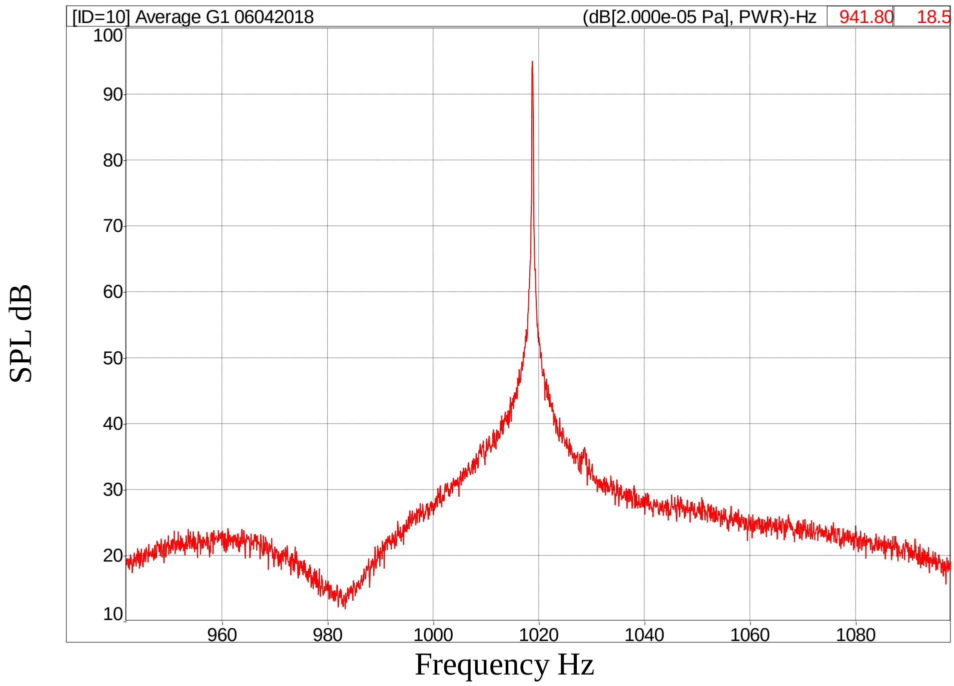


Figure 120 Both valves closed and the reservoir allowed to fall

11.3.2.2 Reservoir top vibration results

The corresponding reservoir top vibration levels are shown in Figures 121 to 125

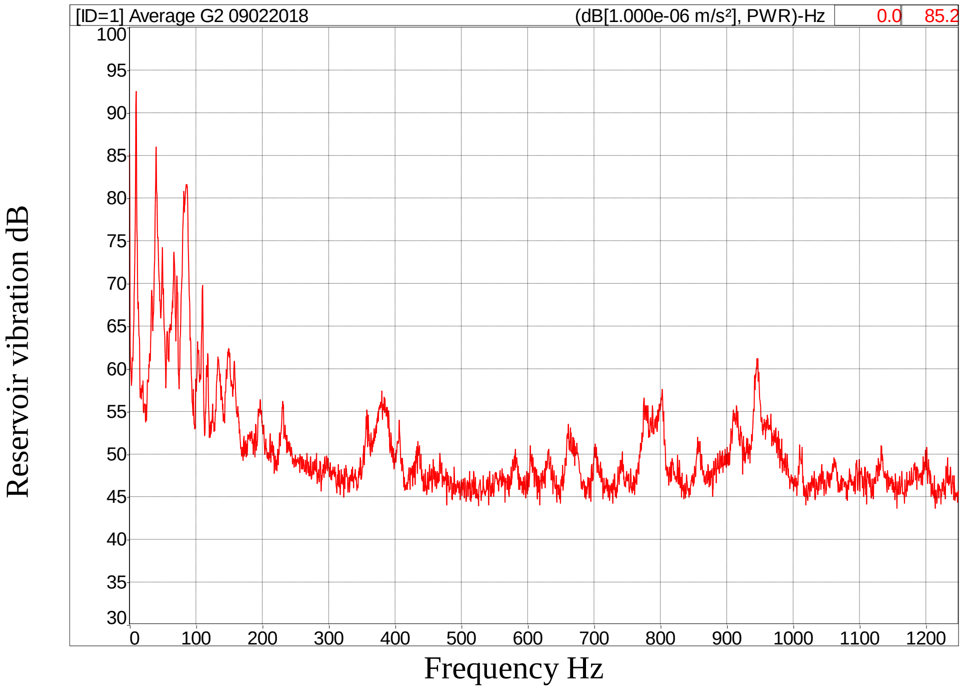


Figure 121 Internal control valve direct connection to the blower

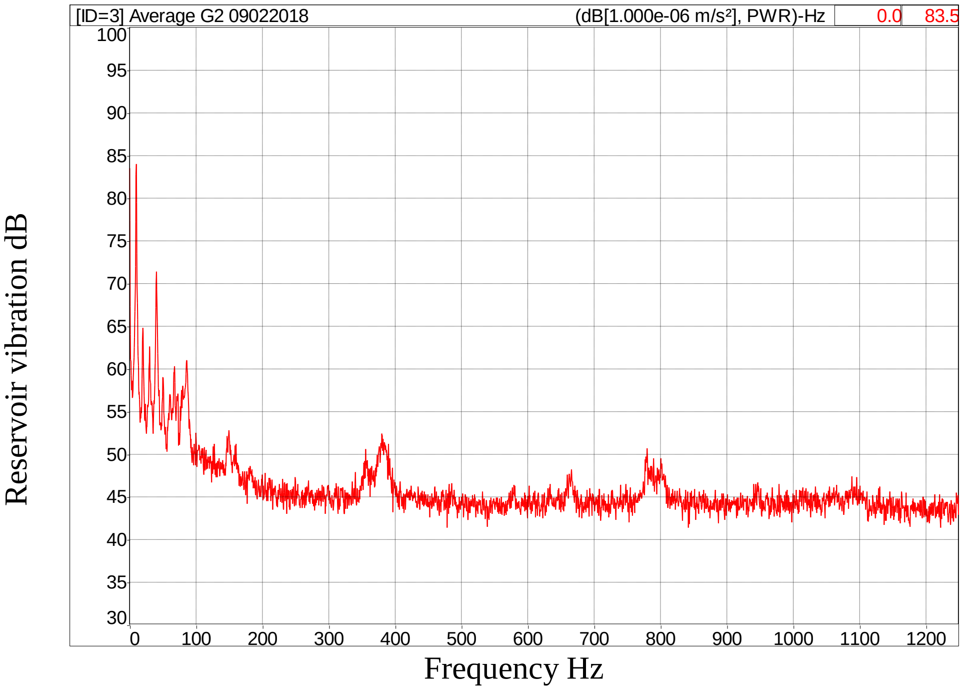


Figure 122 Both gate valves open

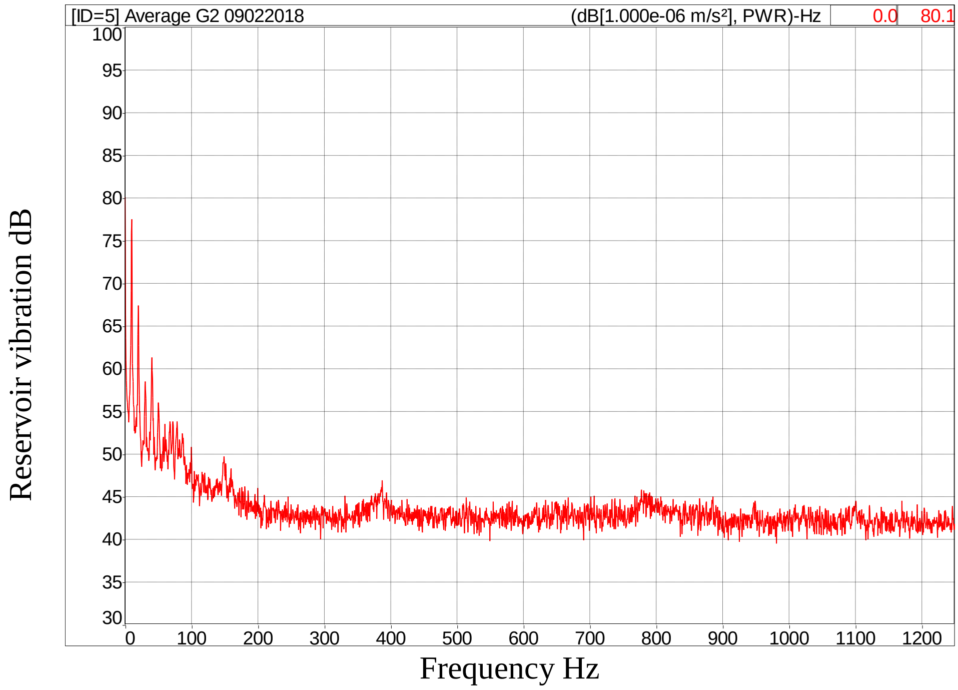


Figure 123 One gate valve closed one gate valve open

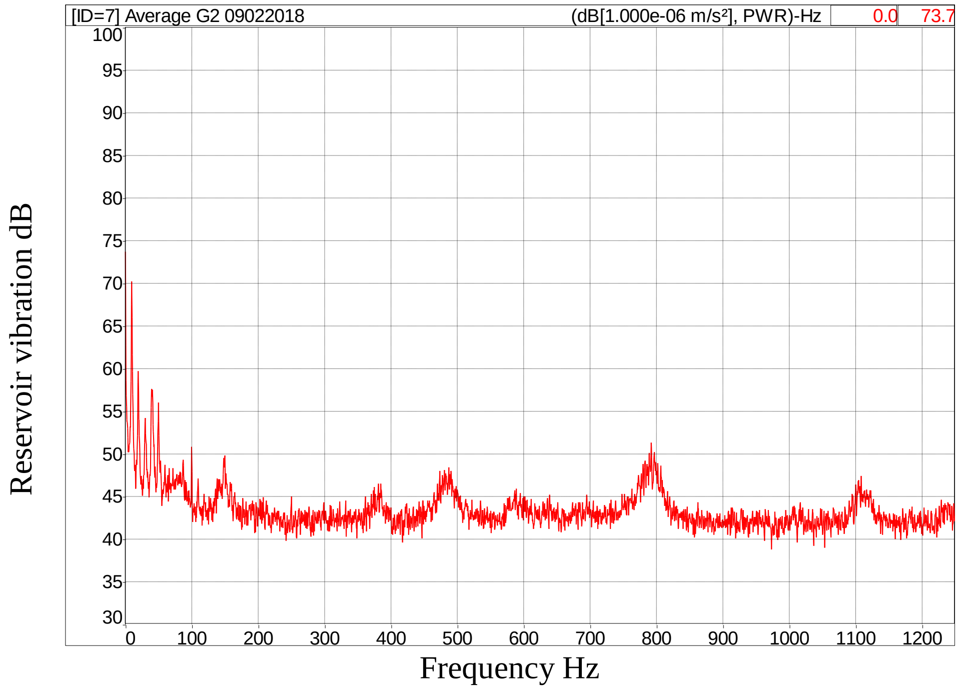


Figure 124 One gate valve closed one gate valve partially open

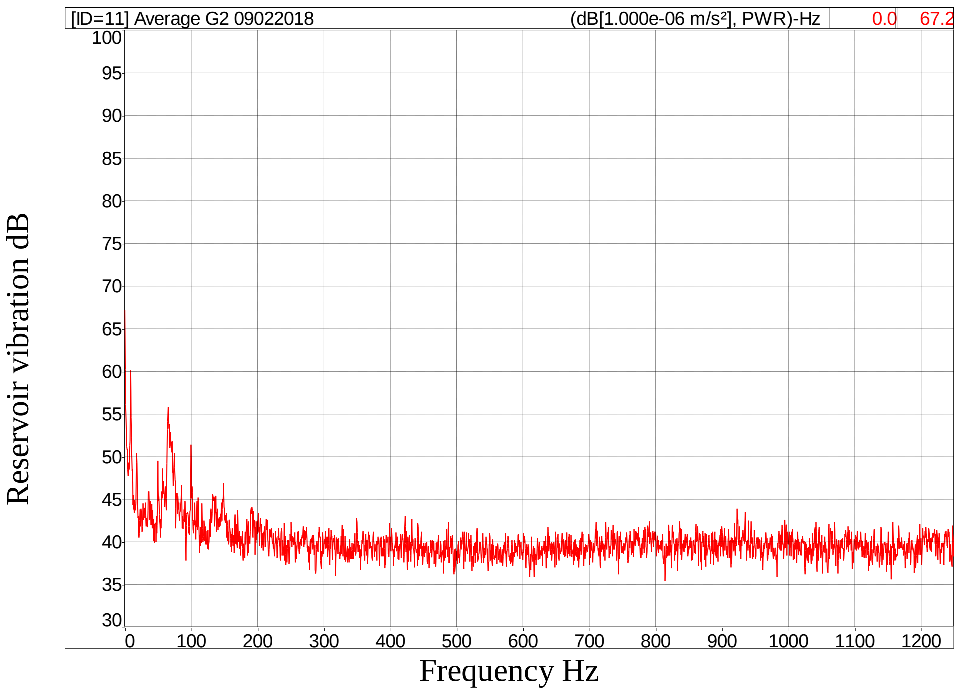


Figure 125 Both gate valves closed and the reservoir allowed to fall

Results for the various gate valve conditions are shown in Figure 126.

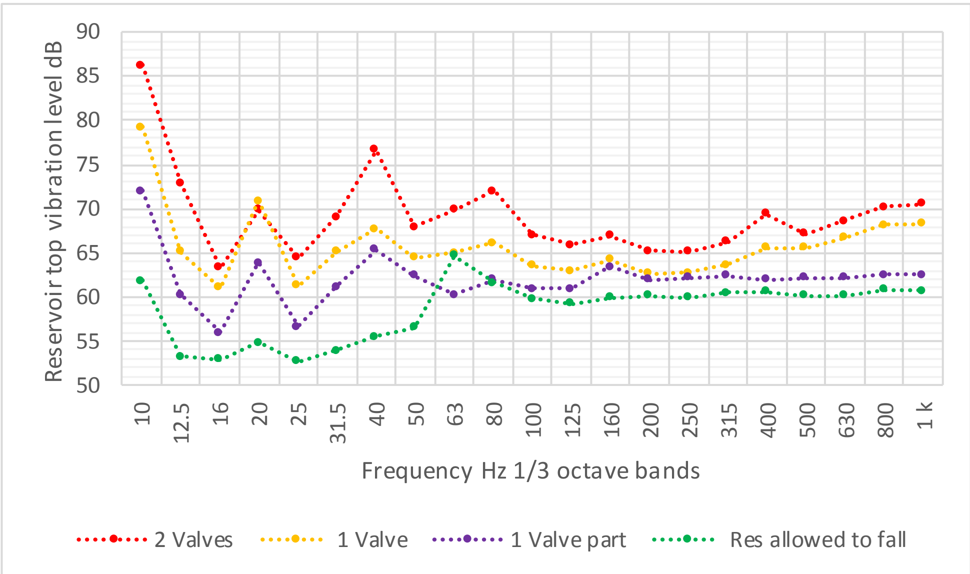


Figure 126 Reservoir top vibration levels for the various gate valve conditions.

11.4 Discussion

11.4.1 Control valves

The results of the reservoir top vibration levels for the three control valves shown in Figure 115 are quite striking. The external control valves have substantially less, almost 20dB, reservoir top vibration in the 10Hz 1/3 octave band and even more in the 40 & 80Hz 1/3 octave bands than the internal valve. The superior performance of the external control valves may be due to the profile of the opening and the effective mass of the moving part of the valve. The guillotine valve needs a heavy plate to overcome friction on the face of the valve produced by the wind pressure from the blower acting on the back of the valve. The roller-blind valve is made from leather and has substantially less mass than the guillotine valve.

11.4.2 Gate valves

Isolating the blower, using the two gate valves, substantially reduces side-bands and reservoir top vibration levels, effectively producing a wind system similar to a feeder system described in Chapter 2 - flutter free.

11.5 Conclusion

The type and construction of the control valve is significant in the control and elimination of organ pipe flutter. The guillotine valve produces lower reservoir top vibration levels than the roller-blind valve.

12 Attenuating Devices

12.1 Preamble

Previous Chapters examined the effects the reservoir, control valve and the blower have on organ pipe flutter. In this Chapter, the last element of the wind system, the connecting duct between the blower and the reservoir, will be examined from an organ building perspective with reference to the inclusion of some form of attenuating device.

The primary consideration when designing the wind system, is to ensure that a sufficient area of duct is provided to satisfy full organ demand without any pressure drop, that may cause pitch changes. Any breakout noise radiating from the wind system must be kept to an absolute minimum. The blower is one source of noise and many blowing plants are sited in cellars or towers. Today, pipe organ blowers are much quieter and placed within the organ case inside an acoustic enclosure. Occasionally, breakout noise from the duct connecting the blower to the reservoir is a problem - normally solved by adding carpet underlay or carpet tiles to the inside of the duct. Sometimes, more “drastic” remedies are necessary and a plenum chamber or a splitter silencer is fitted. Some organ builders have incorporated a wind straightening or anti-turbulent device immediately after the blower, as shown in Photographs 31 & 32.

Most acoustic attenuating devices are dimension sensitive, and if optimal performance is to be achieved both the frequency range and degree of attenuation need to be considered.

12.2 Noise attenuators

Noise attenuating devices are generally referred to as “silencers” or “mufflers”. A silencer may be “reactive”, (the acoustic energy is attenuated by “reflection”), or, “absorptive” (the acoustic energy is attenuated by “absorption”). Many silencers use a combination of both to achieve the required attenuation. Reactive silencers are most effective at low frequencies, and absorptive silencers at high frequencies.

The most common silencing devices are used in automobiles and air conditioning systems. Harmonic analysis of the air flow in the exhaust pipe or air conditioning duct, reveals a pulsating flow superimposed on a constant flow of the gas or air. To be effective, the silencer needs to attenuate the periodic frequencies, without unduly affecting the constant air flow.

12.2.1 Acoustic requirements

The noise reducing properties of silencers are covered by Anderson [36], Barron [37], Bies [38], Beranek [39], Evensen [40], Harris [41] & Smith [42] , and attenuation levels are normally specified as a function of frequency in octave or 1/3 octave bands. Several parameters are used to define the performance:

- **Insertion loss IL** - the difference in sound pressure level for the surroundings due to the inclusion of the silencer.
- **Noise reduction NR -** the difference in sound pressure level between the point immediately up-stream and the point immediately down-stream of the silencer.
- **Transmission loss TL** - the change in sound power level across the silencer if no energy is reflected back to the silencer from the tail pipe.

12.2.2 General requirements

Acoustic requirements are often compromised by other considerations. In organ building applications it is important that the pressure drop across the silencer is kept to a minimum. The pipe organ survey shows that space is limited, and the inclusion of a silencer may have to be incorporated into the connecting duct. Commercial silencers are generally manufactured from galvanised or stainless steel. In organ building the material of choice is wood so that steel is very rarely found in pipe organs. Finally, the cost of including a silencer may be a reason why they are not more commonly found.

12.2.3 Reactive or reflective silencers

Reactive or reflective silencers are characterised by large variations in maxima and minima in the transmission loss curves, as shown in Figure 128, and are generally used where contamination of the unit may take place. The simplest form of reactive silencer, is an expansion of the duct using a chamber with dimensions greater than the inlet duct. The sudden expansion provides a change of acoustic impedance, caused by the change in the cross-section of the duct and reflects sound energy back to the source. The amount of attenuation is a function of the area of the inlet pipe and the area of the expansion chamber. A single expansion chamber is shown in Figure 127.

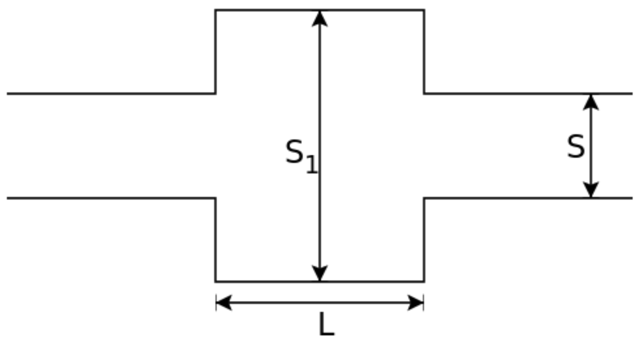


Figure 127 Single expansion chamber (After Davis [43])

The transmission loss, L_{tl} for a simple expansion chamber shown in Figure 127 is given in equation 7 Davis [43].

$$L_{tl} = 10\, log_{10}\left[1 + \frac{1}{4}\left(m - \frac{1}{m}\right)^2 sin^2\, k.L\right] \text{dB} \tag{7}$$

Where S = Cross-sectional area of duct

S_1 = Cross-sectional area of chamber

L = Length of chamber

m = The expansion ratio S_1/S

k = $2.\pi.\, f/c$

c = The speed of sound

The expansion ratio m is the main parameter for determining the attenuation and the transmission loss increases as the expansion ratio increases. Figure 128 shows the transmission loss for expansion ratios 4, 8 & 12.

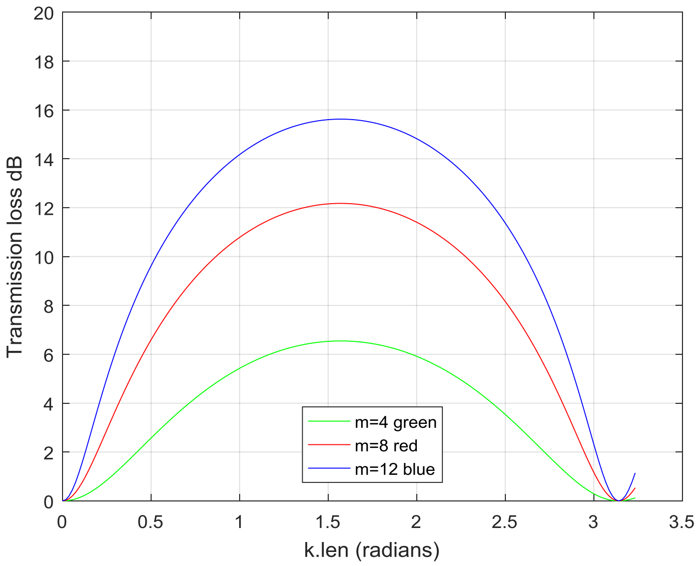


Figure 128 Transmission loss for a single expansion chamber

Minimum attenuation occurs at 0, π, 2π etc and maximum attenuation at π/2, 3π/2 etc. The maximum attenuation occurs for the frequency at which the length of the chamber is a quarter of the wavelength. The attenuation drops off and is at a minimum when the chamber length corresponds to half a wavelength. Other maxima occur at odd numbers of quarter wavelengths, and minima at even numbers of quarter wavelengths.

