Modelling flow-induced vibrations of gates in hydraulic structures

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5 Physical experiment

5.1 Preface

This chapter and the next are connected by a common aim: to investigate the dynamics of a new gate lay-out. The physical model discussed in this chapter was prepared and executed at Deltares in Delft, The Netherlands. A more detailed description of the experiment is reported in Erdbrink (2012a). Chapter 6 continues the study of the same gate shape by numerical physics-based simulation of its FIV response. This will give an idea of the feasibility of applying fundamental numerical models as assessment tools for gate designs, as a complementary tool next to physical modelling. In the present chapter, the set-up and the results of the physical scale model experiment are given. Conclusions and a more comprehensive discussion are included in Chapter 6.

5.2 Introduction

As laid out in Chapter 2, the dynamic response of a hydraulic gate due to its interaction with the flow strongly depends on details of the gate bottom geometry. Numerous experimental studies of flow-induced vibrations (FIV) of gates have previously looked into the characteristics of gate shapes (Hardwick 1974, Vrijer 1979, Kolkman 1984). The gained insight in excitation mechanisms has resulted in widespread rules of thumb for unfavourable designs that should be avoided as well as favourable design features (e.g. Thang 1990, Naudascher and Rockwell, 1994). However, fundamental knowledge and practical experience have not culminated in one ideal universal shape – partly because the surrounding structure is an important factor.

Experimental and numerical models are incapable of capturing all degrees of freedom (d.o.f.) experienced by real-life gates (mass-vibration mode in cross-flow and in-flow direction, bending, torsion). Streamwise (horizontal) vibrations are usually studied separately (e.g. Jongeling 1988) and sometimes in combination with the cross-flow mode (Billeter and Staubli, 2000). In this study we consider the most frequently encountered and investigated mode for a vertical-lift gate: one d.o.f. in the cross-flow direction.

The emergence and severity of flow-related dynamic forces on the gate are related to flow instabilities and body motion effects (Section 2.4). Fluctuations of the separated flow’s shear layer may incite a mechanism called Impinging Leading Edge Vibrations (ILEV) for gates with a sharp upstream edge. If the gate bottom has an extending lip in streamwise direction, the shear layer separated from the upstream edge may reattach to the gate bottom in an unstable way, giving dynamic excitation. In a different mechanism, periodic forces are the result of initially small gate movements. This self-exciting process is called Movement-Induced Excitation (MIE).

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3 This chapter is based on and uses text and content from “Reducing cross-flow vibrations of underflow gates: experiments and numerical studies” by C.D. Erdbrink, V.V. Krzhizhanovskaya, P.M.A. Sloot, currently under review at the Journal of Fluids and Structures.
Previous investigations have proved that most severe vibrations of underflow gates in submerged flow occur at small gate openings and are predominantly caused by ILEV and MIE mechanisms (Hardwick, 1974; Thang and Naudascher, 1986a and 1986b). The current investigation therefore focuses on small gate openings and does not look at the distinctly different mechanism of noise excitation. Other notable experimental studies are Kapur and Reynolds (1967), Naudascher and Rockwell (1980), Thang (1990), Kanne et al. (1991), Ishii (1992) and Gomes et al. (2001). Only the last study also includes numerical modelling.

Assuming that adding structural damping or avoiding critical gate openings are unfeasible options, the shape of the gate bottom is the decisive factor determining the tendency to vibrate. If the flow passes the gate while remaining attached, or if there is a fixed separation point and a stable reattachment, or if the shear layer is kept away from the bottom in all circumstances, then the ILEV mechanism may be avoided. A thin, sharp-edged geometry with separation from the trailing edge is favourable (e.g. the rightmost profile in Figure 2 in Section 2.2.2), because potential shear layer instabilities occur downstream from the gate and a small bottom area inhibits the occurrence of large lift forces on the gate, thus minimizing the risk of MIE vibrations. But such a design is often not possible due to other design constraints.

At the start of this study a number of new ideas for attenuation measures were identified:

(i). Counter-balance the vibrating gate mass by adding an extra elastically mounted weight to the gate that is set into motion to compensate the gate movement.

(ii). Tondl (1998) has made several analytical studies into the quenching of self-excited vibrations by varying the stiffness of the support (this is called parametric excitation).

(iii). Influence the hydrodynamic gate pressures actively by injecting or sucking water through a pump-regulated system of tubes flowing out through the gate bottom. This should work to disturb the excitations.

(iv). Make holes (or shafts or slots) in the gate bottom profile so that an intentional leakage flow can develop, again with the goal of influencing the bottom pressures beneficially.