The transmission loss for a typical device that may be used in organ building with 100 & 150mm inlet and outlet pipes and a chamber length of 0.6, 1.2 & 2.4m together with chamber diameters of 300, 400 & 500mm is shown in Figures 129 to 134.

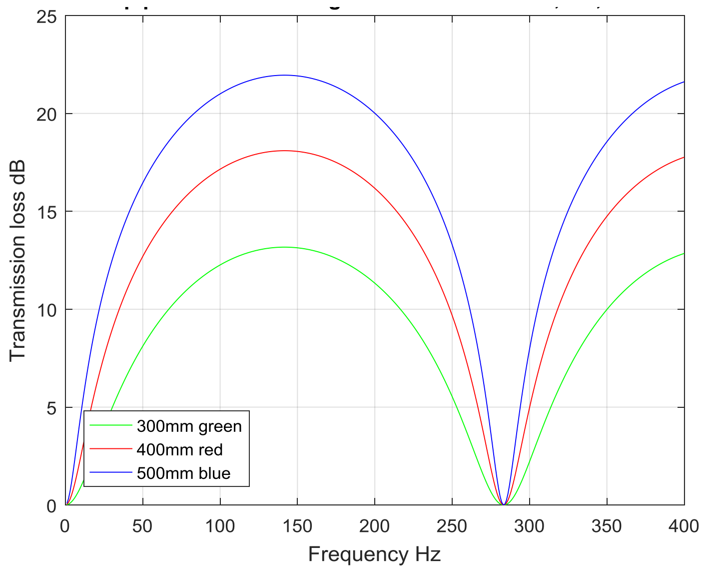


Figure 129 Inlet pipe 100mm dia Length 0.6m Chamber dia 300, 400 & 500mm

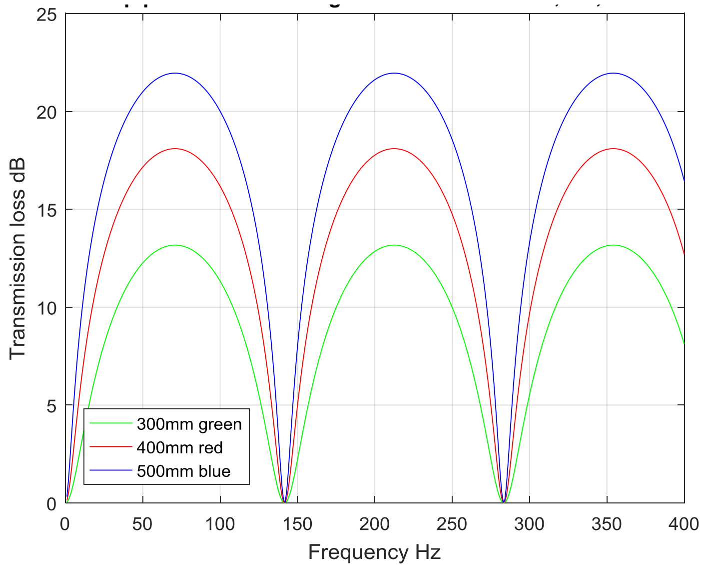


Figure 130 Inlet pipe 100mm dia Length 1.2m Chamber dia 300, 400 & 500mm

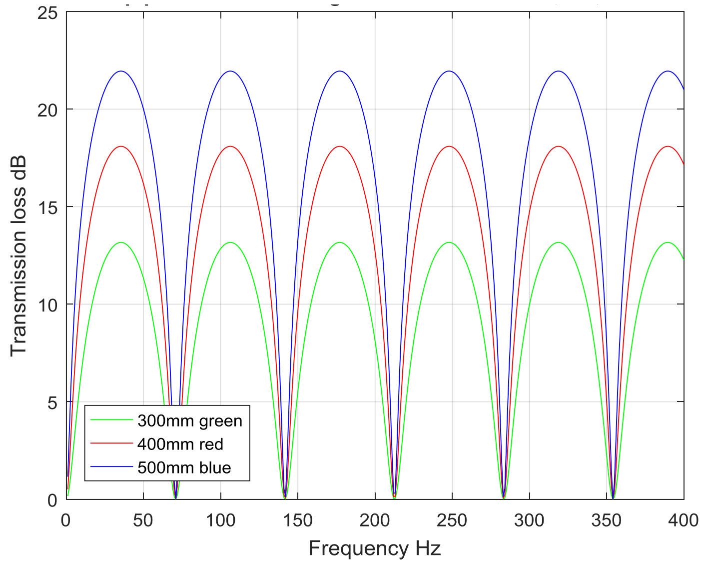


Figure 131 Inlet pipe 100mm dia Length 2.4m Chamber dia 300, 400 & 500mm

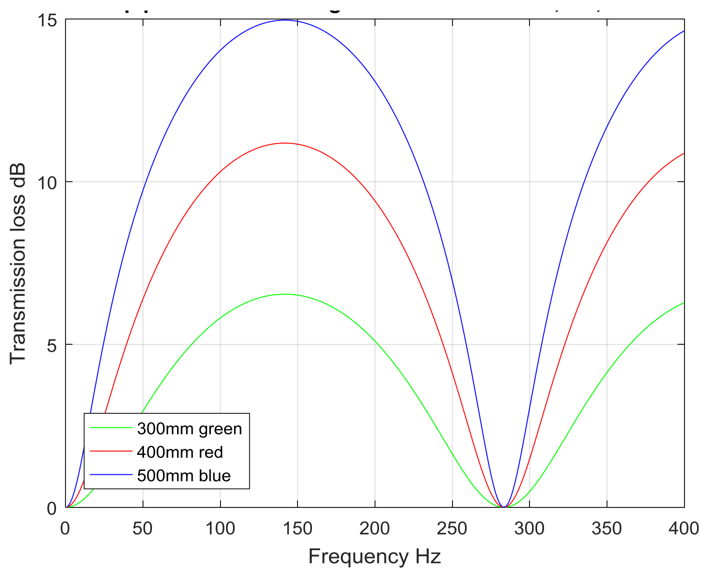


Figure 132 Inlet pipe 150mm dia length 0.6m Chamber dia 300, 400 & 500mm

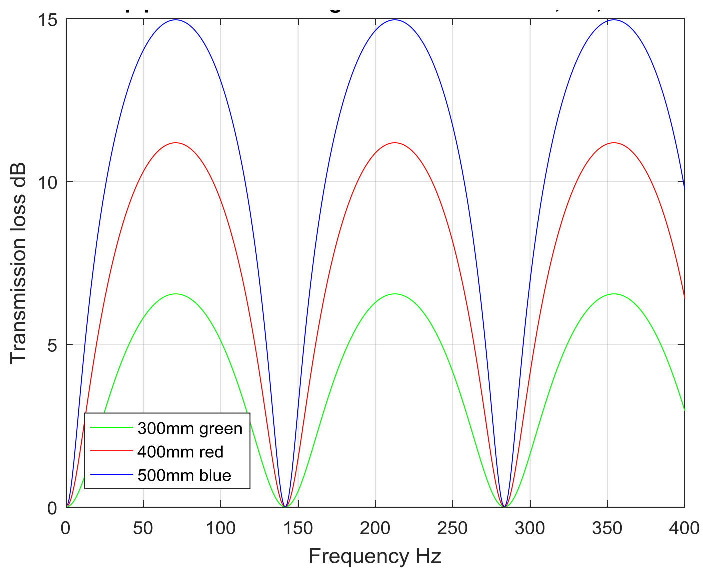


Figure 133 Inlet pipe 150mm dia length 1.2m Chamber dia 300, 400 & 500mm

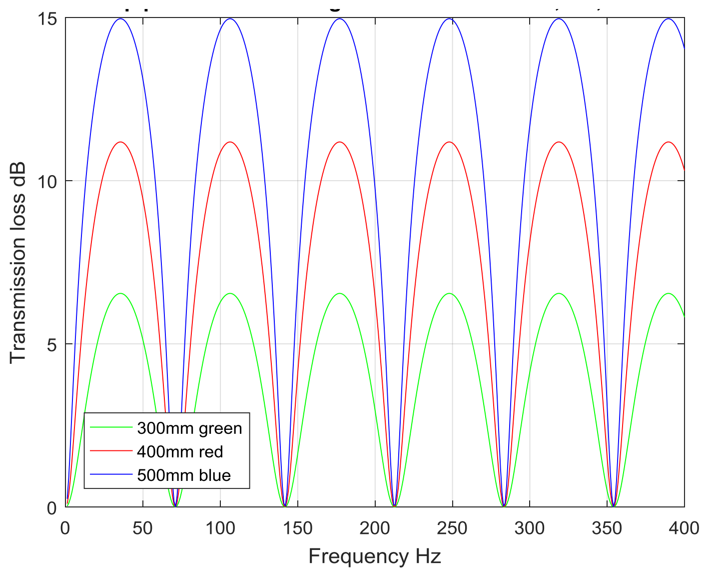


Figure 134 Inlet pipe 150mm dia length 2.4m Chamber dia 300, 400 & 500mm

By analyzing the frequency plots, the relationship between the various parameters becomes clear. Increasing the inlet and outlet pipes from 100 to 150mm, reduces the attenuation by approximately, 7dB. The length of the chamber determines the minimum attenuation frequency. A 600mm long chamber has a minimum attenuation frequency of 140Hz. A 2400mm long chamber has minimum attenuation frequency of 35Hz. The length and expansion ratio are critical for low frequency attenuation. For general attenuation of breakout wind noise, the dimensions are less critical.

From this single expansion chamber, it is possible to develop more complex arrangements with multi-chambers and corresponding transmission loss curves Davis [43].

The transmission losses derived from Equation 7 assumes that the silencer is anechoically terminated. If the silencer is terminated with a length of open ended outlet pipe, this will influence the transmission loss at certain frequencies. In organ building applications, the duct is normally terminated with a control valve. When there is no demand, the reservoir is full, and the control valve closed, and most of the wave energy is reflected back along the duct to the blower.

12.2.4 Dissipative or absorptive silencers

Dissipative or absorptive silencers usually have relatively wide-band noise reduction characteristics, as shown in Figure 136, and do not have the periodic maxima and minima found with reactive silencers. This type of device is much more effective at attenuating medium and high frequencies. Acoustic energy is converted to heat by the sound absorbing process taking place in the fibrous material.

A duct lined with sound absorbing material behaves like a simple absorptive silencer. The amount of attenuation per unit length of run, is dependent on the sound absorption coefficient of the lining material and the ratio of the cross-sectional area of the duct to its perimeter. To achieve the maximum attenuation, it important to have the greatest possible surface area of absorbing material exposed. If space is restricted, then a splitter silencer can be used to split the airway into smaller airways, each lined with absorbing material. Attenuation produced by splitter silencers, tends to be greatest at medium frequencies with less attenuation at the higher and lower frequencies. This poor high frequency performance is because the high frequency sound energy tends to be beamed down the centre of the duct and is unaffected by the absorbing material. This occurs where the wavelength is smaller than the airway width, thus, reducing the airway width increases high frequency performance. The depth of the absorbing material is also important for low frequency performance. The transmission loss, L_{tl} for a simple lined expansion chamber, is given in Equation 8 Davis [43] and shown in Figure 135.

$$L_{tl} = 10\, log_{10}\left\{\left[\cosh \sigma.\frac{L}{2} + \frac{1}{2}\left(m + \frac{1}{m}\right)\sinh \sigma.\frac{L}{2}\right]^2 cos^2 k.L \right.$$
$$\left. + \left[\sinh \sigma.\frac{L}{2} + \frac{1}{2}\left(m + \frac{1}{m}\right)\cosh \sigma.\frac{L}{2}\right]^2 sin^2 k.L\right\} \quad \text{dB} \qquad (8)$$

Where S = Cross-sectional area of duct

S_1 = Cross-sectional area of chamber

L = The length of chamber

m = The expansion ratio S_1/S

k = $2.\pi.f/c$

c = The speed of sound

σ = The attenuation per unit length of duct dB/m

The attenuation levels of unlined and lined chambers with a constant expansion ratio of 8 and attenuation coefficients of 0.1, 1, 2, 4, & 8dB/m, are shown in Figures 135 & 136.

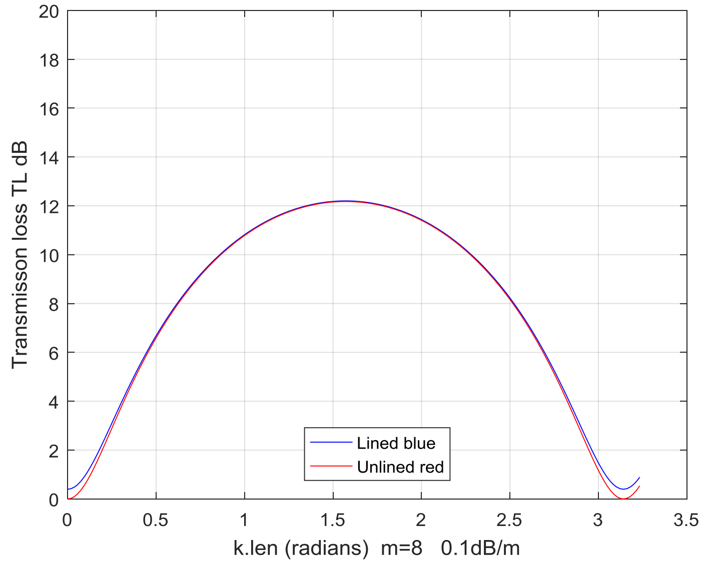


Figure 135 Lined chamber k.L (radians) m=8 σ=0.1dB/m

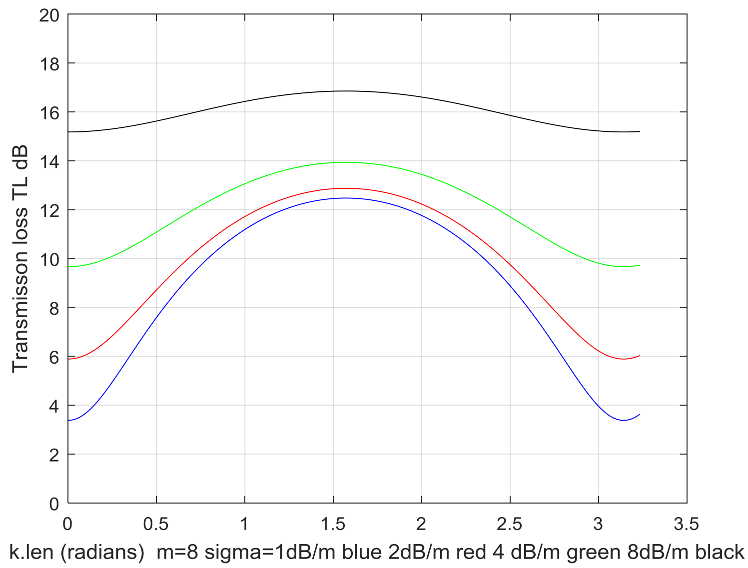


Figure 136 Lined chamber k.L (radians) m=8 σ=1dB/m blue 2dB/m red 4dB/m green 8dB/m black

An attenuation coefficient of 0.1dB/m gives very little attenuation - the silencer effectively behaves like an unlined chamber. As the attenuation coefficient increases from 1 to 8dB/m, the attenuation performance of the silencer increases, particularly at the points of minimum attenuation 0, π, 2π etc.

The transmission loss for a 400 x 400 x 1200mm long lined silencer with 100mm inlet and outlet pipes is shown in Figure 137. This is similar to that described in section 12.5.4 but without the perforated metal central pipe.

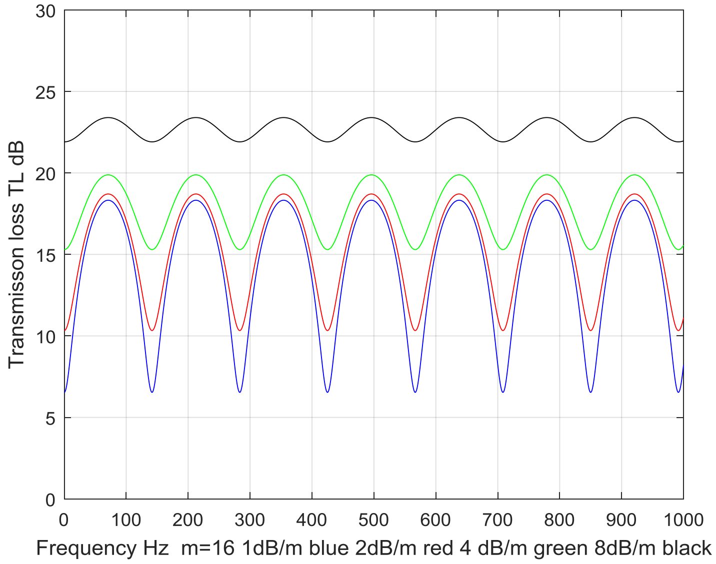


Figure 137 400 x 400 x 1200mm long lined silencer with 100mm inlet pipe

The transmission loss for a 300mm diameter x 525mm long lined silencer with 100mm inlet and outlet pipes is shown in Figure 138. This is similar to the silencer described in Section 12.5.5 but without the perforated metal pipe.

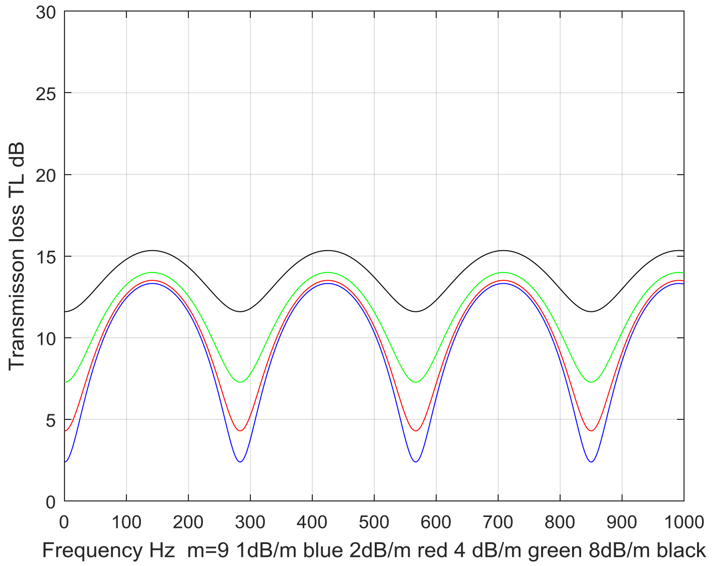


Figure 138 300mm diameter x 525mm long lined silencer with 100mm inlet pipe

12.2.5 The Helmholtz Resonator

A Helmholtz Resonator is shown in Figure 139 and consists of an enclosed volume with a connecting pipe. The most common example is a wine or beer bottle that has a body and a reduced diameter for the neck

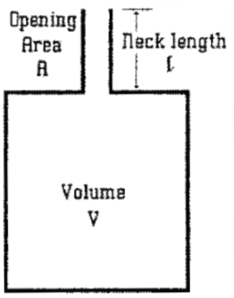


Figure 139 General features of a Helmholtz resonator (After Kinsler [45])

The Helmholtz Resonator is useful for giving high attenuation to specific frequencies, like passing blade frequencies. Where blade frequencies are a problem, one solution is to fit a quarter-length resonator, attached to the fan at the cut-off, tuned to the passing blade frequency Neise [44]. Alternatively, a small length of closed pipe fitted at right angles to the main duct as a side-branch, will also attenuate certain discrete frequencies and maximum attenuation occurs for odd numbers of quarter wavelengths of the pipe.

The resonant frequency of a Helmholtz Resonator is given in equation 9 Kinsler [45].

$$f = c/2\pi . (S/Le.V)^{1/2} \tag{9}$$

Where c = speed of sound S = area of neck Le = effective length of neck including end corrections V = volume of body

The single rise test reservoir used in Section 4 is connected to the blower with a 100mm diameter duct 4m long. Figure 140 shows the relationship between the resonant frequency and duct length.

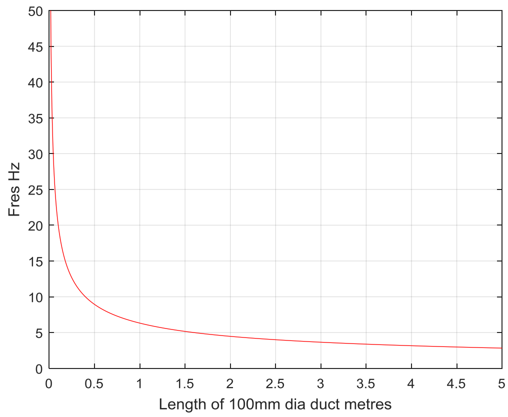


Figure 140 Resonant frequency of a single rise reservoir and connecting duct

12.2.6 Plenum Chamber

The general features of a plenum chamber are shown in Figure 141. The main difference, compared with the lined expansion chamber, is that the inlet and outlet openings are not positioned in line but are usually opposite with an offset. A plenum chamber may be a purely reactive device and have none of the internal surfaces covered. At low frequencies, attenuation is produced by the expansion chamber principle. The overall performance is greatly improved by the addition of an absorbent lining to the internal surfaces of the chamber Wells [46] Cummings [47] [48] Li [49]. The performance can also be improved by including internal baffles which may also be covered with some form of acoustic absorbent material.

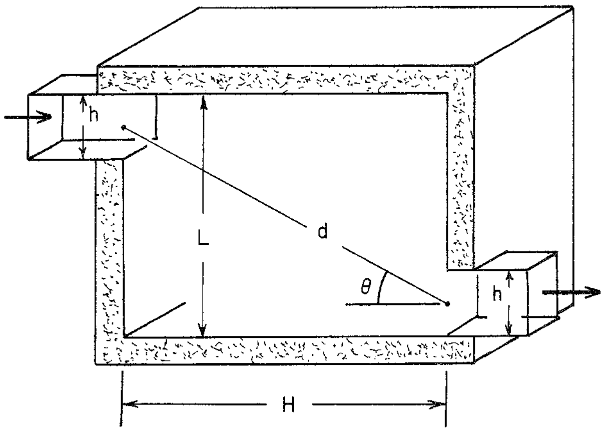


Figure 141 General features of a plenum chamber (After Barron [37])

The general expression for the transmission loss of a single plenum chamber is given in equation 10 Barron [37].

$$TL = 10\log_{10}(1/\acute{\alpha}_t)\ \text{dB} \qquad (10)$$

Where $\acute{\alpha}_t = (S_o.Q\cos\theta/4\pi d^2) + S_o/R$

S_o = Area of inlet and outlet openings

Q = Directivity factor assumed to be 4 for this geometry

R = S.ά/(1-ά)

ά = ($\acute{\alpha}_L.S_L+2.S_o)/S$

S = Total surface area

S_L = Total lined surface area

The transmission loss for a plenum chamber described in Section 12.5.2, lined with four thicknesses of mineral wool, is shown in Figure 142. No data for surface absorption coefficients is available for octave bands lower than 63Hz.

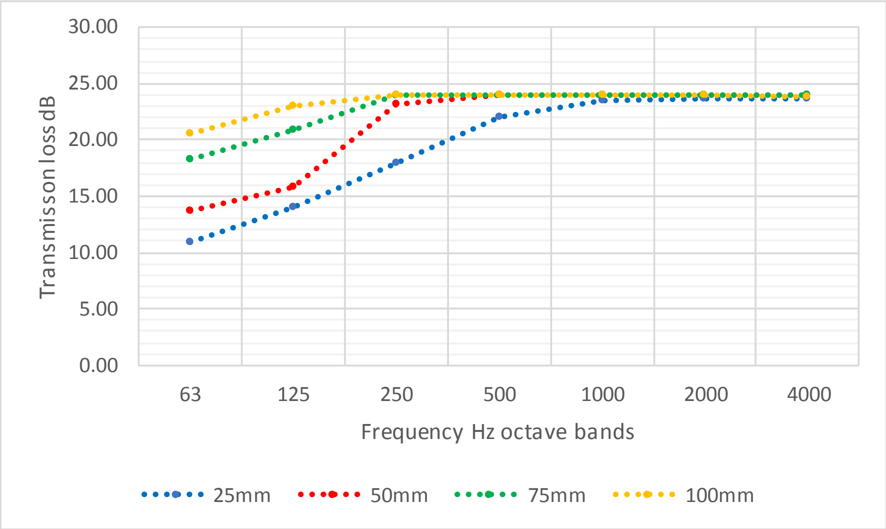


Figure 142 Plenum chamber with different fill thicknesses

In organ building applications, it is not possible to terminate the plenum chamber anechoically - if the control valve is placed at the reservoir, most of the sound wave is reflected back to the chamber. For this reason, the attenuation of any reservoir top vibration and organ pipe flutter is examined in Section 12.5.

12.3 Commercial Silencers

A range of commercial silencers are available for general heating and ventilation applications. Photographs 28 & 29 show typical circular and rectangular units. A Splitter Silencer is shown in Photograph 30.

Photograph 28 Commercial circular silencer (After Lindab)

Photograph 29 Commercial rectangular silencer (After Lindab)

Photograph 30 Commercial splitter silencer (After Lindab)

The performance of silencers is determined in accordance with ISO 7235:2003 (en) “Acoustics – Laboratory measurement procedure for ducted silencers – insertion loss, flow noise and total pressure loss”.

The main variables that determine the acoustic performance are:

- Inlet and outlet diameters
- Overall body diameter, fill thickness and material
- Length

The attenuation curves for six commercial silencers each having 100mm diameter inlet and outlet pipes are shown in Figure 143 for fill thickness of 50mm &100mm. The four circular units, 50/600, 100/600, 50/1200 & 100/1200, are imperial units and the lengths of 600mm & 1200mm correspond to 2 & 4 feet. Also shown in Figure 143, are performance curves for a metric rectangular silencer, 210mm wide and 158mm high, with 100mm diameter inlet and outlet pipes, 500mm and 1000mm long

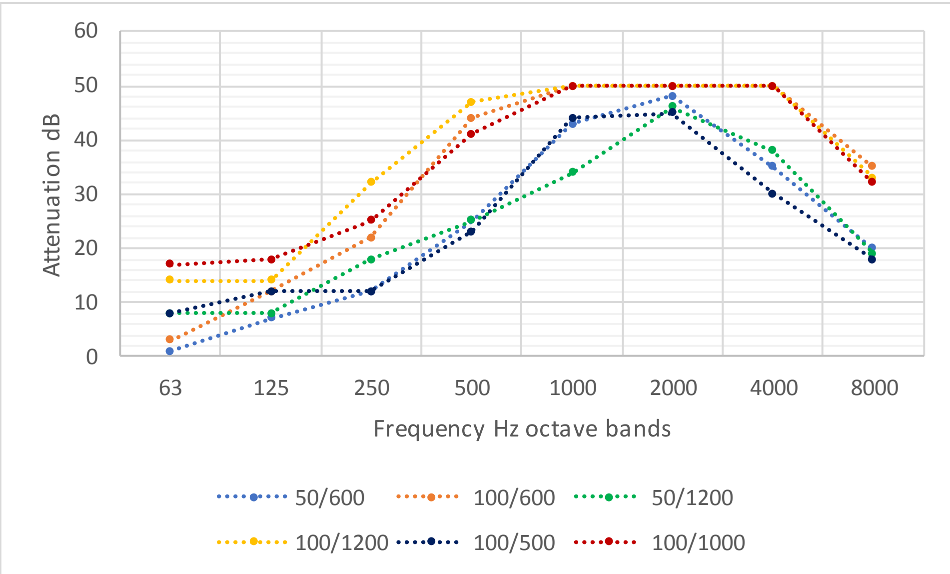


Figure 143 Performance details of six commercial silencers

The longer units, 1000mm & 1200mm, with the greatest fill material, produce the greatest attenuation. It is significant that, no data is available below the 63Hz 1/3 octave band level.

The above information was also used in the design and construction of the devices detailed in Section 12.5.

12.4 Organ building attenuating devices

Most of the organs surveyed, Figure 22, had the blower positioned within 1.5m of the reservoir. Many organs have the control valve fitted as part of the blower enclosure to reduce the level of breakout noise from the main connecting wind duct. Therefore, it is not surprising that attenuating devices are rarely used and none of the organs surveyed had attenuating devices.

With no technical acoustic data available for the various devices when used in an organ building application, further experiments were conducted to determine the effectiveness of the various attenuating devices, and their effect on reducing reservoir top vibration levels and organ pipe flutter.

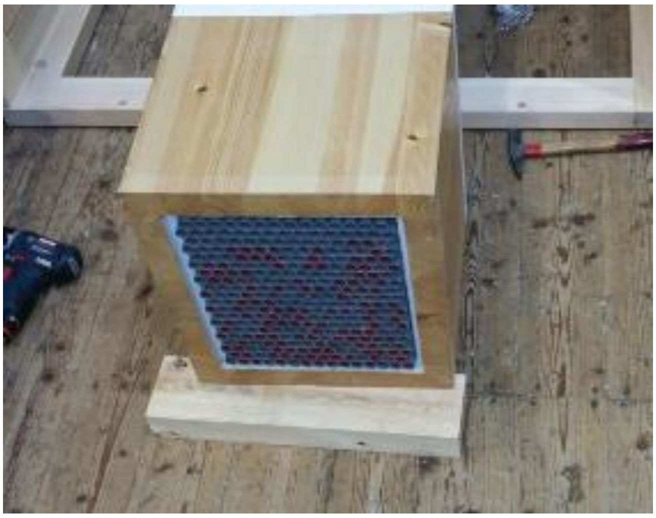
12.5 Methodology

Using the apparatus and measuring equipment described in Chapter 4, measurements were taken for various silencer constructions inserted in the duct between the blower and the reservoir. The baseline condition, blower 6 with no silencer, is used to compare the performance of each attenuating device. Blower 1 was also used without any attenuating device, for comparison.

12.5.1 Anti-turbulent device or flow straightener

The purpose of this device is to reduce or eliminate any turbulent air created by the blower and is rarely used by organ builders. Photographs 31 & 32 show a typical anti-turbulent or flow straightening device, fitted immediately after the blower. The device has an inlet section that expands into a central section which contains a series of small diameter tubes. It is this central section that is responsible for reducing any turbulent air. The outlet section is similar to the inlet section and provides a gentle reduction in the cross-sectional area and connects to the main wind duct. No technical data is available regarding the effectiveness of this device in reducing organ pipe flutter.

Photograph 31 Typical anti-turbulent device

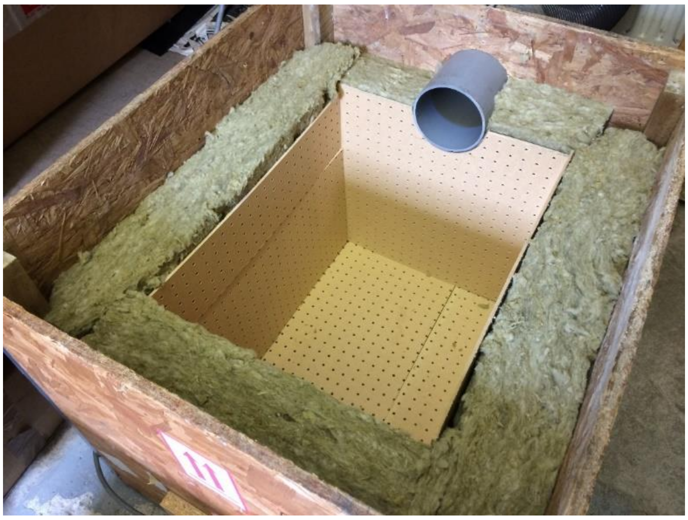


Photograph 32 Central section tube array

12.5.2 Plenum chamber

The plenum chamber is 600mm wide, 600mm high, 700mm long and constructed from 12mm thick plywood. A 100mm diameter, plastic inlet pipe is located 125mm from the bottom. A 100mm diameter, plastic outlet pipe is located 125mm from the top of the opposite side. The inside of the chamber can be lined with mineral wool and perforated metal or wood to improve low frequency performance Davern [50] Lee [51].

Photograph 33 shows a lining of 100mm mineral wool and a 3mm thick perforated hardboard face.



Photograph 33 Plenum chamber with 100mm mineral wool and perforated hardboard lining

12.5.3 Duct silencer

The duct silencer is 315mm wide, 530mm high, 1200mm long and constructed from 12mm & 18mm plywood. The 12mm top is removable so that various lining configurations can be assessed. 100mm diameter, plastic inlet and outlet pipes are located in the centre of opposite sides. The duct silencer is shown in Photographs 34 & 35 with 100mm thick mineral wool and perforated metal lining.

Photograph 34 Duct silencer with 100mm mineral wool and perforated metal lining

Photograph 35 Duct silencer with 100mm mineral wool and perforated metal lining

12.5.4 Pepper-Pot silencer

The pepper-pot silencer shown in Photograph 36 is 400mm wide, 400mm high, 1200mm long and made from 6mm plywood. A 100mm diameter galvanised perforated steel pipe is placed centrally in the silencer and the cavity is filled with mineral wool.

Photograph 36 1200mm long pepper-pot silencer

Normally, any silencing device would be placed immediately after the blower. In this position the silencer substantially reduces the amount of breakout noise radiated by the duct between the blower and the reservoir. The purpose of this device, as with the duct silencer, is not primarily as a silencing device, but as a flutter reducing device so low frequency attenuation is important. Measurements were also taken with the silencer fitted immediately before the reservoir. In this position, the length of duct from the blower to the silencer radiates any breakout noise and is not as effective as when placed next to the blower. Finally, measurements were taken with the silencer half way between the blower and the reservoir. In this position, the section of duct between the silencer and the reservoir benefits from the silencing effect.

12.5.5 Small circular silencer

The small circular silencer is 300mm diameter, 525mm long and constructed from 1.5mm thick galvanised mild steel. A 100mm diameter perforated metal pipe is placed centrally. The silencer is shown in Photographs 37 & 38. The void between the outer wrapper and the perforated metal pipe is relatively small, which is particularly useful when considering other, more expensive, fill materials such as Activated Carbon. Activated Carbon is known for its low frequency properties and is used by some loudspeaker manufacturers to improve the bass response of loudspeaker enclosures Wheeler [52] Bechwati [53] Venegas [54].

The 100mm cavity is filled with mineral wool and then with activated carbon contained in 0.85Kg cloth bags.

Photograph 37 Small 300mm x 525mm long circular silencer

Photograph 38 Activated carbon filling

12.6 Results

For clarity the frequency spectra are shown in 1/3 octave bands.

12.6.1 Plenum chamber results

The reservoir top vibration level results are shown in Figures 144 & 145 for direct no silencer (blue), plenum chamber only (red), 40mm mineral wool and perforated hardboard (green), 100mm mineral wool and perforated hardboard (yellow).

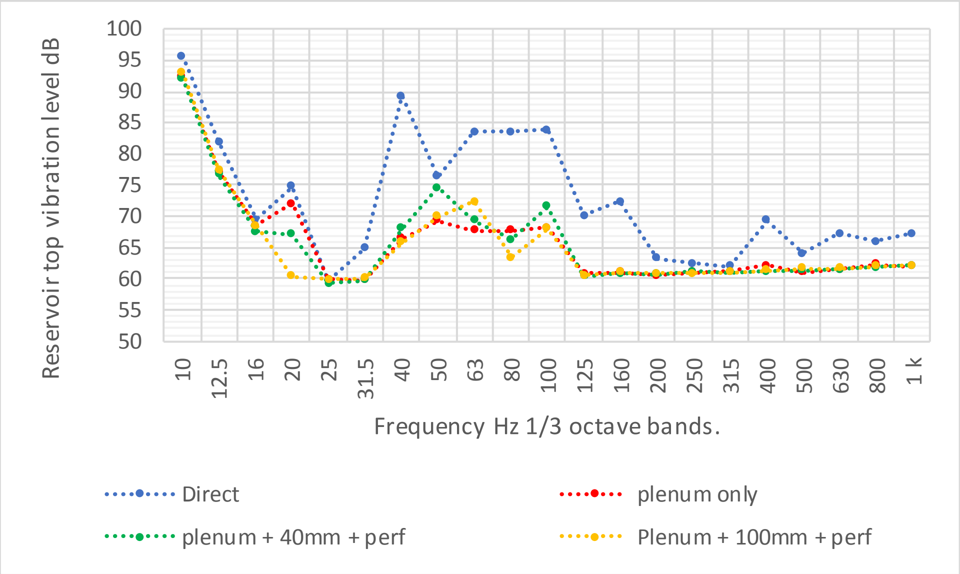


Figure 144 Plenum chamber results 0 to 1kHz

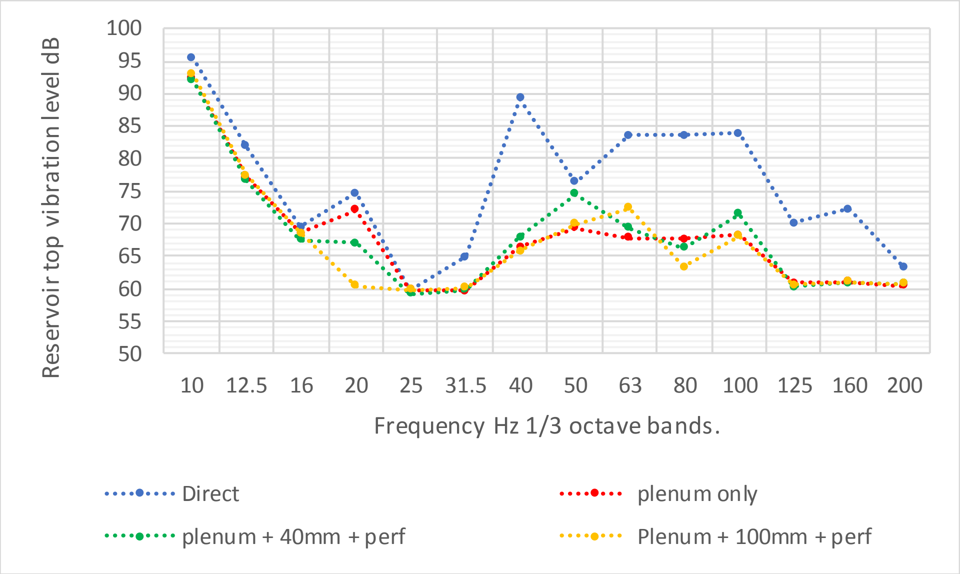


Figure 145 Plenum chamber results 0 to 200Hz

12.6.2 Duct silencer results

The reservoir top vibration levels are shown in Figures 146 & 147 for direct no silencer (blue), duct silencer only (red), 50mm mineral wool (purple), 50mm mineral wool and perforated hardboard (green) 100mm mineral wool and perforated hardboard (yellow). 100mm mineral wool and perforated metal (grey).

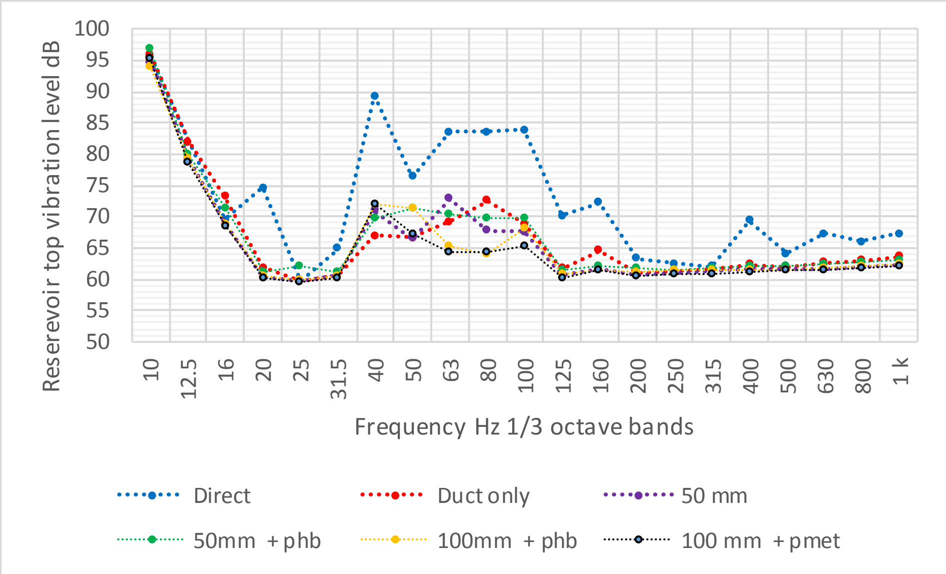


Figure 146 Duct silencer results 0 to 1kHz.

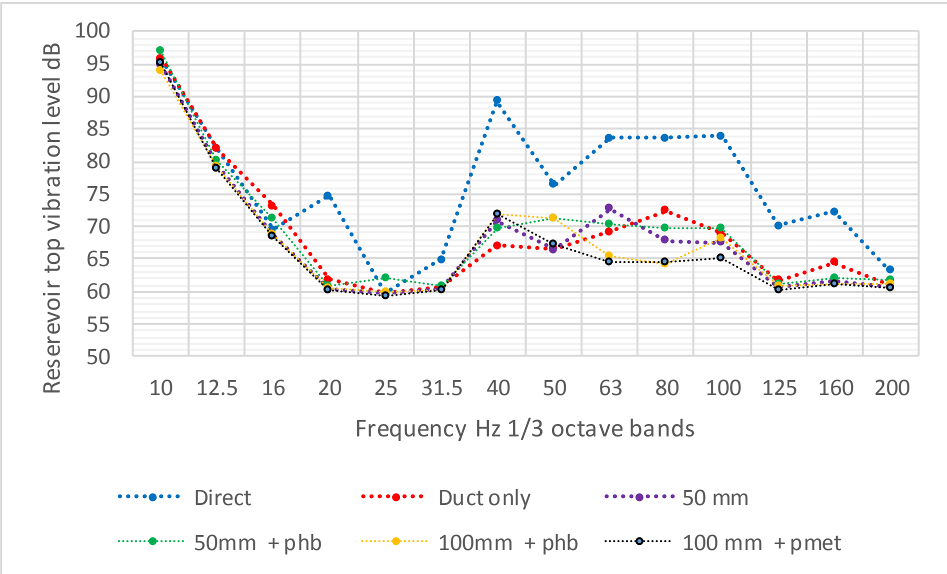


Figure 147 Duct silencer results 0 to 200Hz.

12.6.3 Pepper-Pot silencer results

The reservoir top vibration levels are shown in Figures 148 & 149 for direct no silencer (blue), silencer at the blower (purple), silencer in the middle (green) and at the reservoir (red).

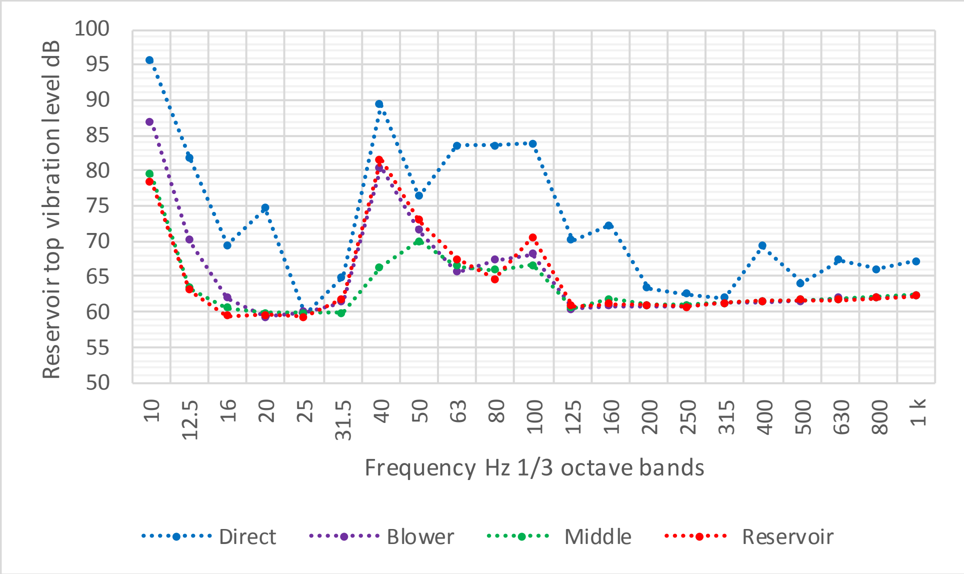


Figure 148 Pepper-pot silencer results 0 to 1kHz.

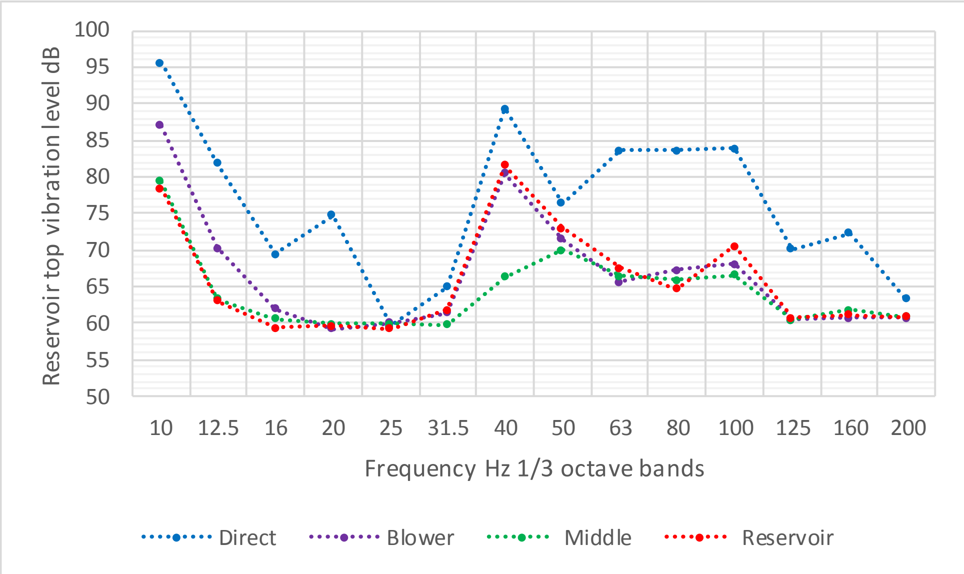


Figure 149 Pepper-pot silencer results 0 to 200Hz.