(v). Adjust the bottom geometry in such a way that the profile acts as a (semi-)aerofoil. This would enable active ‘steering’ of the gate by using (rotations of) the profile to increase or reduce the steady lift force. This acts as an aid for gate lifting and lowering under a head difference and may help to avoid critical gate openings where vibrations occur.

As far as known from literature studies, none of these measures have been investigated before or tested for their achievable for use in hydraulic structures. The investigation at hand chooses measure (iv) as the central new idea. An unfavourable thick flat-bottom rectangular gate is taken as a reference gate and a new design with leakage flow through two openings in the bottom section is investigated as a potential way of improving the vibration properties, see Figure 5.1.

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4 Ideas (i) and (iii)-(v) originate from discussions at Deltares with Tom Jongeling, who helped enormously in the design stage of the experiment.
Figure 5.1. Streamwise cross-section of gate configuration showing ventilated gate design (detail of bottom element on the right). Dimensions in millimetres; not drawn to scale.

5.3 Experimental set-up

The experiment of a gate section was performed in a 1 m wide and about 90 m long laboratory flume. A straight, vertically placed underflow gate is suspended in a steel frame that is fixed to the flume. Figure 5.2 contains drawings of the placement of the gate and the frame in the flume. The dimensions of the gate are 1100x600x50 mm (height x width x thickness); it is a stiff plate and thus acts as a linear mass-spring oscillator body with one degree of freedom in the cross-flow vertical direction. To prevent measurement equipment and frame parts from influencing the flow, the flume was locally narrowed to 0.5 m by constructing side walls out of waterproof film-coated plywood and a sloped ramp of the same material around the gate. In the section closest to the gate, the walls were made out of transparent Perspex plastic to allow visual inspection. The flow directly upstream from the gate was attached to the walls and has low turbulence intensity.
Figure 5.2. Set-up of gate experiment in the flume.
Figure 5.3 shows the gate’s front view and suspension. As mentioned in the Introduction, two gate types were tested. The flat rectangular-shaped bottom (with smooth surface, sharp edges and without extending lip) will be called ‘original gate’. The adapted gate differs from the original gate in that it has five horizontal slots in the upstream face of the bottom section, as shown in Figure 5.3, and five identical slots in the bottom face of the bottom section, as shown in Figure 5.4. This gate will be called ‘ventilated gate’. The slots on the upstream side act as inflow openings for leakage flow and the slots on the downward facing side act as outflow openings. The dimensions of the slots were chosen such that the effect of the leakage would be distinctly perceptible, but without compromising the rigidity of the gate.

![Figure 5.3. Front view from upstream side, showing inflow ventilation slots in the bottom section of the gate and the three supporting springs. Measurement frame is left out for clarity. Dimensions are in millimetres. Only the bottom element is drawn to scale.](image)

The gate is supported in vertical direction by three springs, one spring in the center with adjustable stiffness and two side springs of lower stiffness. Prior to each measurement, the gate was set to the desired height and the main central spring was set to the desired stiffness. Then the tension in the two side springs was adjusted (symmetrically) by changing the length of the chains connecting the side springs with the frame, see Figure 5.3. This was done in such a way that the two low stiffness side springs carried most of the static loads in vertical direction. The dynamic loads were mostly carried by the stiff central spring. Its adjustable stiffness enabled a controlled variation of the natural frequency. For the two weak side springs, linear coil springs (Alcomex TR-1540) were used; for the stiff main spring a double leaf spring was custom-built with high yield strength steel and a high elastic limit (Armco 17-7PH, hardening condition TH1050).

The two bending blades of the main spring have dimensions 600 mm x 30 mm x 4 mm (l x b x t). The bending length L (< 600 mm) can be adjusted by movable blocks with clamps, thus varying structural stiffness, see Figure 5.3. Linearity of this spring was confirmed by static
loading tests for different bending lengths. The main spring is installed in parallel with two side springs, each with constant $k_{\text{side}} = 0.57 \text{ N/mm}$. The relation between bending length and total stiffness of the three springs is derived from constitutive relations plus Hooke’s law:

$$k_{\text{total}} = k_{\text{main}} + 2k_{\text{side}} = \frac{8Ebt^3}{L^3} + 1.14,$$

(5.1)

with $k_{\text{total}}$ the combined spring stiffness in N/mm, $E$ the modulus of elasticity in N/mm$^2$, $b$ the width and $t$ the thickness of the leaf spring blades in mm and $L$ the length between the clamps in mm. This formula was calibrated with free vibration tests in air to increase accuracy between chosen $L$ and achieved $k_{\text{total}}$ and dry natural frequency $f_0$.