The percentage amplitude modulation using blower 6, without the silencer, is shown in Figure 150. The percentage amplitude modulation with the silencer in the mid position, is shown in Figure 151. Figure 152 shows the percentage amplitude modulation for blower 1 without the silencer.

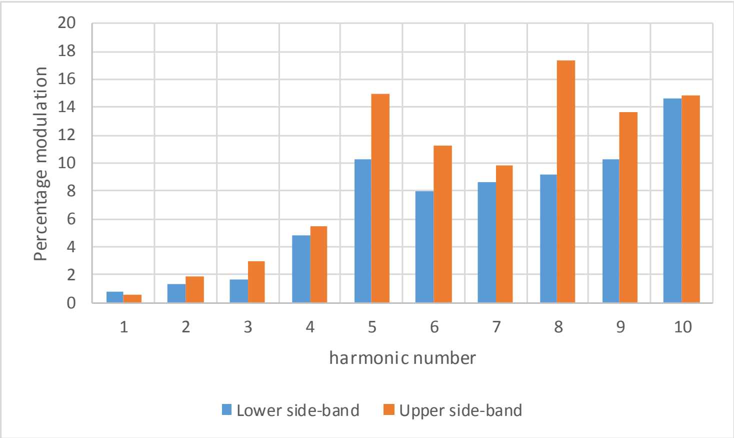


Figure 150 Percentage amplitude modulation blower 6 without silencer

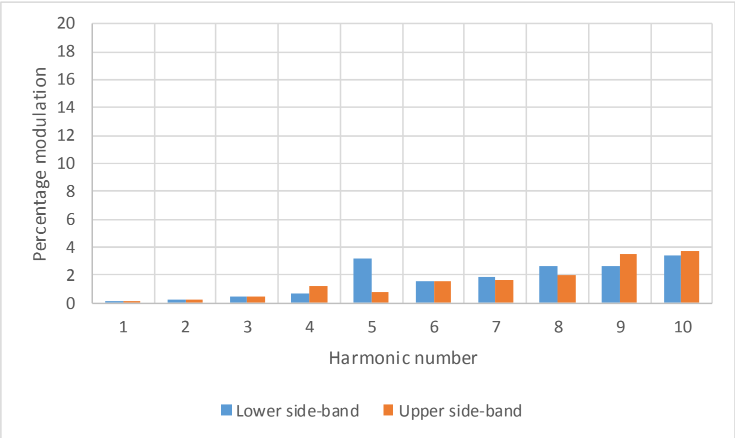


Figure 151 Percentage amplitude modulation blower 6 with silencer in mid position

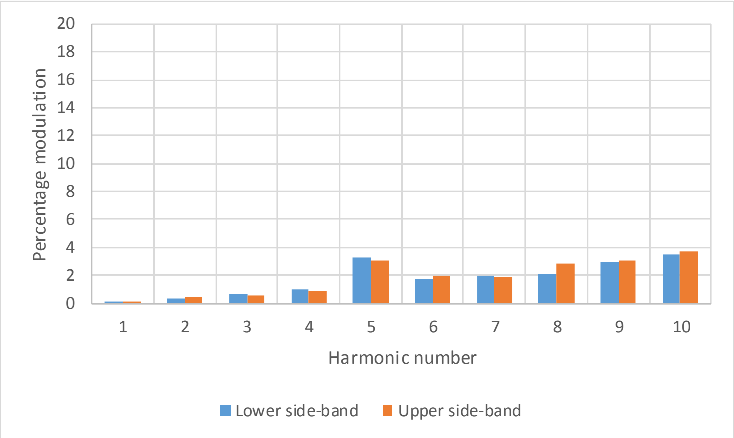


Figure 152 Percentage modulation blower 1 without silencer

12.6.4 Small circular silencer results

The reservoir top vibration levels are shown in Figures 153 & 154 for direct no silencer (blue), silencer with 100mm mineral wool filling (red), silencer with 100mm Activated Carbon filling. (green).

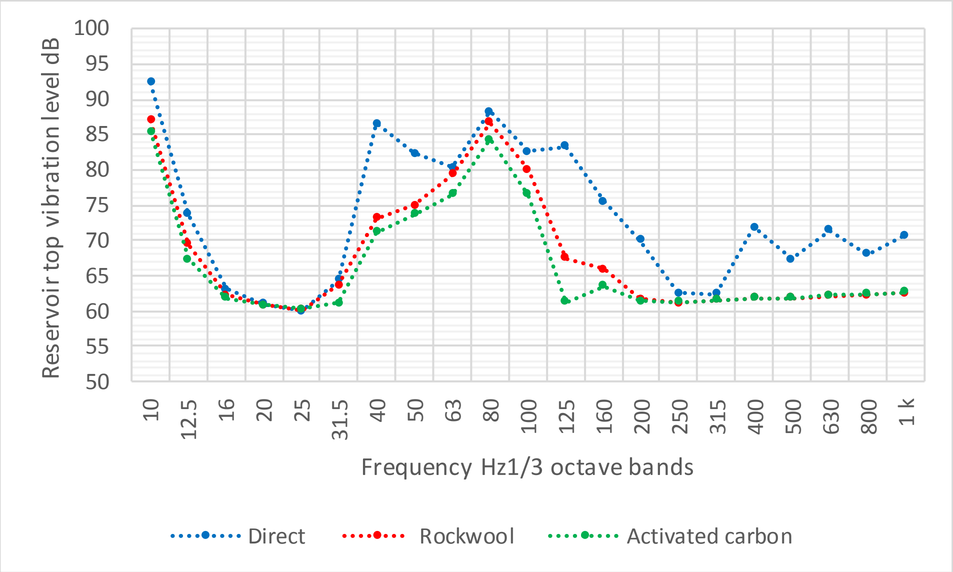


Figure 153 Silencer results for mineral wool and Activated Carbon 0 to 1kHz

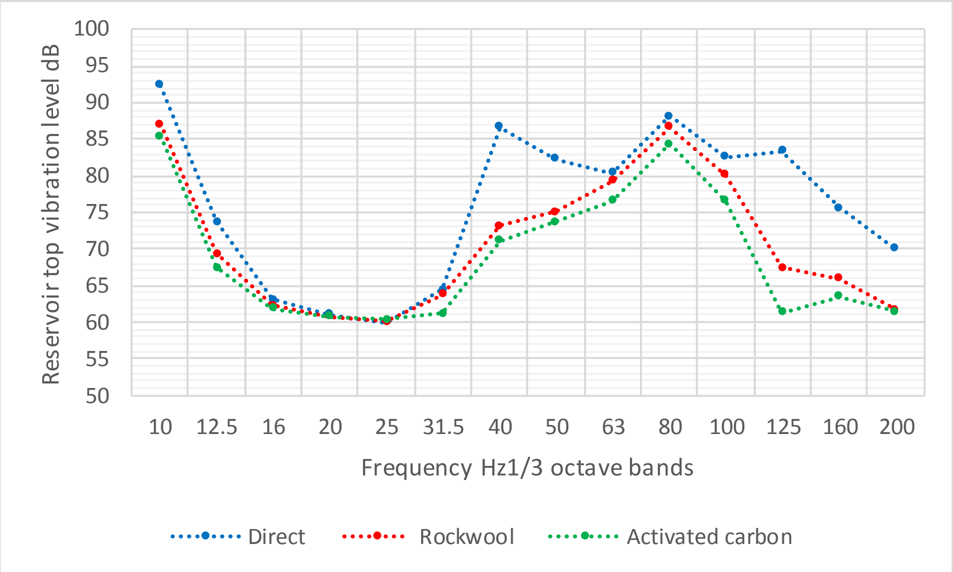


Figure 154 Silencer results for mineral wool and Activated Carbon 0 to 200Hz

12.7 Discussion

12.7.1 General

The inclusion of an attenuating device in the wind system is very rare, and is normally included to address excessive breakout noise from the main duct. Commercially available splitter silencers, positioned next to the blower, have been used for the control of breakout noise in the connecting duct between the blower and reservoir. Most manufacturers specifications stop at the 63Hz 1/3 octave band level, so no data is available in the primary organ pipe flutter region of 10Hz to 20Hz. To my knowledge, no silencing device has been fitted to overcome organ pipe flutter.

The theoretical performance curves for reactive and dissipative silencers, give an insight into how these devices perform in traditional air conditioning applications, where the assumption that the outlet is terminated anechoically may be true. In organ building applications the inclusion of a control valve in the duct means that the sound is reflected back to the blower. This condition necessitated the experiments described in Section 12.5 to determine the effectiveness of these devices in reducing reservoir top vibration levels and organ pipe flutter.

12.7.2Helmholtz resonator

Figure 51 shows side-bands at approximately 3Hz on each side of the fundamental frequency of the test pipe. Figure 140 shows that this frequency is the Helmoltz frequency of the test reservoir and the 4m long 100mm diameter connecting duct. This is less of a problem for weighted systems, where the mechanical mass dominates.

12.7.3 Plenum chamber

Figures 144 & 145 show reservoir top vibration levels for the plenum chamber. The device acts as a silencer and reduces breakout noise from the connecting duct but has only limited effect in the principal area of organ pipe flutter of 10Hz. Adding various lining materials has some additional effect in the 20Hz 1/3 octave band, but only limited effect reducing reservoir top vibrations in the 10Hz region, reducing the reservoir top vibration level by approximately 4dB.

12.7.4 Duct silencer

Figures 146 & 147 show reservoir top vibration levels for the duct silencer. The device gives good attenuation of duct breakout noise but has little effect on the principal area of organ pipe flutter of 10Hz. Adding various lining materials reduces the levels of duct breakout noise, but makes no improvements to the reservoir top vibration levels in the 10Hz region.

12.7.5 Pepper-pot silencer

Figures 148 & 149 show reservoir top vibration levels for the pepper-pot silencer. The device gives good attenuation of duct breakout noise and has significant effect in the principal area of organ pipe flutter of 10Hz. Frequency spectra are shown for three positions – blower, middle and reservoir. Results show that the attenuation of reservoir top vibration levels is position sensitive. The lowest levels of reservoir top vibration occur with the silencer located in the mid position. In this position, the device drastically reduces the reservoir top vibration level by approximately 15dB in the 10Hz 1/3 octave band and 25dB in the 40Hz 1/3 octave band. With the silencer placed next to the reservoir, similar results are obtained to those obtained with the silencer in a central position, but the attenuation in the 40Hz 1/3 octave band level is only 10dB. The poorest results occur with the silencer placed next to the blower. In this position, the reduction in the 10Hz 1/3 octave band level is approximately 7.5dB, half the attenuation of the other two positions, with further reduction in performance in the 12.5Hz and 40Hz 1/3 octave bands.

The results are significant in showing that the effectiveness of the device is position sensitive. This indicates that, in this particular application, the connecting duct may be behaving like a transmission line and termination is critical in achieving the best performance Harris [41] Morse [55]. Further research is needed to verify this.

The reduction of 15dB shown in Figures 148 & 149 in the 10Hz 1/3 octave band, is also reflected in Figures 150 & 151 which show the percentage amplitude modulation of the sounding test pipe with no silencer and then with the silencer in the mid position. These results clearly show the benefit of including a pepper-pot silencer to dampen reservoir top vibration levels and reduce any amplitude modulation of the organ pipe. For comparison, Figure 152 shows the level of percentage amplitude modulation for blower 1 that produced one of the lowest levels of amplitude modulation.

12.7.6 Small circular silencer

The small circular silencer results are shown in Figures 153 & 154. The main purpose of these measurements was to determine what effect the fill material has on attenuating reservoir top vibration levels. The use of Activated Carbon as a fill material produced disappointing results with only a 2dB lower level than using mineral wool. Better results were expected.

The overall performance of the device in reducing wind breakout noise is less effective than the other devices particularly in the 31.5 to 125Hz 1/3 octave bands. Reductions in reservoir top vibration levels in the 10Hz 1/3 octave band of 5 & 7.5dB were achieved with mineral wool and Activated Carbon fill, respectively. These results are better than those achieved from the plenum chamber and the duct silencer in the 10Hz 1/3 octave band.

12.8 Conclusions

The inclusion of some form of attenuating device sited next to the blower, can substantially reduce break-out wind noise from the connecting wind trunk and the amount of attenuation is dependent on the type of device selected. Generally, these devices have little effect on reducing reservoir top vibration levels in the 10Hz 1/3 octave band and so have little, if any, effect on reducing organ pipe flutter.

For basic duct breakout noise attenuation, the wood duct, that connects the blower to the reservoir could be modified to include a section of absorbent material in the duct walls. From a practical point of view, this may be the best organ building solution. The longer the duct, the better the low frequency attenuation.

The small silencer gave reasonable results in attenuating unwanted duct breakout wind noise and had some effect on reducing low frequency, reservoir top vibration levels.

The pepper-pot design, which is effectively twice the length of the small silencer, produced good attenuation of duct breakout noise, together with significantly less reservoir top vibration levels. This was particularly noticeable in the 10Hz 1/3 octave band, where reservoir top vibration levels are significantly reduced.

13 Summary, Conclusions and Further Work

In this final Chapter the main outcomes and conclusions of the research are summarised and potential avenues for further research are presented.

13.1 Summary

13.1.1 Pipe organ wind systems

Pipe organ wind systems have evolved over centuries, and the introduction of the electric motor as a prime mover changed the way in which the air supplying the organ pipes was generated. The simple feeder system was replaced by a blower and control valve.

The use of an electrically driven blower to replace the feeder system of generation, may seem a logical progression, but what is not obvious is that the simple mass spring resonant system of the reservoir is compromised by the addition of a control valve, a length of connecting duct and a blower.

Many organ tuners, active in the 1950s & 1960s, preferred the 3-crank electric driven feeder system. It is surprising that the Austin Universal Wind Chest, described in appendix E, is not found in instruments made by other pipe organ builders. The large volume of air, which may approach 120m^3, and control system, provides a good flutter free wind supply.

13.1.2 Pipe organ survey

The survey was conducted by 8 experienced UK organ tuners collecting over 2000 pieces of data in answer to 31 questions. The tuners were asked to determine four, possible degrees of flutter on the sounding pipes. Over 1/3 (39%) of the sounding test pipes had flutter. Only slightly less flutter was detected in the body of the church than at the soundboard. A key finding of the survey was that pipework scale is a contributing factor. Wide scale flute pipes having more flutter than narrow scale string pipes.

Further information about pipework is detailed in Appendix C, including the mechanism for making the pipe speak. Wide jets of air are less stable than narrow jets, so it is not surprising that the wide scale flute pipes with wide mouths, have more organ pipe flutter than narrow scale string pipes.

The limited results relating to tremulants, show that those pipe organs with a working tremulant had a frequency of approximately 4Hz, which is the frequency that the ear is most sensitive to amplitude modulation.

13.1.3 Reservoir resonance

The most important feature of any pipe organ wind system, is that the air pressure in the reservoir must remain constant, otherwise the pitch and tuning of the pipes will vary. The simplest form of pipe organ hand blown wind system, consists of a reservoir containing a volume of air fed by a set of hand operated feeders. The air inside the reservoir acts as a spring and is pressurised by the weight of the reservoir top. The reservoir top and any additional weights provide the mass. A hand blown reservoir behaves like a simple, mechanical mass spring system and the comparison of theoretical and actual resonant frequencies of a weighted reservoir was excellent.

The resonant frequency and vibration levels of the reservoir vary as the reservoir wind pressure is changed, with one wind pressure producing a maximum level. At higher and lower wind pressures, the level diminishes as the resonant frequency changes. This is confirmed by the frequency spectra of the sounding test pipe which shows the presence of side-bands on each side of the fundamental, reaching a maximum at the resonant frequency of the reservoir. The magnitude of side-bands on each side of the fundamental, is a good measure of the amount of amplitude modulation. Results for the reservoir top vibration levels correlate with the level of side-bands found on the sounding test pipe. Also, reservoir top vibration levels show a distinct low frequency maximum at the side-band frequency, indicating that the modulation of the sounding pipe is reservoir driven.

With a single rise reservoir, it is also possible to pressurise the reservoir using only external springs. In this situation, the resonant frequency of the reservoir does not change with increasing wind pressure but assumes the resonant frequency of the mass of the un-sprung reservoir top.

Pressurising the reservoir using weights, produces different resonant frequencies and frequency spectra than pressurising the reservoir using springs. Weighted wind systems, have a resonant frequency around 8 to 15 Hz whilst sprung wind systems have a higher natural resonant frequency of 15 to 20 Hz. The combined use of weights and springs is an effective way of tuning the reservoir to reduce reservoir vibration.

The system can be de-tuned by changing the wind pressure (Figure 63) or by removing weights and adding springs (Figure 72A). Alternatively, the reservoir height can be changed (Figure 58).

13.1.4 Blower comparisons

The pipe organ survey shows that all the organs had an electric blower with more 3000 rpm (58%) than 1500 rpm. The 8 blowers tested, represent only a very small sample of the many different blowers commercially available for organ building applications. Construction details are slightly different for each blower. The main difference is impeller geometry and motor speed.

The results for 1500 & 3000 rpm blowers, show a clear difference in reservoir top vibration levels for weighted and sprung systems. The weighted reservoir and 3000 rpm blower combination producing considerably more reservoir top vibration at 10Hz (94dB) than the weighted 1500 rpm sprung combination (82dB). The results also show that reservoir top vibration levels for 1500 rpm blowers, are typically 10dB lower than 3000 rpm blowers at critical frequencies

Blower performance, particularly under no flow conditions, is of great importance in reducing or eliminating organ pipe flutter. Comparison of the 8 blowers, clearly shows that some units excite the reservoir at its resonant frequency more than others and cause levels of reservoir top vibration, sufficient to produce organ pipe flutter. Blower 6, with a very wide impeller, large cut-off, 3000 rpm motor, produces the highest reservoir top vibration levels and organ pipe flutter.

There is a significant difference of almost 20dB in reservoir top vibration levels at a frequency of 10Hz and this variance is confirmed by the levels of side-bands and percentage amplitude modulation for each blower. Examination of the harmonic content of the sounding test organ pipe, shows that each blower produces a slightly different frequency spectrum. The pipe organ builder should be aware that the installation of a new blower to replace a time expired unit, may give rise to small changes in the harmonic content of the pipework. In addition, any new blower may give rise to higher levels of reservoir top vibration sufficient in magnitude to cause unwanted, organ pipe flutter.

13.1.5 Blower blade frequencies and inter-blades

Blade frequencies are produced by all centrifugal fans. Some of the blowers tested produced large blade frequency levels and the variation can be attributed to the cut-off distance, which is small for blowers with high levels of blade frequency and large for blowers with low levels of blade frequency. This was particularly noticeable when a wooden wedge was used to reduce the cut-off distance of blower 6. Adding the wedge increased the blade frequency level by 10dB to an audible level. The siren is an example of a very small cut-off distance that produces a very strong blade frequency.

Despite the very low levels of blade frequencies produced by blower 6, this blower produces the greatest amount of reservoir top vibration and organ pipe flutter. In comparison, blower 4 produces a very noticeable blade frequency at 500Hz yet one of the least amounts of reservoir top vibration and organ pipe flutter.

Five of the blowers have small inter-blades fitted between the main blades. The inclusion of these small inter-blades in an impeller design, is normally to increase output performance. The addition of inter-blades was shown to be slightly counter-productive and of no benefit in reducing organ pipe flutter.

Fans are normally made to standard geometrically similar designs and the Fan Laws are used to calculate volume flow and pressure for different size fans. The results do not determine precisely how improvements to reduce reservoir top vibration can be made to blower design. Improved blower design requires specialist knowledge of Computational Fluid Dynamics (CFD) and is beyond the scope of this research.

13.1.6 Control valves

An important and surprising result concerns the effect that the control valve has on reservoir top vibration and organ pipe flutter. The type and construction of the control valve is significant, with the simple guillotine valve producing the least reservoir top vibration. Using two gate valves to gradually isolate the blower, effectively simulating the feeder system of air generation, the level of reservoir top vibration and the presence of organ pipe flutter is reduced and ultimately, removed. This is clearly shown on the side-band levels on each side of the fundamental frequency spectrum of the sounding organ pipe, and the amplitude of the reservoir top vibrations. The results show conclusively, that organ pipe flutter is generated by reservoir top vibration.

13.1.7 Attenuating devices

Noise radiating from the blower is normally contained by a silencing enclosure. The main source of wind noise in the organ is breakout noise radiating from the duct connecting the blower and the reservoir. To limit this, the control valve is normally fitted at the outlet of the blower silencing cabinet. In this position, the wind pressure in the wind duct is at reservoir pressure. Occasionally, a simple plenum chamber or, more rarely, a commercial splitter silencer is fitted immediately after the blower, to reduce any excessive breakout noise.

The purpose of the experiments with the various attenuation devices was not to determine their noise attenuating properties, but to determine their effect on reducing reservoir top vibration levels and organ pipe flutter. The frequencies of interest are below 50Hz, typically 10 to 20Hz, and very little data is available for commercial silencing devices at these frequencies. More importantly, no data is available relating to how they perform in an organ building application at reducing reservoir top vibration levels and organ pipe flutter. Results show that the amount of attenuation is dependent on the type of device selected. All the devices tested had good noise reduction properties above 25Hz.

For basic duct breakout noise attenuation, the wood duct connecting the blower and the reservoir could include a section of absorbent material in the duct walls. The thicker the absorbent material: the greater the attenuation. The longer the duct: the better the low frequency attenuation.

The small silencer gives reasonable results in attenuating unwanted duct breakout wind noise and has some effect on reducing any low frequency 10Hz reservoir top vibration levels.

The pepper-pot silencer, which is effectively twice the length of the small silencer, produces good attenuation of duct breakout noise, and significantly less reservoir top vibration levels. This was particularly noticeable in the 10Hz region with reservoir top vibration levels reduced by 15dB. Results also show that the device is position sensitive, with the best results obtained in the “mid” position.

The results of the various attenuating devices must be compared with those obtained for the three different control valves, the two external control valves showing reductions in reservoir top vibration levels of 20dB compared with the internal valve. This is not surprising, for the external control valves in the “no flow” condition, effectively isolate the reservoir from the blower and approaches the feeder generated wind system.

13.1.8 The tremulant

The pipe organ survey reported details of only 31 organs with working tremulants and 80% had traditional wind dumper type units. Most of the tremulants had a frequency of approximately 4Hz, which is the frequency that people are most sensitive to amplitude modulation.

The operation of the tremulant can sometimes be problematic. Many of the wind dumping tremulants were designed for use with reservoirs supplied by a feeder system where the only connection to the reservoir is the connecting duct to the soundboard. With blower generated wind systems the control valve, connecting trunk between the blower and the reservoir, and the blower must also be considered. Preliminary experiments with a wind dumping tremulant, outlined in appendix D, show that weighted reservoirs have different response characteristics to sprung reservoirs.

13.2 Conclusions

The pipe organ survey clearly shows the prevalence of organ pipe flutter and the association with organ pipe scale.

This research shows that a pipe organ wind system cannot be regarded as a simple constant wind source but must be considered as also having frequency components that can cause unwanted reservoir top vibration and organ pipe flutter.

Weighted and sprung reservoir systems have different reservoir resonant characteristics and the mechanical mass will always have a dominant effect for weighted reservoirs. Acoustic attenuating devices will be more effective in reducing reservoir vibration for sprung reservoirs, where the mechanical mass is less dominant. Also of significance is the type of control valve and its effects on reservoir vibration.

Blowers are normally selected for static wind pressure and discharge characteristics. Under “no flow” conditions, each pipe organ blower produces different levels of reservoir vibration, which may be of sufficient magnitude to cause organ pipe flutter.

The implication of this research is that the various wind system elements interact and influence the resonant frequency and vibration levels of the reservoir. With this knowledge, the organ builder is more able to understand the significance of each part of the wind system. Using this understanding the organ builder is better equipped to design and construct a wind system that will minimise reservoir resonance and produce flutter free sounding pipework.

13.3 Further Work

13.3.1The dynamic behaviour of pipe organ wind systems

Many subjective pronouncements have been made by organ builders regarding the “best” type of wind system. Is it a weighted or sprung, double or single rise reservoir combination that provides the “best” wind system?

This question can only be answered by conducting further research to make direct comparisons between the various types of wind systems. Using a series of listening tests, based on a short piece of music played on a single set of pipes, it may be possible to determine the dynamic characteristics of weighted and sprung wind systems to see any discernible differences. The difference between single and double rise reservoirs and the effects of internal and external control valves also need to be considered.

13.3.2 Mechanical and acoustic model

The hand-blown reservoir behaves like a simple mechanical mass spring system. The mass of the test reservoir top is approximately 75kg.

Adding a blower transforms this simple mechanical system into one that has both, mechanical and acoustic, properties. The acoustic mass of the air inside the test reservoir is approximately 0.7kg, which is a factor of 100 less than the mechanical mass. This means that for a weighted reservoir wind system, the mechanical mass will always have a dominant effect. For a sprung reservoir wind system, the mass of the reservoir top is approximately 15 to 20kg, and possibly even smaller for chest regulated wind systems, in this application the mechanical system is less dominant.

In addition, the inclusion of an electrically driven blower necessitates the addition of a control valve which is controlled by a connection to the reservoir top. The feedback of this secondary vibration system is very difficult to determine, with each type of control valve having a different response.

Further research is needed to analysis the wind system from both a mechanical and acoustic perspective and a model that satisfies both conditions developed.

13.3.3 Blower improvements

Most fans are designed using the Fan Laws and the principal of geometrically similar modelling.

Further research using CFD computer software could develop new impeller and case designs with improved “no flow” characteristics. The improved designs could then be tested to verify that reservoir top vibration levels are improved.

13.3.4 Improved silencer and attenuator designs

Commercial silencers are designed for the suppression and control of air generated noise.

Further research, using computer modelling techniques, could investigate how existing designs could be improved to reduce low frequency reservoir top vibration levels. The research also needs to consider the use of alternative fill materials, such as meta-materials. From this work, a new design that best suits organ building applications and the reduction of reservoir top vibration and organ pipe flutter may evolve.

13.3.5 System resonance using an impact hammer

The resonance and modal response of a vibrating system can be determined using an impact hammer, several accelerometers and computer analysis software.

Further work could use this technique to determine the resonant response of the reservoir and wind system for weighted and sprung wind systems including various control valves. Also, to be considered is the effect that various attenuating devices have on improving the dynamics and damping properties of the wind system.

13.3.6 Tremulants and system resonances

The pipe organ survey shows that only 31 of the 83 organs surveyed had working tremulants. For the tremulant to work effectively it must be able to excite the natural resonant frequency of the reservoir and other parts of the wind system. For a feeder system this involves only the reservoir, connecting duct to the soundboard and the soundboard bar. For the blower generated wind system, the dynamic response of the control valve, the connecting duct to the blower and the blower inlet flap must also be considered.

Further research is needed to investigate the effectiveness of various types of tremulant and their effect on weighted and sprung wind systems including Schwimmers and modern wind chest regulation.

13.3.7 Control valves

This research shows a significant difference between internal and external control valves.

Further research is needed to understand more fully the relationship between the control valve and the dynamic behaviour of the reservoir top. From this research a new type of control valve may evolve that suppresses reservoir top vibrations to a level obtained by hand blowing. The “butterfly valve” used by Austin on their universal wind chest may be a good starting point.

14 Appendix A Fan Design

14.1 Early Fan Designs

Mine ventilation machines are described in De Re Metallica [56], but one of the earliest publications detailing the mathematical relationships between the various elements of a centrifugal fan was published in 1905 Kinealy [57]. These early fans replaced the traditional use of maintaining a fire at the bottom of the mine shaft to produce an updraft. This type of fan very rarely had a casing unless it was necessary to protect the wheel from inclement weather.

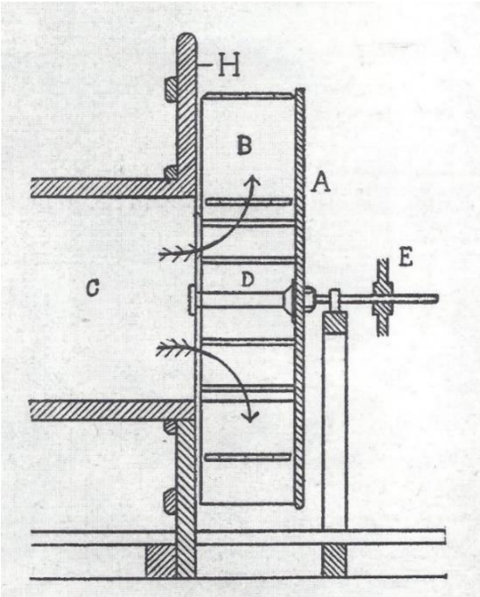


Figure 155 A typical early ventilation fan (After Kinealy [57])

Figure 155 shows a typical arrangement. C is connected to the top of the mine shaft and rotating the wheel A, creates a flow of air through the blades B, which ventilates the workings. The wheel had to be a close fit to the wall and it was not uncommon to be 25 feet or more in diameter.

Figure 156 shows the next stage in the development of the fan, attributed to a Frenchman named Guibal. The wheel is now enclosed between two solid sides. Air is drawn in the inlet A by the rotating blades B and passes through the outlet D. A blind E moves in the slot F, allowing regulation of the air though the fan. Again, it is very important that the sides of the housing fit very close to the wheel so that the air is forced through the outlet H. It was found that for every fan there was a particular outlet opening H which gave the greatest efficiency.

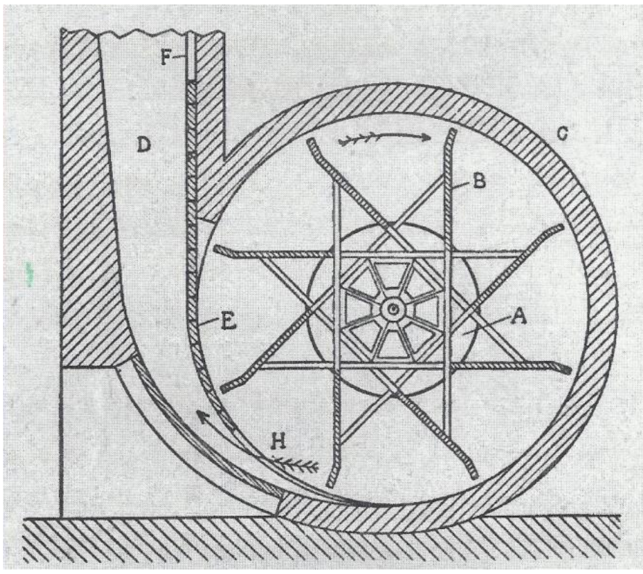


Figure 156 Section through Guibal fan (After Kinealy [57])

The Guibal fan was improved by mounting the wheel off the centre of the casing creating a scroll with the wheel closer at one point, gradually increasing the distance. Figure 157 shows a fan of this design. The object of the spiral casing is to enable the air to be discharged from all points on the periphery of the wheel allowing the use of smaller fans and increased speeds. This type of fan is still the basis of today's centrifugal fan.

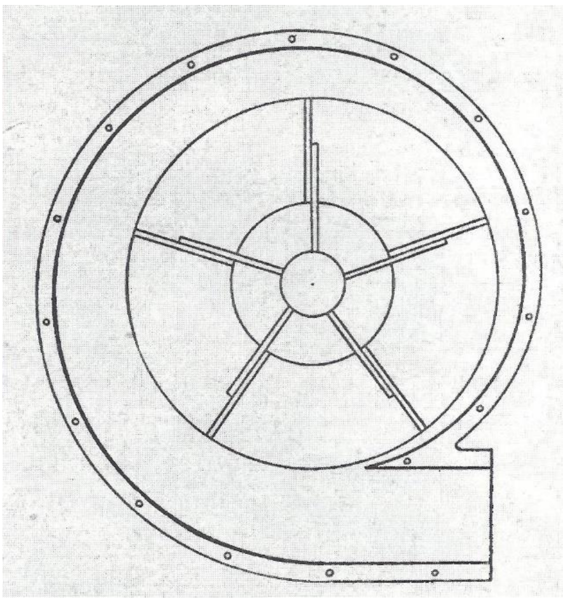


Figure 157 Section through scroll fan (After Kinealy [57])

14.2 Modern fan design

Since Kinealy's publication in 1905 other textbooks relating to the design of fans have been written. In England the subject was very comprehensively covered in 1952 by Woods of Colchester Ltd [58]. After many editions in German, the standard work on Fan Design by Eck [59] was translated in 1972 into English. Other texts by Osborne [60], Jorgensen [61], Mcpherson [62], Bleier[63] and Cory[64] have continued to expand the knowledge base. The greatest influence on fan performance has no doubt been the development of the electric motor as a prime mover. The early fans running at slow speeds had to be big. More power from smaller motors together with the introduction of motor speeds of 3000 rpm has made for smaller and quieter fan units.

14.3 Basic fan types.

The maximum flow rate and pressure usually dictates the type of fan to be used. Organ building fans, which are usually called “blowers”, need to supply a range of flow rates and wind pressures from 50 to 200mm water gauge. Higher pressures are sometimes needed to supply high pressure reeds.

There are two basic fan classifications:

Axial flow

Centrifugal flow

In an axial flow fan, the movement of air is in an axial direction. Axial flow fans are mainly for low wind pressure applications and not suitable for organ blowing applications.

In a centrifugal fan the air is moved from the centre of the fan to the outside of the impeller, the centrifugal force causing the moving air to compress. Centrifugal fans are capable of producing higher wind pressures which make them suitable for organ blowing applications.

There are six basic centrifugal fan designs with different blade geometries. Each type of blade has advantages and disadvantages and is generally suited to a particular application. Figure 158 shows the various blade geometries together with the maximum efficiency that is attainable by each blade type. It is not surprising that aerofoil blades produce the lowest noise levels and have the highest efficiency of all centrifugal fans compared to the more primitive and brutal simple radial blade fan.

Aerofoil blades (AF).	Radial tip blades (RT).
Backward curved blades (BC).	Forward curved blades (FC).
Backward inclined blades (BI).	Radial blades (RB).

Figure 158 Fan impeller types (After Bleier [63])

14.4 Fan performance Curves

The corresponding performance curves for each type of fan are shown in Figure 159, 160 &161

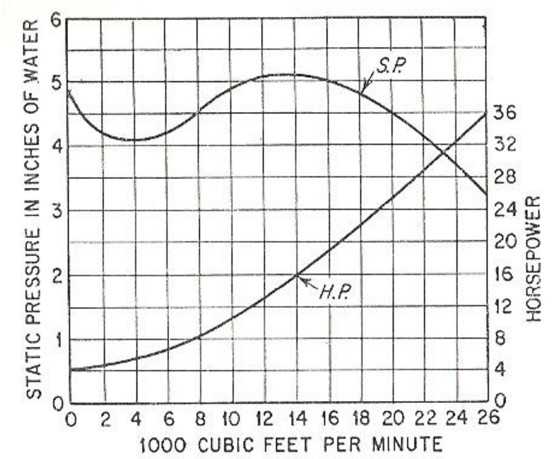


Figure 159 Performance curves for forward curved blades

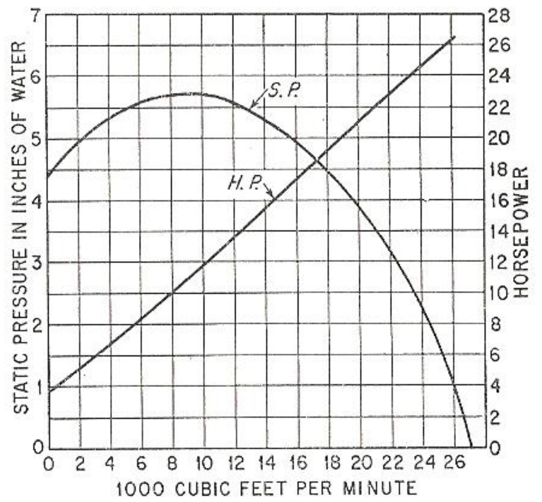


Figure 160 Performance curves for radial curved blades

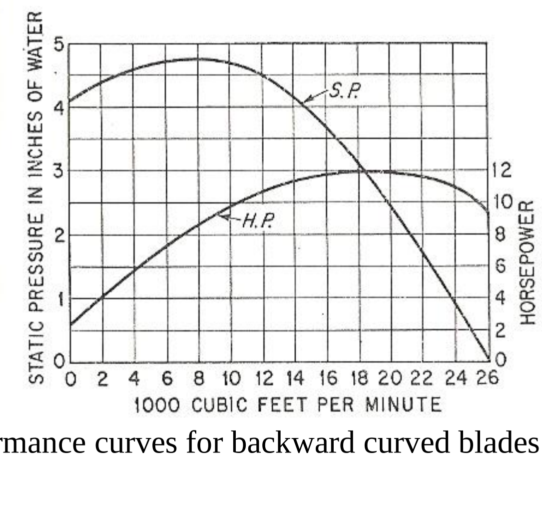


Figure 161 Performance curves for backward curved blades (After Bleier [63])

14.5 Fan outlet velocity diagrams

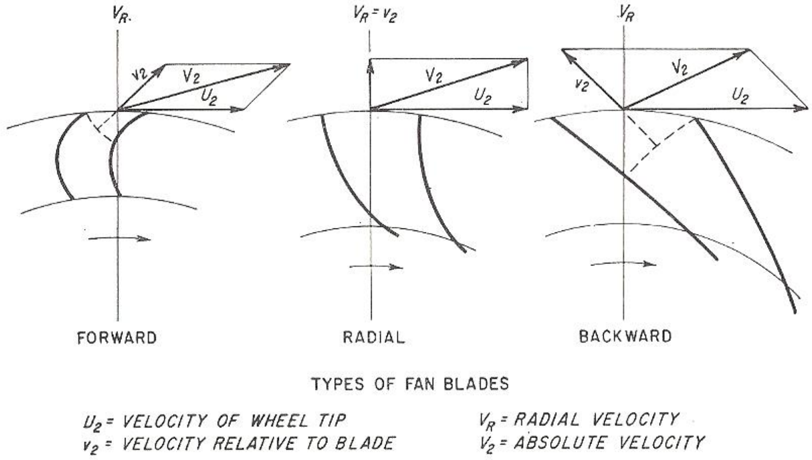


Figure 162 Fan outlet velocity diagrams (After Bleier [63])

Figure 162 shows outlet velocity diagrams for the three basic fan types. The forward curved blade fan produces the highest absolute velocity V2. Generally, this type of fan for a given capacity runs at a lower speed and occupies less space than the other two types. The trade-off is higher turbulence and noise. Care must be taken to ensure that the electric motor is of a sufficient size to cope with the rise in horse power with the rise in output flow rate. The backward curved blade fan has the smallest absolute velocity V2 but for a given airflow and pressure the motor speed needs to be twice that of a comparable forward curved bladed fan. The advantage is that the aerodynamic noise is lower in frequency and intensity than that of the forward curved blade fan, because the relative air velocity between the blade and the fan housing is reduced. The noise characteristics of the radial blade fan lie between the other two types of fan. It is very important that fans used for organ blowing applications are as quiet as possible and are best served by the backward curved blade design.

14.6 Fan losses

The theoretical performance of a fan does not take account of any losses that occur in the real world. The losses are such that the theoretical pressure volume curves are not achieved. This under performance is due mainly to friction and shock losses. Figure 163 shows the actual pressure and flow rate relationship for a backward bladed fan. The maximum flow rate occurs at zero pressure when the outlet is open circuit with no external resistance. Friction losses occur on the faces of each blade and this is a good reason for using aerofoil blades to limit such effects. Shock and separation losses occur at the inlet where the air makes a 90 degree change in direction. Other losses associated with the turbulent flow at the inlet can be partially overcome by using inlet guide vanes.

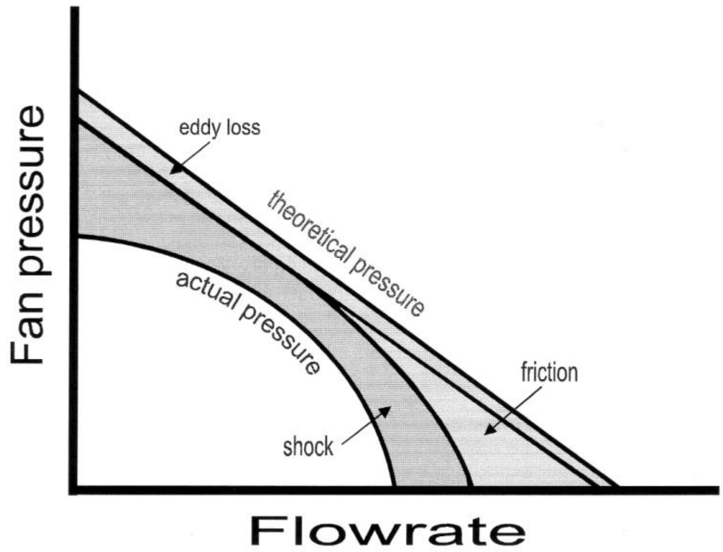


Figure 163 Pressure and flow rate for backward bladed fan (After McPherson [62])

14.7 Duty cycle

Duty cycle is an important consideration when selecting a fan. Figure 164 shows the power consumed by the three types of fan for varying airflow. The forward blade is normally only used for a fixed duty cycle so that the motor is not overloaded as the airflow increases. The radial blade has a less pronounced increase in power for increasing airflow. The backward blade consumes almost constant power for wide variations in airflow. This safeguard against motor overload for varying airflows is an important characteristic of the backward blade fan. This is necessary for pipe organ blowing applications, where for long periods of time the wind control valve is closed, and when the demand for full organ is needed the motor is not overloaded. It cannot be over emphasised, that for large installations, the non-overloading power characteristics and the steepness of the pressure curve at high flow rates, are a major factor for using backward bladed fans.

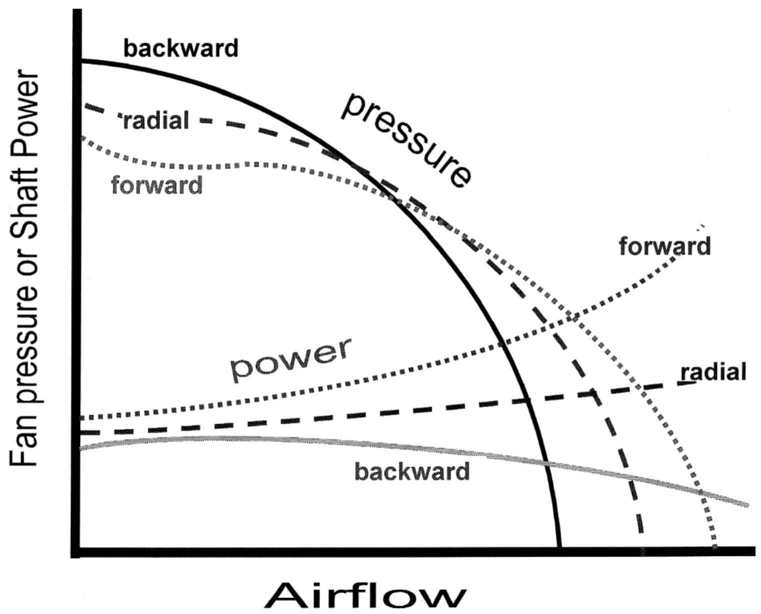


Figure 164 Power consumption for forward, radial and backward bladed fans (After McPherson [62])

14.8 Aerofoil impeller fans

Figure 165 shows a typical aerofoil impeller with inlet and outlet velocity diagrams. The associated performance curves are shown in Figure 166 for a 27 inch diameter impeller with a direct drive from a 5hp 1160 rpm motor. Significant is that the minimum noise level of 76dB occurs at a point of maximum mechanical efficiency of 88%. Aerofoil fans are expensive to manufacture and often fans with backward curved blades or backward inclined blades are used as a cost-effective alternative with associated loss in efficiency and increased noise levels.

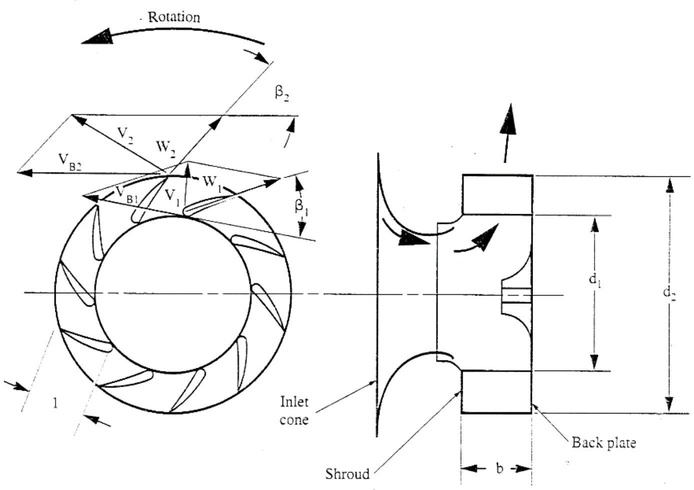


Figure 165 Typical airfoil inlet and outlet velocity diagrams (After Bleier [63])

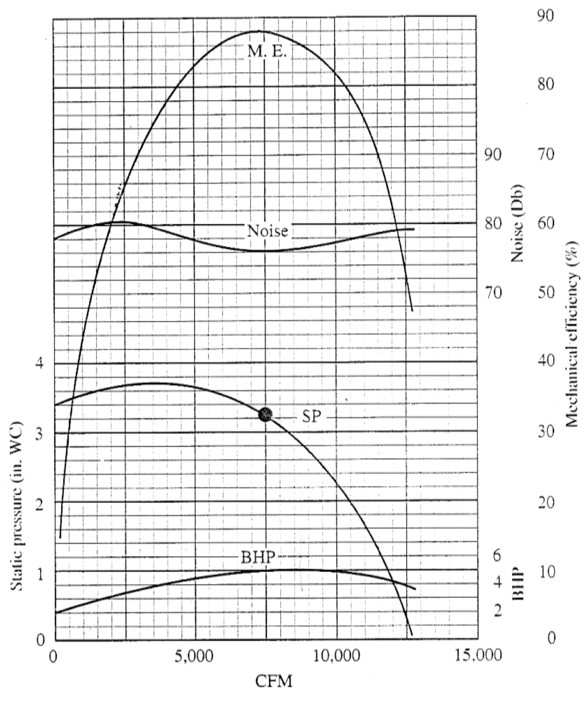


Figure 166 Aerofoil performance curve (After Bleier [63])

14.9 The Fan Laws

Pump and fan manufacturers use dimensional analysis to design a range of pumps and fans which are geometrically similar.

It is not always possible to find a pump or fan that exactly satisfies the requirements of a particular application. The volume or pressure output of the fan may be just too small. In this situation knowing the relationship between the output volume, pressure and speed of the fan, it is possible to adjust the speed to achieve the extra performance. The relationship between volume, pressure and speed is shown below.

Air volume varies directly with speed.

$$\frac{cfm2}{cfm1} = \frac{rpm2}{rpm1}$$

Pressure varies as the square of the speed

$$\frac{sp2}{sp1} = \left(\frac{rpm2}{rpm1}\right)^2$$

In addition to the volume and pressure relationship it is important that the power requirement of the modified unit does not exceed the capacity of the motor.

Brake horse power varies as the cube of the speed

$$\frac{bhp2}{bhp1} = \left(\frac{rpm2}{rpm1}\right)^3$$

Example

A blower has a 1hp motor running at 3000 rpm and a maximum pressure of 130mm WG for a volume of $22m^3$. The organ builder wishes to increase the pressure to 140mm WG.

$$\frac{140}{130} = \left(\frac{rpm2}{3000}\right)^2$$

The new speed is 3113 rpm or an increase from 50Hz to 51.9Hz and the power requirement is increased to 1.12hp.

15 Appendix B Pipe organ survey

The pipe organ survey was subject to the approval of the University of Salford Ethics Panel. The paperwork was submitted on 14.10.2016 and conditional approval was given on 13.12.2016. The conditions were implemented, and the pipe organ survey packs sent to 23 selected pipe organ builders on 17.03.2017 by first class post together with a stamp address envelope to return the completed survey forms.

The pack consisted of:

An invitation letter.	AJT4 V3 17.03.17
An information sheet.	AJT5 V2 13.12.16
A consent form.	AJT2 V1 29.09.16
A pipe organ survey questionnaire.	AJT6 V4 17.03.17

The survey was returned by 8 organ builders

15.1 Pipe organ survey invitation letter

University of Salford MANCHESTER

School of Computing, Science and Engineering

The Effects of Centrifugal Blowers and Reservoir Resonance on Organ Pipe Flutter

Pipe Organ Survey Invitation

Dear

You are probably aware of my research into an area of organ building that has been talked about for many years. This is an opportunity for you to participate in assisting in understanding more fully the effect of organ pipe flutter by taking part in a pipe organ survey of approximately 100 instruments of varying sizes and locations.

If you decide to support the research you will be asked to provide details of 5 or 6 two manual and pedal organs in your care of varying size and location. I have tried to keep the attached questionnaire simple yet contain sufficient information to be useful in determine the extent of organ pipe flutter.

No special arrangements need to be made. I anticipate that the questionnaire will be completed as part of a planned tuning visit. Space has been left for any comments which should be completed to highlight any specific irregularities.

Finally, I would be grateful if you would return the completed survey forms together with the informed consent declaration in the self-addressed envelope as soon as you have completed the survey.

Yours sincerely

Alan Taylor

AJT4 V3 17.03.17

15.2 Information sheet

School of Computing, Science and Engineering

The Effects of Centrifugal Blowers and Reservoir Resonance on Organ Pipe Flutter

Pipe Organ Survey Information Sheet

What is this survey about?

The survey is to investigate the occurrence of organ pipe flutter in 100 pipe organ installations.

You will be asked to fill in a questionnaire providing information on the presence of organ pipe flutter on selected organ pipes. Further information will be recorded relating to the type of wind system, blower type and general information relevant to each specific instrument.

What will happen to my data?

Your data will be stored electronically and anonymously without any identifying information. The data will be archived for up to three years from the submission of my thesis and held on password protected media. If you withdraw from the study, all your data will be destroyed.

Can I stop at any time?

Yes. You are under no obligation to complete the study. You are free to stop at any time for any reason and you need give no explanation.

What if I have any questions?

You are free to ask questions before, during or after the session. If you need to contact me afterwards regarding the session, I can be contacted at a.taylor22@edu.salford.ac.uk

AJT5 V2 13.12.16

15.3 Consent form

School of Computing, Science and Engineering

The Effects of Centrifugal Blowers and Reservoir Resonance on Organ Pipe Flutter

Informed Consent Declaration

1.	I have read and understood the information sheet associated with this project provided to me by the researcher.	
2.	I have been given opportunity to ask questions to clarify the activity and the nature of my participation.	
3.	I voluntarily agree to take part in the project.	
4.	I understand I can withdraw from the project and activity at any time for any reason and do not need to specify any reason for withdrawal.	
5.	The confidentiality and anonymity of my participation has been explained to me.	
6.	The type of data recorded in the study and how it will be used has been explained.	
7.	I agree to the data being stored for analysis and am satisfied that it is stored anonymously.	

Participant

Name:

Signature:

Date:

Researcher

Name:

Signature:

Date:

AJT2 V1 29.09.16

15.4 Pipe Organ Survey Questionnaire

University of Salford MANCHESTER

School of Computing, Science and Engineering

The Effects of Centrifugal Blowers and Reservoir Resonance on Organ Pipe Flutter

Pipe Organ Survey Questionnaire

Section 1 Organ Details

Location						
Builder		Date				
Number of stops	Swell	Great			Pedal	
Position		Size of building	Small.		Medium	Large.
Approximate reverberation time in seconds		Less than 1	1 to 3.			over 3
Temperament		Equal			Other.	

Comments

Section 2 Pipe Flutter Details

No Flutter =NF **Slight flutter = SF** **Moderate Flutter =MF** **Extreme Flutter = EF**

Organ Pipe		Scale	Soundboard Degree of Flutter	Body of the church Degree of Flutter
Great 8ft open 6”	c 49			
Great 4ft principal 6”	c 37			
Gt or Sw 8ft stp flute 6”	c 49			
Gt or Sw 8ft string 6”	c 49			

Comments.

1

Section 3 Wind System Details

SR = Single Rise DR = Double Rise W = Weighted Sp = Sprung

Example: A Swell 4 ft. x 6 ft. double rise weighted reservoir = 4x6 DR W

	Break down	**Swell**	**Great**
Reservoir type & construction			
Full Reservoir height inc well (inches)			
Approx. Wind pressure (inches) WG			
Cut-off valve details			
Blower make		Motor HP.	1 or 3 phase
Blower speed		Low 1500 rpm	High 3000 rpm
Blower location			
Approximate distance from blower to reservoir (feet)			

Comments

Section 4 Tremulant Details

Swell stop selected for use with the tremulant				
Type of tremulant				
Speed & effectiveness of the tremulant	Slow	Fast	Good	Poor

If possible, please give the approximate number of vibrations per second.

Comments

Name Date

Signature

AJT6 V4 17.03.17

15.5 Ethical Approval

University of Salford
MANCHESTER

Research, Innovation and Academic Engagement Ethical Approval Panel

Research Centres Support Team
G0.3 Joule House
University of Salford
M5 4WT

T +44(0)161 295 5278

www.salford.ac.uk/

9 January 2017

Alan Taylor

Dear Alan,

RE: ETHICS APPLICATION STR1617-15: The Effects of Centrifugal Blowers and Reservoir Resonance on Organ Pipe Flutter

Based on the information you provided, I am pleased to inform you that your application STR1617-15 has been approved.

If there are any changes to the project and/ or its methodology, please inform the Panel as soon as possible by contacting S&T-ResearchEthics@salford.ac.uk

Yours sincerely,

[redacted]

Dr Anthony Higham
Chair of the Science & Technology Research Ethics Panel

16 Appendix C Pipework

16.1 Pipework materials.

Organ pipes are made from a variety of materials.

16.1.1 Pipe metal.

Pipe metal is an alloy of lead and tin and is used for most metal organ pipes. The minimum amount of tin is 12% which makes for pipes that are easily deformed. To give additional strength, particularly on larger pipes, 30% or 40% of tin is added. This gives a mottled or spotted appearance and is referred to as spotted metal. For more prestige organs, the case pipes may be made from 99% tin which gives a bright silver appearance. Pipe metal is the material of choice because it is easily worked and jointed. Any offcuts are saved and added to the melting pot for reuse. Pipe metal is cast into sheets 600 to 900mm wide and 3m long on a casting bed. The rough cast sheets can be used without any further operations or the sheets can be planed to thickness by hand or machine. In some instances, the sheets are hammered to work harden the material. Terry Shires[65] recalls a project to restore some organ pipes originally made in 1653 by a Germany Organ Builder. It was important that any new material was compatible with the original so a small sample was sent for analysis. The results surprised the client. The material had the same analysis as lead flashing and included many impurities that gave the pipes their strength.

16.1.2 Zinc.

Zinc is used as a replacement for pipe metal for larger pipes where strength is of primary importance. Zinc is generally too hard to use for the mouth area of the pipe, so this area is made from pipe metal. Zinc is normally left with its natural dull finish. Alternatively, if zinc is used for case pipes the surface is polished and lacquered to give a bright finish. This is a poor man’s substitute for pure tin case pipes. Often zinc case pipes are painted and, in some applications stencilled.

16.1.3 Copper.

Copper, particularly flamed copper, is used to make case pipes. This is used mainly for artistic and decorative reasons to give interest to what may be a rather uninteresting case. Sometimes copper or brass is used for reed resonators, particularly the resonators of en-charmade reeds, where appearance is of prime importance.

16.1.4 Wood.

Many species of wood are used to make organ pipes. Wood is used for large pedal organ pipes where strength and self-supporting properties are important. The bottom C of a large 32ft pedal open wood is approximately 10m long and weighs of over 1 tonne. The pipe is made from 75mm thick timber with a plan size of 700 x 750mm. Organ builders have their own preference and use a selection of soft and hardwoods for different stops. Softwoods include various species of pine. Hardwoods include oak and mahogany. Small wood pipes are either left natural or polished.

16.1.5 Paper

Mark Wicks[66] describes in detail how to make organ pipes from paper. His book was intended for amateur organ builders who wished to construct a pipe organ with a limited budget.

16.2 Classes of pipes

There are two main types of pipes.

Flues

Reeds

16.2.1 Flue Pipes

Most organ pipes are classified as flue pipes and are relatively easy to make from pipe metal or wood. Figure 167 shows the main parts of a typical metal and wood pipe.

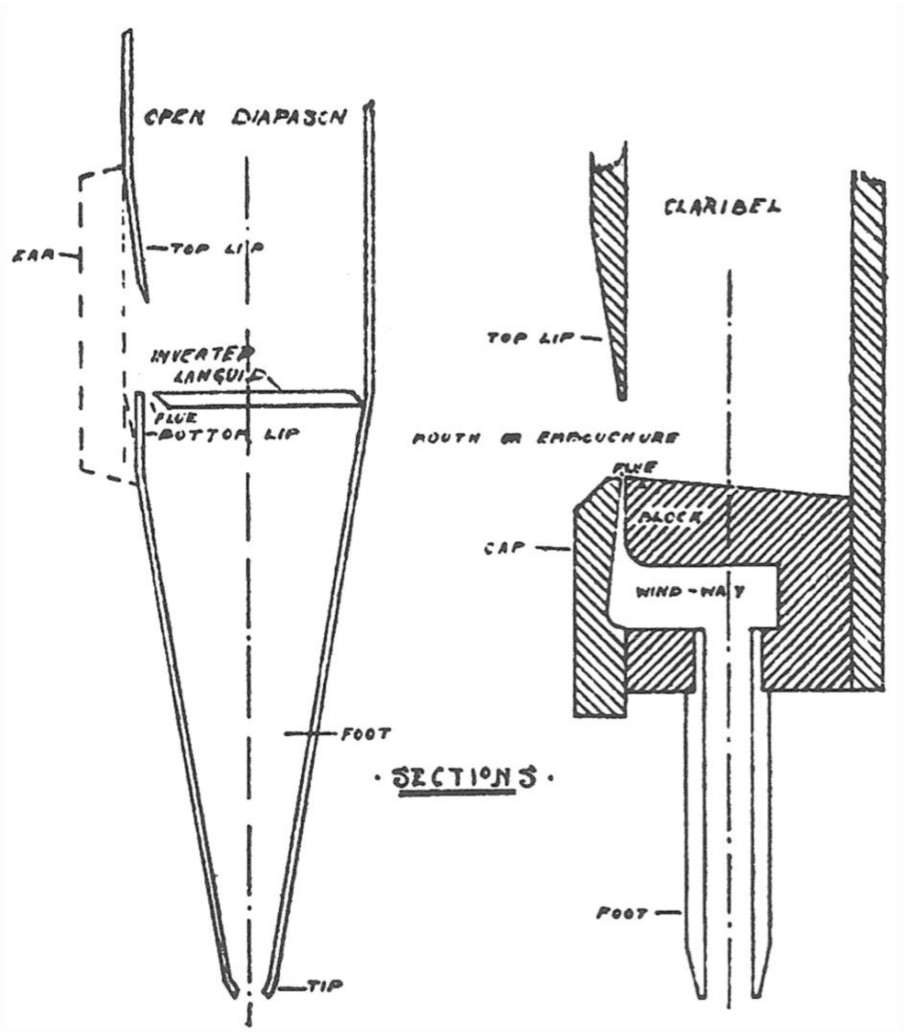


Figure 167 The main parts of a flue pipe (After Barnes [1])

Flue pipes can be open or closed:

Open -	half wavelength body
Closed (stopped) -	quarter wavelength body

Using $\lambda = c/f$

where λ = wavelength. c= speed of sound. f = frequency.

An 8ft open pipe has a fundamental frequency of 65Hz.

Wavelength of 343/65 = 5.3m or 17.39ft.

The speaking length is approximately 8.7ft (8ft) neglecting end correction.

An 8ft closed pipe with a fundamental frequency of 65Hz has a speaking length of approximately 4ft.

Table 18 shows the fundamental frequency of the bottom note of an 8ft stop (65 Hz) and the top note (note 61 2080Hz). The highest fundamental frequency of the top note of a 2ft open stop is 8320Hz and has a speaking length of approximately 10mm.

Note number			ft	frequency Hz
1			8	65
13	1		4	130
25	13	1	2	260
37	25	13	1	520
49	37	25	$^1/_2$	1040
61	49	37	$^1/_4$	2080
	61	49	$^1/_8$	4160
		61	$^1/_{16}$	8320

Table 18 Pipework frequencies

Harmonically, open pipes support all harmonics, whilst closed pipes only support odd harmonics. Going down in frequency the lowest note of a 32ft open wood is approximately 16Hz and is more of a rumble than a distinctive fundamental sound.

16.2.2 Tuning

Tuning of organ pipes is necessary at periodic intervals. The purpose of tuning is to adjust the body length so that the pipe sounds the correct frequency. Small pipes have very little leeway and take more skill to tune than larger pipes which have more latitude. There are two main methods of tuning organ pipes, cone tuning and tuning slides. Case or front pipes use a tuning flap on the rear of the pipe to adjust the length. Stopped or closed pipes use adjustable stoppers to adjust the length of the pipe. The various methods of tuning are shown in Figure 168

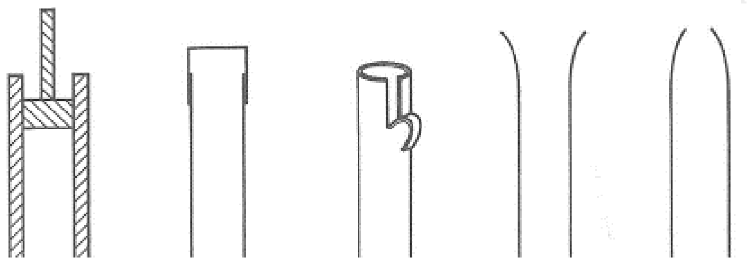


Figure 168 Methods for tuning organ pipes (After Audsley [3])

16.2.3 Pitch

The pitch of the organ is set by cone tuning the pitch pipe which is normally the 1ft c of the 4ft principal stop. The next step is to tune each note in the 1ft octave by reference to some scale and temperament. Modern instruments use equal temperament. Older instruments use a mixture of temperaments. Once the first octave has been tuned to satisfaction the rest of the stop is tuned in octaves until all the pipes of the stop are in tune. Each other rank is tuned to single notes of the 4ft principal starting at middle c and finishing at treble b. Once this middle octave of the stop has been tuned the rest of the stop is tuned in octaves until the whole stop is tuned. Many pipe organs were built with a pitch higher than standard A =440 Hz. The organ in Peterborough Cathedral, A =451 Hz, has recently been rebuilt and the pitch lowered to A=440 Hz so that the organ can play with instrumental players.

16.2.4 Temperament

The pipe organ is one of the very few instruments where the sound is continuous. Benade[67] addresses this point and the systematic beats that arise between certain intervals on an instrument that produces sustained tones. This is not a problem when octave related notes are played because they are tuned perfect. The problem arises when other intervals are played. Pre-Bach temperaments allow a limited number of keys, (c, g, d, a, f & b flat), to be used before “wolf” notes occur. Bach had his organs tuned in a different way that allowed more key signatures to be used without the occurrence of “wolf” notes. This is achieved by flattening the fifths and sharpening the fourths. This produces a slight beating when an interval of a fifth or fourth is played and is further exaggerated if the third is added.

Bach composed his famous 48 preludes and fugues for the well-tempered clavichord to demonstrate the advantages of this method of tuning Czerny [68]. They were written in two sets, each contains 24 pairs of preludes and fugues.

This question of temperament and angry thirds is examined by Berg[69], Backus[70] and John Norman[71]. Figure 169 shows John Norman’s results for four different temperaments. Berg also describes how composers used ornaments and trills to mask the out-of-tune-ness. The question of which temperament will best suit the room is becoming more relevant. Equal temperament is favoured by organ tuners who mostly tune organs in reverberant buildings where the treble filter caused by air absorption tends to blunt the aggressiveness of the upper work. The tuners of Westminster Abbey and St Paul’s Cathedral favour equal temperament. Alternatively, in a more intimate and non-reverberant building the problems with angry thirds in common keys is intolerable. In such a situation Young’s tuning was used for the organ in St Paul’s Cathedral crypt.

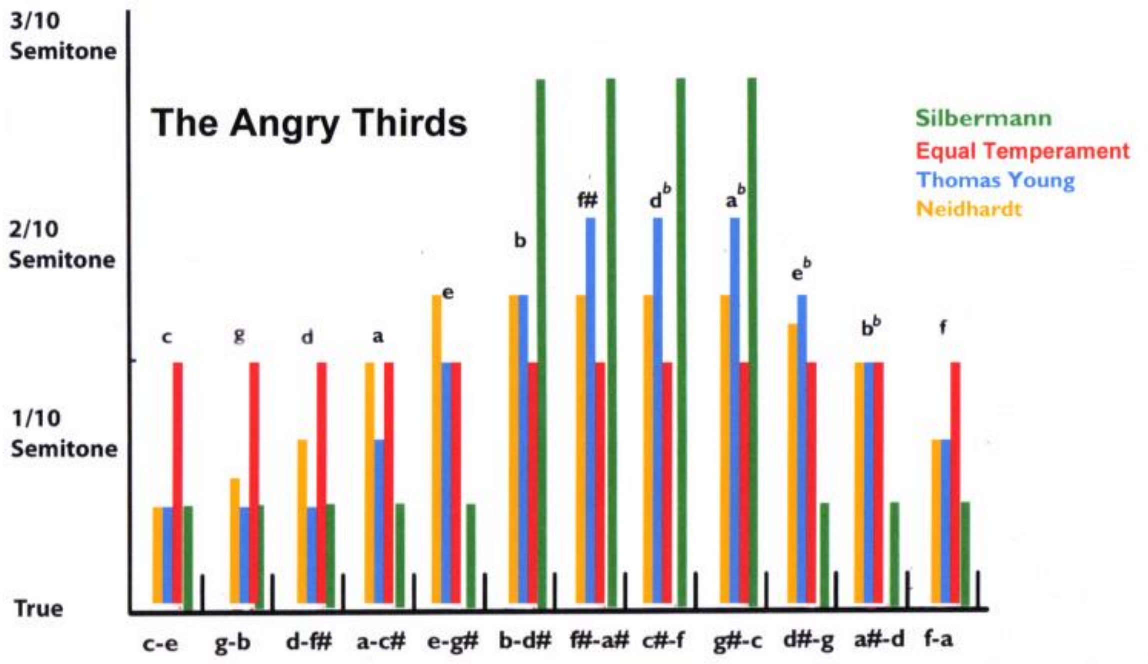


Figure 169 Angry Thirds (After Norman [71])

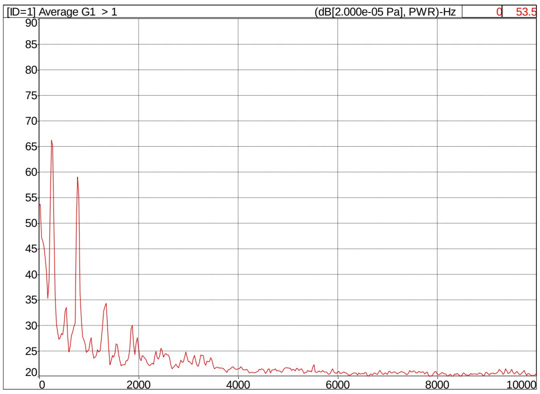
16.2.5 Siting

No room has the same acoustic properties, and this is one of the reasons why it is difficult to find identical sounding organs. The siting of the organ is import if the pipework is to speak freely into the room. The question of matching organs to acoustics is considered by Norman [71]. West End positions with small case depths are to be preferred. The use of tone cabinets to house the pipes also assists projection of the sound. From this perfect solution any other design is a compromise. Many modern concert halls can be problematic when the main design criterion is to achieve the best possible conditions for orchestral music with a reverberation time approaching 2 seconds. The question of making an organ that will live up to expectations is far from an easy task. To overcome site conditions normal scales may need to be increased or decreased by a few pipes. Pipe organs sound better in cavernous cathedral spaces with large reverberation times.

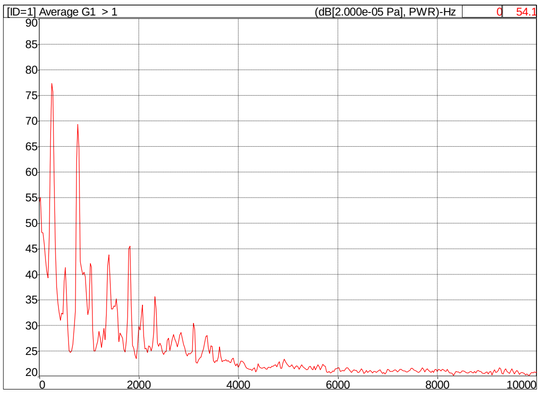
16.2.6 Harmonic content, quality and frequency spectrum

The acoustic spectra of organ pipes was first described in some detail by Boner[72], using what may be considered today as very primitive measuring equipment. Boner placed the organ pipes under test on a 24 ft high outdoor tower. Using acoustical instruments described by Went [73], and techniques developed by Hall [74] and Saunders [75] used for analysing violins. Boner analysed 21 open organ pipes which included diapason, string and chorus reed pipes. He was able to achieve repeatable measurements with a 3dB degree of accuracy. Four American organ builders loaned the pipes and assisted with the tests. Pipe details recorded pipe length, cross section, cut-up, mouth width, nicking, shape, wind pressure, reed tongue and shallot details.

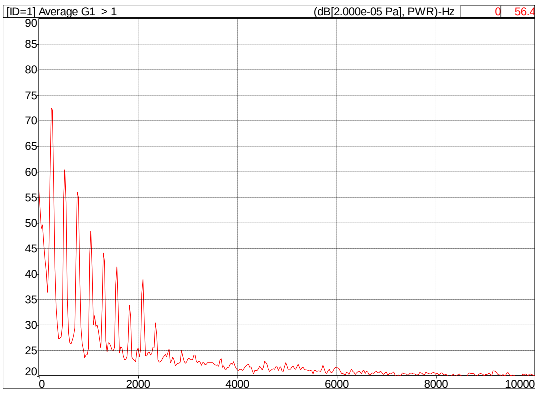
The subject of organ pipe analysis was further advanced by Jones [76] who reviewed work by Boner and Mokhtar [77]. Jones also mentions work carried out by the psychologist Carl Stumpt [78] who investigated how the starting and ending transients of musical tones are important in distinguishing various musical instruments. More specific related work was carried out by Trendelenburg, Franz and Thienhaus [79] who studied the initial effects in organ pipes. In the case of a Trumpet reed pipe it was found that the sound developed very quickly, typically less than 0.1 seconds. Conversely, the development of sound for a principal flue pipe, with a similar frequency of 61Hz, took 0.5 seconds to fully develop. Frequency spectra for six common organ pipes are shown in Figure 170.



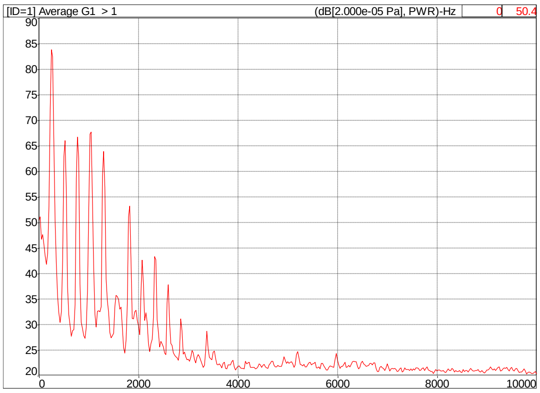
8ft stopped metal flute



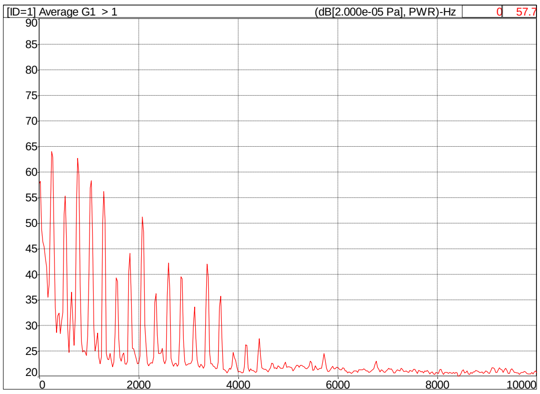
8ft stopped wood flute



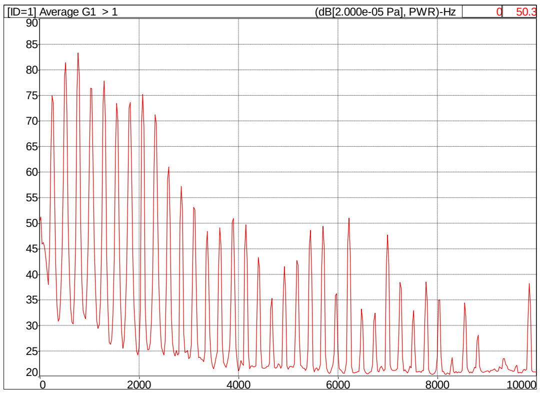
8ft gemshorn



4ft principal



8ft string



8ft trumpet

Figure 170 Frequency spectra for six common organ pipes

16.2.7 The organ pipe drive mechanism.

Brown [80] showed how the organ pipe vortex action and frequency depended on the distance from the slit (flue) to the wedge (cut-up). Figure 171 shows as the wedge moves farther from the slit, the pitch falls gradually except at certain points where it jumps upward. He found that there are four "stages" in each of which the change in pitch is continuous. Photographs of the four stages are shown in Figure 172. The edge tone of a stage appears to depend on the number of vortices between the slit and the wedge. As the wedge moves away from the slit the distance from one vortex to the next increases and the pitch of the sound falls.

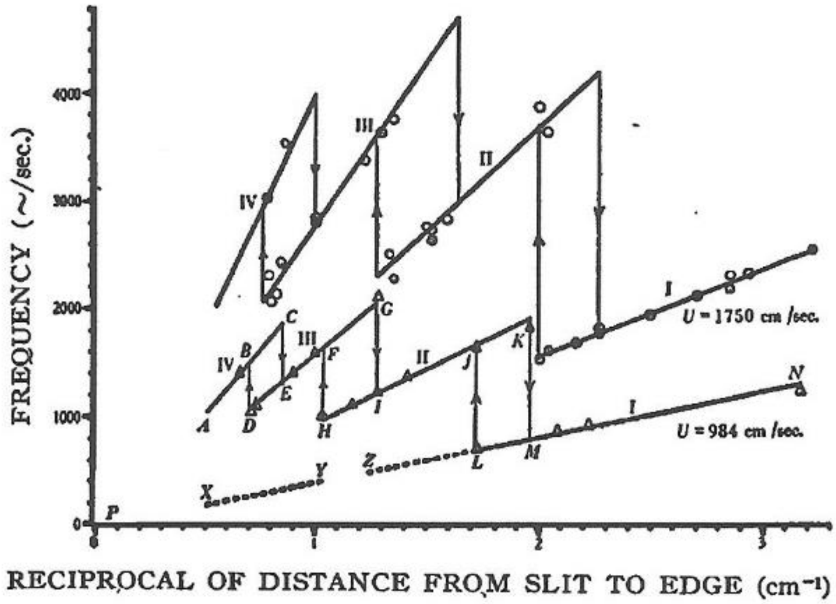
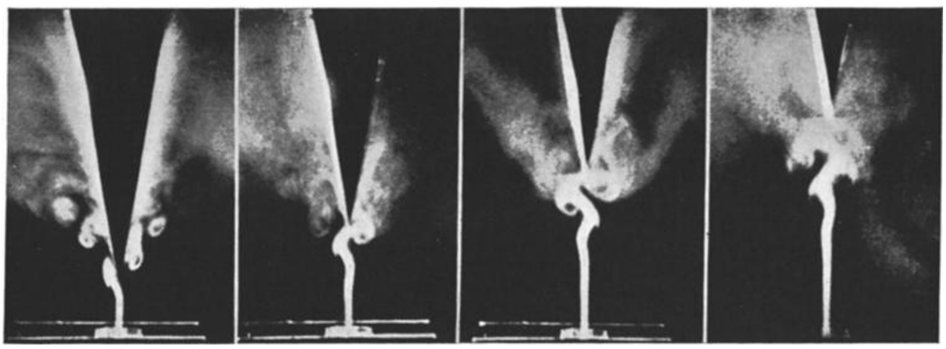


Figure 171 The four stages in pitch of an edge tone (After Brown [80])



Stage 1 h =0.81 cm Stage 2 h = 1.73 cm Stage 3 h =2.65 cm Stage 4 h =3.5 cm

Figure 172 The four stages of vortex formation at an edge, h is the distance from slit to edge (After Brown [80])

Klug [81] conducted an experiment that brought in from the side a "disturbing wedge". The wedge was set at some chosen level and was then caused to move inward from the side until it affects the sound. Details of the apparatus are shown in Figure 173.

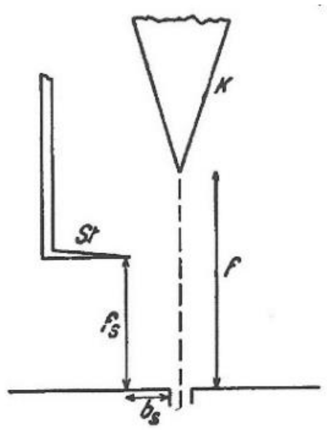


FIG. 10. The "disturbing wedge." The blowing wind comes up through the slit to the right of b_s and strikes the wedge K. St is the disturbing wedge. (Klug.)

Figure 173 The disturbing wedge (After Klug [81])

Figure 174 shows one set of results. A jump in pitch occurs when the wedge has a distance between those shown in c and d, and another between g and h. For the distances shown in f and g either of two tones may be obtained. Further studies of the curves of disturbance led to the remarkable result that they are symmetrical about the plane through the slit and wedge.

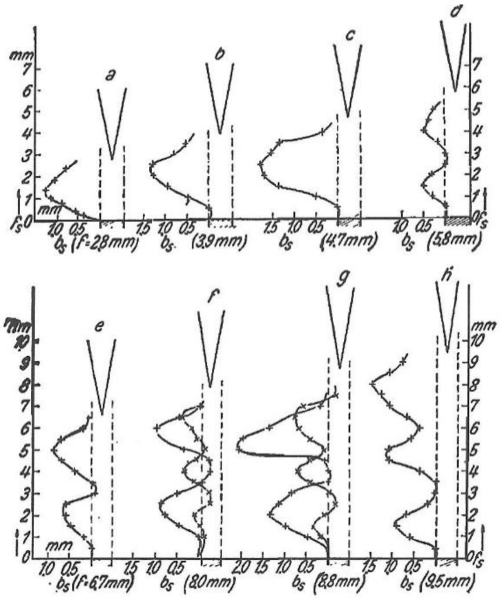


Figure 174 Curves showing disturbing wedge position at which the disturbing wedge produced an effect on the sound for a distance of slit from wedge of 6.7,8.0,8.8 & 9.5 mm. (After Klug [81])

Figure 175 was produced by Bonavia-Hunt [82] in 1947 and is the organ builders understanding of the edge producing mechanism.

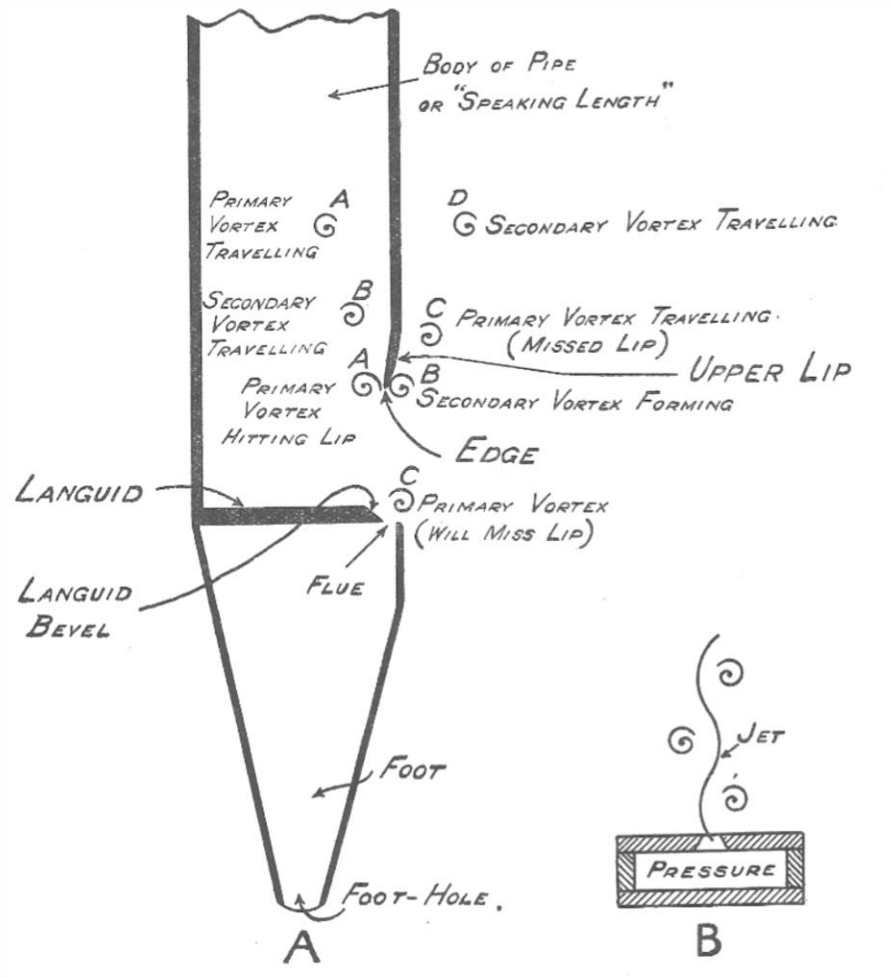


Figure 175 Organ flue pipe vortex diagram (After Bonavia-Hunt [82])

Since these early experiments, the vortex action that occurs at the organ pipe mouth has occupied many researchers. It should not be forgotten that the voicer intuitively manipulated the various parts of the pipe mouth, not only to make the pipe speak, but more importantly, make it speak with the correct timbre.

16.2.8 Scaling.

Scaling is the art of matching the power of the organ to the room acoustics. The scale of a pipe or its width is the main feature that gives the pipe its distinctive sound or timbre. A wide scale pipe reinforces the fundamental and low harmonics and has few if any high harmonics. Flute stops normally have wide scales.

The other extreme is a narrow scale pipe which has little fundamental but is rich in high harmonics. String stops normally have narrow scales and fall into this category. Scales which are neither wide nor narrow give rise to the diapason tone which is regarded as the true pipe organ tone or timbre. The diapason stop is both rich in fundamental and harmonics, vital for supporting singing and the playing of contrapuntal music.

Figure 176 shows the variation in the three main scales.

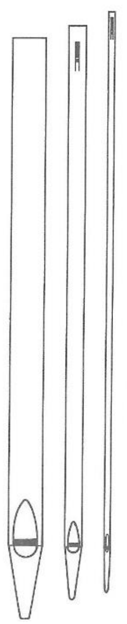


Figure 176 Pipework Scales (After Audsley [3])

It is the primary function of the Tonal Director to determine the diameter for each pipe and how this progresses throughout the set of pipes. In addition, he must determine mouth widths, cut-up and wind pressures. The general principals of scaling are described by Barnes [1], Audsley [3], Norman[83], and a more theoretical treatment is provided by Fletcher[84].

Up to the middle ages pipe organs had only two octaves of pipes. Fletcher [84] describes how these early organ pipes all had the same diameter irrespective of the length of the pipe. The diameter of the pipes was the size of a “Pidgeon’s” egg, approximately 25 to 30mm, with the longest pipe having a length to diameter ratio of 8 to 1. This uniform scale was difficult to extend downwards without the bass pipes becoming too narrow and with a stringy tone. Extending the treble pipes caused the opposite and produced a dull fluty sounding tone.

Progress was made in the thirteenth century to overcome this problem, and a new method of determining the scale of the pipes was introduced by making the diameter of the pipe proportional to its length. This new method gave a length to diameter ratio of 2 to 1 and produced a workable rank of pipes. Unfortunately, now the bass pipes were too wide, producing a dull sound, and the treble pipes too narrow giving a soft stringy sound. Work by Dom Bedoes and Topfer in 1833 evolved the principle of halving the diameter of the pipe on the 15 to 19 note. It is this principle that is universally used by pipe organ builders. Normally flute pipes would halve on the 15th pipe and strings on the 19th pipe. Diapason pipes normally halve on the 17th pipe.

The diameter of the pipe is not the only key dimension. Also important is the width of the mouth and the cut-up. Normal practice is to specify the mouth width as a fraction of the pipe circumference. In general, the wider the mouth, the more powerful and less refined the tone of the pipe. The opposite exists for narrow mouth widths. Powerful diapason pipes would have a 2/9 mouth with a normal diapason having a 1/4 mouth. String pipes normally have a 1/5 mouth. The associated cutup is normally a proportion of the pipe mouth dimension and is set using proportional dividers. A large cutup reduces the harmonic content and a typical value for a diapason rank would be 1/4. Flutes would be more typically 2/3 or 1, and in some stops the upper lip would be arched. The cut-up for a string stop would be 1/4 or 1/3. Adjustment of the cutup is very important in creating the right conditions in the mouth area to produce the edge tone drive mechanism. The degree of latitude in some pipes is very small; too much and the pipe will be scrap. Fortunately, the pipe can be melted down and remade.

16.2.9 Voicing

The pipes are made by the pipe-maker and it is the task of the voicer to make the pipes speak. The voicer works intuitively adjusting the cut-up, windway, languid and toe hole to obtain the correct timbre for the pipe and the complete stop. It is very important that the passage from one note to the next should be smooth. The general principles of voicing are described by Barnes [1], Audsley [3], and Norman [83] . A more theoretical treatment is given by Steenbrugge[10] and Rioux[85]. At the heart of voicing is the maximisation of the vortex action that occurs at the mouth of the pipe by the jet drive mechanism.

There are two basic methods of voicing. The Classical style uses open toe voicing with no nicking on the windway, producing a sound with a distinctive chiff. The Romantic style allows nicking of the windway to break up the air jet, and the sound has less of an initial transient. The voicer makes a number of nicks uniformly across the windway as part of the voicing of the pipe. Some pipes require few nicks, others may need to be heavily nicked.

The cutup together with the width of the windway and size of the toe hole, are the main parts of the pipe that the voicer intuitively manipulates to give the organ pipe the desired sound. The pipes are pre-voiced in the workshop on a voicing machine, which is a small organ that can accommodate several ranks of pipes. Once voiced, the pipes are delivered to site for final regulation in the building.

Bonavia-Hunt gives the following factors which influence the speech and tone of flue pipes:

- The scale of the pipe
- The area of the mouth
- The shape of the languid
- The adjustment of the relative positions of the upper and lower lips, and the position of the languid
- The amount and style of nicking
- The pressure and the volume of the wind supplied from the windchest
- The size of the pipe foot hole
- The material used to make the pipe and any slotting or treatment of the end of the pipe

From a practical point of view, it is important that the pipe should not only make the correct tone, but it must not be slow or fast in speaking. The adjustment of the languid and position of the upper lip are the two parts of the pipe that control this area of voicing. The idea is to make the pipe speak as slow as possible without it actually being slow. If the height of the languid is too low the pipe will be quick to speak. Alternatively, if too high the speech will be too slow. Unfortunately, most pipes have the languid made from pipe metal which tends to sag with time and results in making the pipes speak more quickly with time. Often when a pipe organ is restored or rebuilt, it is this area of the organ pipe that needs the greatest attention. In addition, the position of the upper lip is important if the pipe is to speak promptly. If the pipe is fast pushing out the upper lip will generally slow the speech.

16.2.10 Reed pipes

Reed pipes form the other group of organ pipes and are used to add richness and “fire” to the sound and assist sound projection. Figure 177 shows a typical reed pipe with boot, block, reed and resonator. The sound is produced by a vibrating brass reed held in place by a wooden wedge and tuning wire. The resonator is tuned to the vibrating reed. The shape of the resonator, Figure 178, is the main influence on the tone and timbre of the pipe [82]. Wide scales are rich in the fundamental and lower harmonics like the Trumpet stop. Thin scaled resonators, like the orchestral oboe, produce a thin tone. The parallel resonator of the clarinet produces a rich and woody tone.

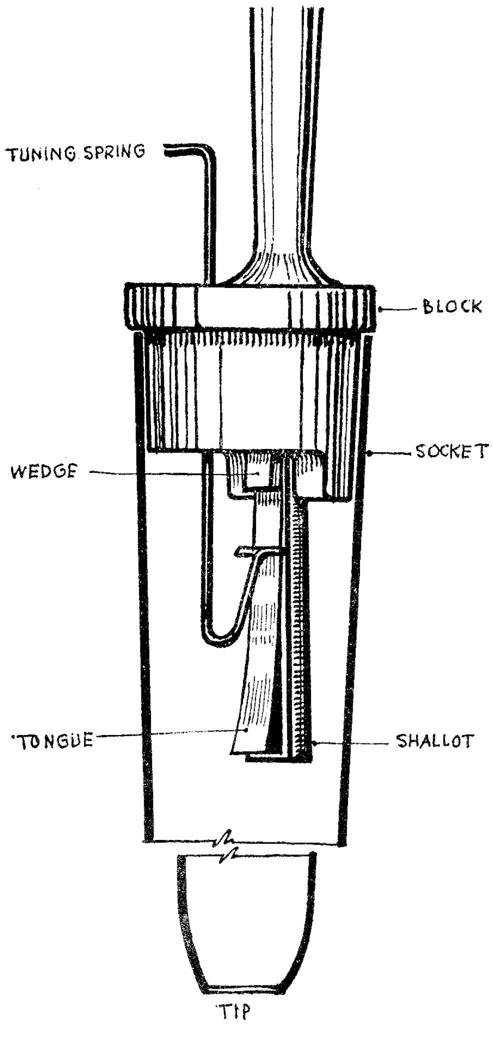


Figure 177 The various parts of a reed pipe (After Norman [83])

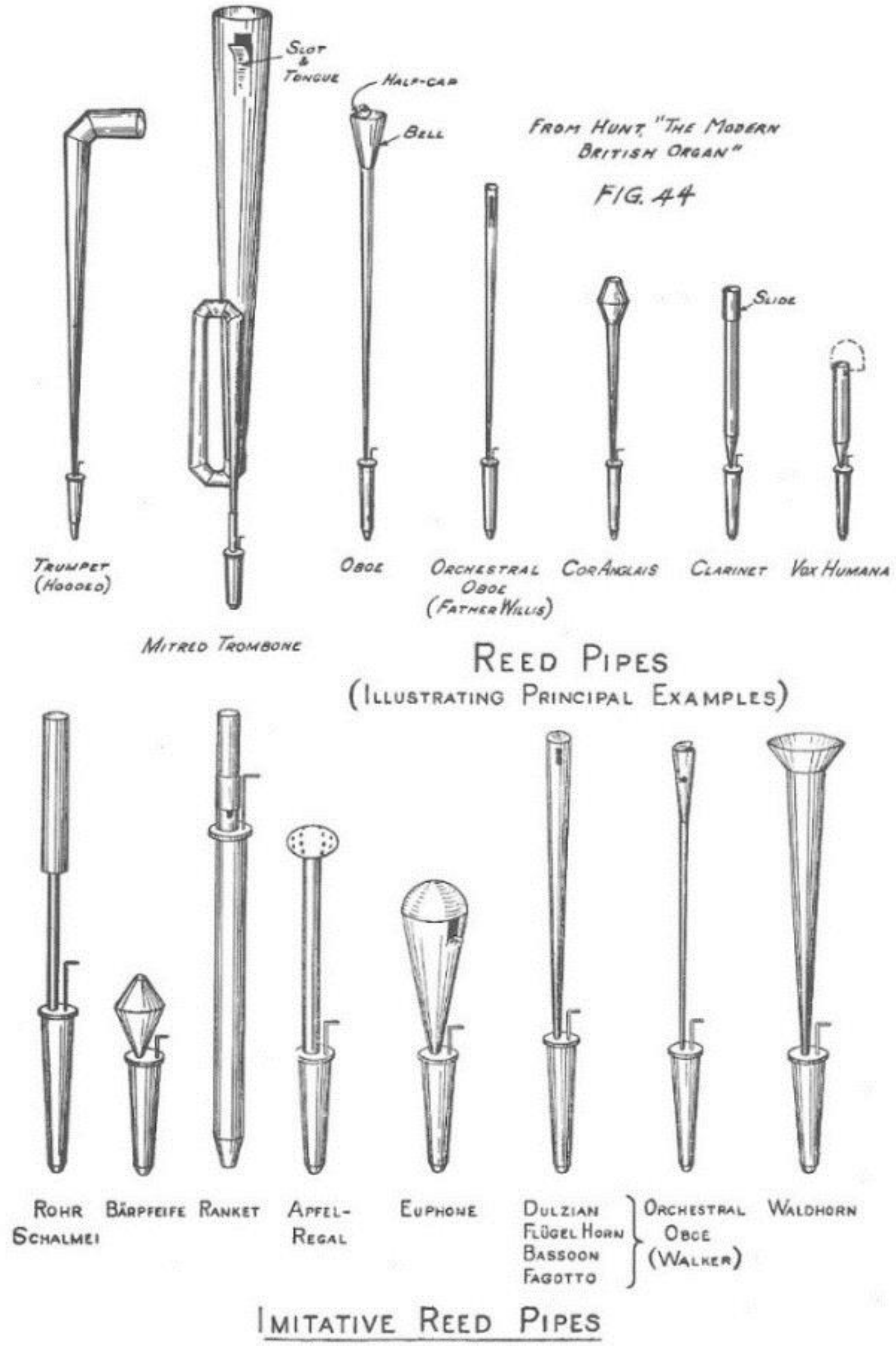
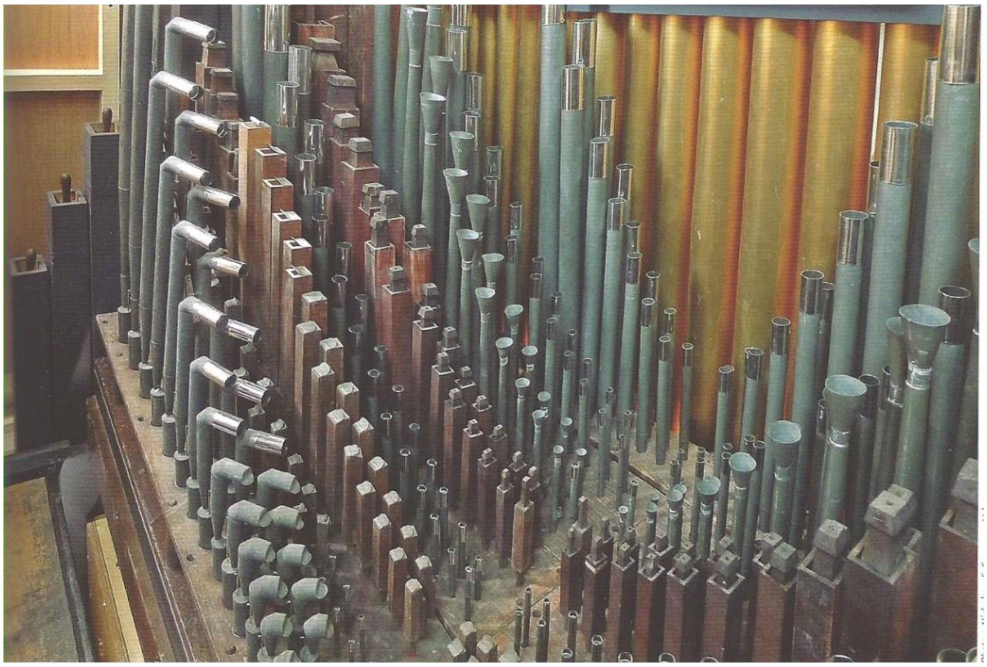


Figure 178 The many possible variations in resonator styles (After Bonavia-Hunt [82])

16.2.11 Choir organ soundboard

A small 7 stop choir organ soundboard is shown in Photograph 39. The soundboard contains open and stopped flue pipes made from wood and pipe metal. Also included is a single rank of metal clarinet reed pipes.



Photograph 39 Seven stop choir organ

16.3 Pipe organ control system

Unlike small orchestral wind instruments, where the keying system is compact and immediate, there are several possible ways of operating the pipes.

The normal keyboard compass ranges from 54 to 61 notes. Pedal keyboards normally have 30 or 32 notes. This means that for each key there is a single pipe that varies in pitch with the adjacent key. The pipework is arranged into departments or divisions associated with each keyboard. Each department contains several sets of 54 or 61 pipes and each set is called a rank or stop. The keyboards may be connected by couplers so that the whole organ may be played from a single keyboard.

16.3.1 Mechanical action.

The first pipe organs used a simple system of levers to connect the keys to the pipe valves. A further system of levers was also employed to operate the stop mechanism.

16.3.2 Pneumatic action

The use of mechanical action is suitable for small pipe organs, but for larger instruments some form of mechanical assistance is necessary. The first form of assistance was pneumatic, using air contained in small lead tubes to connect the keys and stop knobs to the pipe valves stop machines. The success of this form of action requires stable climatic conditions and can prove troublesome in damp conditions.

16.3.3 Electric action

The availability of reliable low voltage solenoids, power sources and multi-contact switches was a natural progression that made the construction of very large pipe organs possible. With the development of transistors and microprocessor software control systems it is now possible to produce a control system with no moving parts other than the action solenoids, greatly improving reliability.

17 Appendix D the tremulant.

The Tremulant is a device designed to create a vibrato effect on the sounding organ pipe. It is normal to control the depth and frequency of the amplitude modulation by mechanical or electronic means. There are four basic methods of producing a tremulant effect. The first, and most popular method shown in Figure 179, is to use a device that dumps air from the wind system in a periodic way. A second is the use of the Dom Bedos style tremulant which is mounted inside the wind duct and is designed to cause perturbations to the wind. A third and less popular method is to mount a device that has an eccentric rotating weight on the top of the reservoir. The eccentric weight imparts a periodic motion to the top of the reservoir. For pipe organs with no reservoirs, similar to pipe organs made by Austin using their universal wind-chest, a fan type of tremulant must be employed. This is achieved by placing a set of large rotating blades over the organ pipes. The slow movement of the blades gives a gentle vibrato effect.

Samuel Scheidt (1587-1654) described the stop as "a dignified stop and one of importance on the organ" Sumner [86]. In the accounts for the organ of King's College Chapel, Cambridge, made by Thomas Dallam in 1601, an item is listed "For brasse for the shaking stoppe"Audsley [87]. The popularity of the tremulant in the sixteenth century waned a little in the seventeen century. In 1666 Mertel commented that the device should be confined to sad and penitential songs and during the Sanctus Sumner [86]. At the Halberstadt Convocation in 1693 it was decreed that the tremulant must not be used with full organ "as its beating will shake up the instrument and send it out of tune". Perhaps the most notable mention of the tremulant is by Dom Bedos [88] who describes its construction and how it could be used to cover up irregularities in reed tone.

The early tremulants were designed to work with weighted double rise or wedged shaped reservoirs and wind pressures less than 50mm WG. This compares with their use in cinema organs where the wind pressures tend to be over 150mm WG supplied from single rise sprung reservoirs. Today, some organ builders still prefer to use a Dom Bedos style tremulant, particularly with low reservoir pressures.

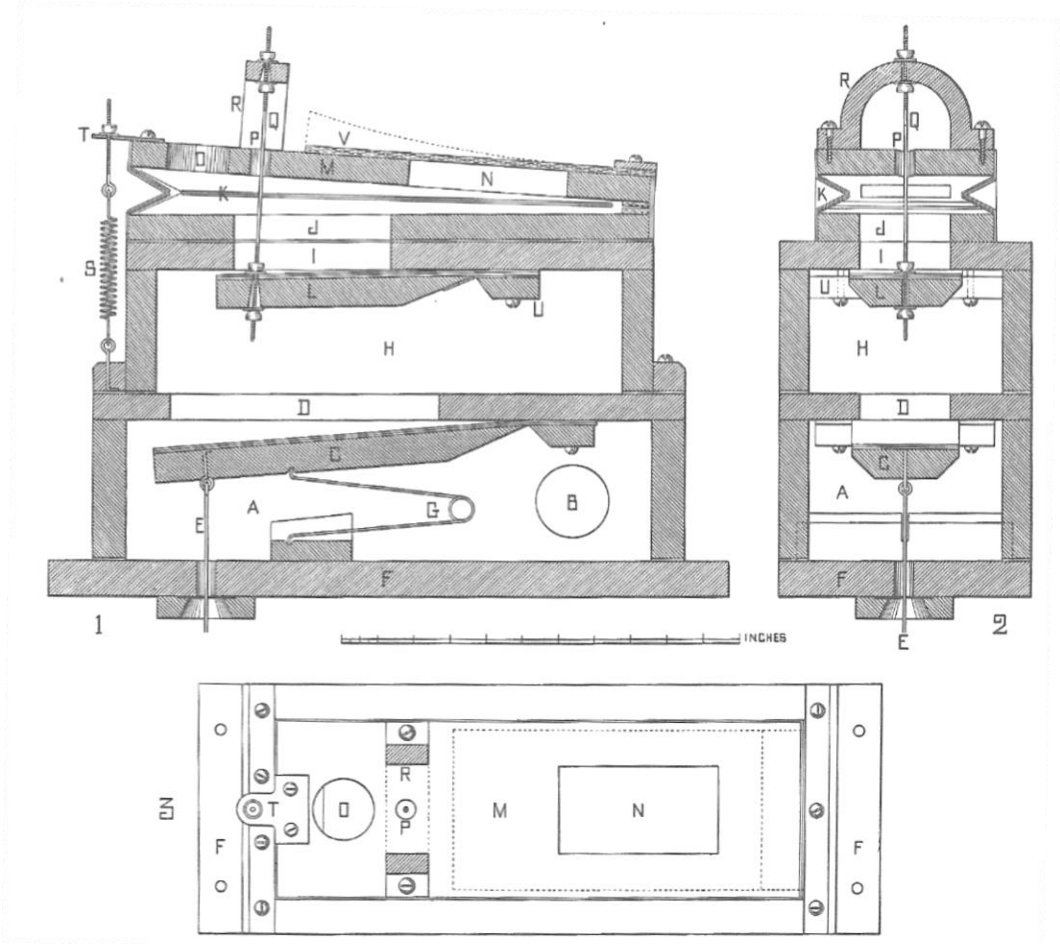


Figure 179 Traditional wind dumping tremulant (After Audsley [3])

Photograph 40 Wind dumping tremulant

17.1 Methodology

Using the apparatus and measuring equipment described in Chapter 4, weighted and sprung combinations are examined with the tremulant shown in Photograph 40. The wind pressure was set at 62.5mm WG and the frequency of the tremulant set at 2Hz. The resonant frequency of the weighted reservoir is 13Hz and the sprung reservoir 19Hz.

17.2 Results

The frequency spectra for weighted (blue) and sprung (red) are shown in Figure 180

Frequency Hz.

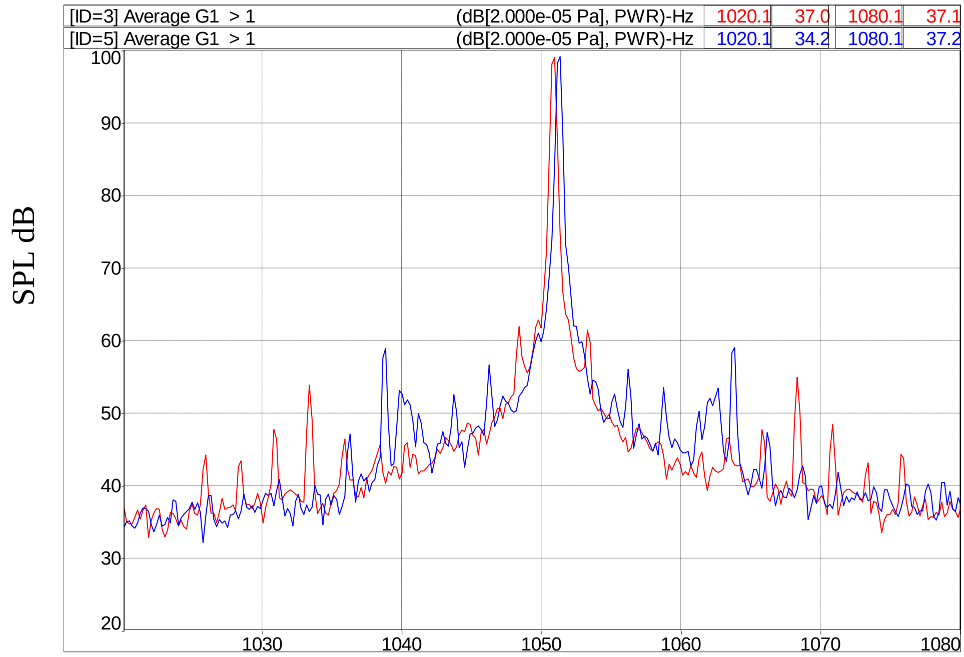


Figure 180 Comparison between weighted (blue) and sprung (red) tremulant

17.3 Conclusions

It is significant that there is a difference between weighted and sprung reservoir systems.

Each frequency plot shows the strong multiple side-bands at 2Hz intervals on each side of the fundamental.

18 Appendix E The Austin Universal Wind Chest.

The Austin Universal wind chest does not exhibit the resonance found with weighted or sprung reservoirs Barnes [1]. The Universal Wind Chest was devised in the late 1880s by the Austin brothers during their time at Farrand & Votey in Detroit. They had become concerned at the difficulties of servicing the actions of the multi-pallet chests used by Farrand &Votey. Access to the action was achieved by removing the bottom board or, after pipes had been removed, by removing the top board. The Austin brothers were principally engineers, who devised a system that allowed the serviceman to enter the chest to service any action faults. The new style of chest was a combination of pneumatic with interlinking trackers. A drawing of the Austin Universal wind-chest is shown in Figure 181.

The system used three types of chest; the walk-in, the crawl-in and the panel chest. The crawl-in and panel chests were devised for installations in spaces that did not allow a walk-in chest to be fitted. Until 1936, most of installations had a single walk in chest.

The method of winding the walk-in chests was relatively simple. The trunk from the blower usually entered through the chamber floor. In the side of the chest was a 2000 x 1200mm single rise regulator C, sprung to the open position with springs B. When the wind entered, the chest pressure rose and pushed against the regulator fold, which moved back, taking the control valve cord with it, thereby closing a trap over the trunk entry point A. This was a very crude, but very effective method of controlling the air supply from the blower. The standard wind pressure was 100mm WG with higher pressures in larger rooms.

Originally, and for the most part, the whole organ was planted on a single air chest, the manual departments side by side and the pedal divided at each end. Entry to the chest interior was through an air-lock, an outer door leading to the vestibule and a second door giving access to the chest interior. Both doors had small holes bored at shoulder height with a non-return valve behind so that the pressure could be equalized in the vestibule by poking one's finger in to open the valve.

The enormous volume of air in the chest and the relatively small flexible volume, the regulator fold, made the wind very steady, so steady that a conventional tremulant was not effective. This necessitated the use of a fan tremulant, driven first by an air motor, then by a dc electric motor. The tremulant was placed over the pipes and consisted of two vanes made from card, about 1200mm long, fitted to a spindle turned by the motor.

One of Austin's largest installations was at City Hall, Portland ME. Here the chest ran the entire width of the hall and had a volume of approximately 120m^3.

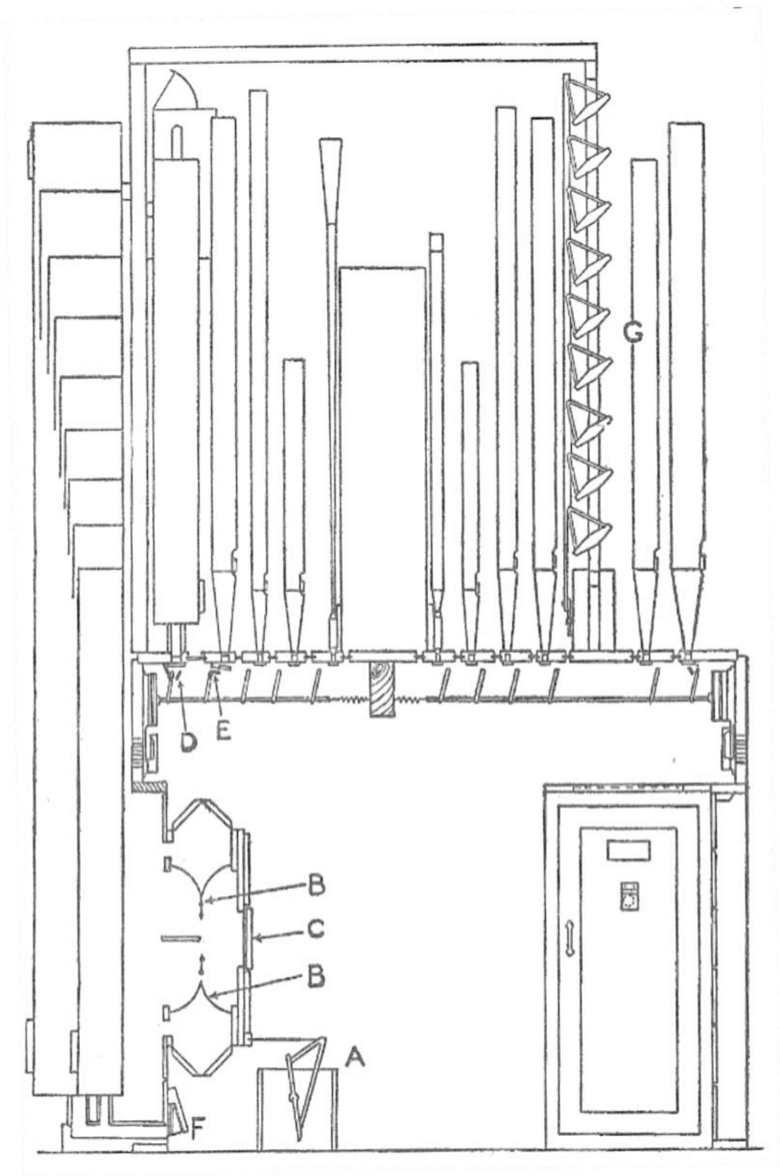


Figure 181 The Austin Universal wind-chest (After Bonavia-Hunt [82])

19 Appendix F Glossary of organ terms

Action. The mechanism that connects the keyboards to the pipes.

Blower. The pipe organ building term for centrifugal fan.

Choir Organ. A secondary division to the great organ.

Compass. The number of keys on each keyboard.

Console. The part of the pipe organ where the player sits.

Coupler A device that allows keyboards to be joined together.

Cut-up. The height of the mouth as a fraction of the pipe width.

Division. The name given to a section of the pipe organ, ie Great Organ.

Duct. Organ building term used for a connecting pipe or trunk.

Flue. The windway of an organ pipe.

Flute. A metal or wood organ pipe with a more fundamental tone.

Great Organ. The principal division of an English pipe organ.

Keyboard. The part that the player uses to play the organ.

Manual. The name given to a keyboard played with the hands.

Pallet. The valve below the organ pipe that controls the flow of air into the organ pipe.

Pedalboard. The name given to a keyboard played with the feet.

Pedal Organ. The division connected to the pedalboard and played with the feet.

Principal. The main pipe organ tone rich in fundamental and harmonics.

Rank. The term given to a complete set of organ pipes, ie 8ft Trumpet.

Reed. The vibrating part of a reed organ pipe.

Reservoir. A box with flexible sides to hold air under pressure.

Scale. The variation in size of the diameter of an organ pipe in a predetermined ratio.

Stop. A device for selecting the individual ranks of organ pipes.

Swell. A division that is enclosed in a box with louvres that can be open and closed to give expression to the sounding pipework.

Trunk. Organ building term used for a connecting pipe or duct.

Windchest. The box that stores the wind on which the organ pipes sit.

Wind pressure. The pressure of air used to sound the pipes usually measured by the height of a column of water.

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