The main gate body consists of a rigid steel grid filled with water-resistant foam and covered with thin plastic plates on both sides. The bottom element is carved out of PVC material and is screwed onto the main body, thus forming one stiff mass with it. In order to reduce weight, the bottom element is hollow; it contains five chambers (see Figure 5.4). The total mass of the original gate was 17.2 kg. The bottom element of the ventilated gate is lighter due to removal of material, but the openings allow more water into the cavities of the bottom element so that the total mass of the modified gate was only slightly higher: 17.3 kg. These values exclude added mass due to water displacement during oscillation.

A plan view of the gate is sketched in Figure 5.4. The water flows between the perspex walls, in the figure from bottom to top. The space between the perspex walls and the flume walls is filled with still water at the downstream water level. The side seals consist of thin vertical strips that are carefully installed such that side leakage is minimized and at the same time no contact is made with the gate. Observations during the experiment indicated that the inevitable sideways leakage was only significant at relatively high hydraulic heads, although the leakage appeared to be only a small fraction of the total gate discharge. It was also
observed that the distance between gate and side walls on the downstream side had little impact on the underflow discharge. Figure 5.4 contains a gate cross-section through the bottom section, such that the outflow openings of the ventilated gate become visible. These five openings are identical to the five openings on the upstream face of the gate. All slots were cut perpendicularly to the faces, see Figure 5.1.

Five horizontal supports, three hinged steel rods in longitudinal direction and two in cross direction, enable vertical movement of the gate while fixing the gate position horizontally. Six force meters were installed: vertically one for each spring and horizontally one for each longitudinal support. Only the main vertical support force is used in the analysis of the gate response. The sample frequency was 200 Hz. The length of the analysed data files was 90 s on average and had a minimum of 60 s, yielding a frequency resolution of at least 0.0017 Hz. Recording started after reaching equilibrium water levels and horizontal support forces. Figure 5.5 shows a sample from an analysed measurement signal of the mean leaf spring. The signal analysis was done in MATLAB and consisted of a fast Fourier transforms analysis (FFT). First, the time average of the remaining steady component of the main spring force was subtracted from the signal. Then a sliding window was used to calculate the moving average and the amplitude envelopes, which were smoothed with a simple triangular smoothing function. The Hilbert transform was used to find the envelopes. For each measurement signal, the representative vibration amplitude was determined as the mean difference between the envelopes and the moving average.

Furthermore, the discharge and the water levels on the upstream (h1) and the downstream side (h2) of the gate were measured using resistance-type water level meters. The locations of the water level meters are shown in Figure 5.2. See Erdbrink (2012a) for more details on the experimental set-up.

![Figure 5.5. An example excerpt from a measured signal of the main spring, with computed moving average and envelopes added.](image-url)
5.4 Definitions

The motion equation in vertical $z$-direction for partly submerged bodies includes hydrodynamic coefficients (cf. Sections 2.3-2.4):

$$ (m + m_w) \ddot{z} + (c + c_w) \dot{z} + (k + k_w)z = F(t). \quad (5.2) $$

Here, $m$ is mass, $c$ is damping, $k$ is stiffness, $F$ is the excitation force and the subscript $w$ indicates the added coefficients. The excitation is a time-dependent hydraulic force which can be written as

$$ F = \frac{1}{2} C_F \rho U^2 W D = C_F F_0, \quad (5.3) $$

with $W$ the cross-flow width of the gate section on which $F$ works, $F_0$ a stationary reference fluid force and $C_F$ is a periodically varying force coefficient. In this chapter not the displacement $z$ but the response force $F_z$ is measured, the amplitude of which is denoted $F_z$.

Furthermore, for the pressure on the gate bottom boundary $p_{\text{bound}}$ we use the pressure coefficient $C_F$ defined logically as $p_{\text{bound}}$ divided by $\rho g \Delta h$. The two-dimensional discharge formula for an underflow gate section in submerged flow is

$$ q = C_D a U = C_D a \sqrt{2 g \Delta h}, \quad (5.4) $$

with $q$ the discharge per unit width in m$^3$/s/m or m$^2$/s, $C_D$ the dimensionless discharge coefficient for submerged flow, $a$ the lifting height or gate opening, $U$ the flow velocity in the vena contracta, estimated with Bernoulli’s formula, in m/s and $\Delta h = h_1 - h_2$. The reduced velocity $V_r$ is used as dimensionless descriptive quantity for the flow-induced vibrations. It is defined here as

$$ V_r = \frac{\sqrt{2 g \Delta h}}{f_z D}, \quad (5.5) $$

where $f_z$ is the dominant response frequency, the numerator represents the characteristic flow velocity and the gate thickness in flow direction $D$ (see Figure 5.1) is taken as characteristic length scale. To find the added mass $m_w$, it is here estimated experimentally from free vibration tests in air and still water. It follows from the ratio of the wet and dry natural undamped frequencies. For $f_{0,\text{water}}$ we have

$$ f_{0,\text{water}} = \frac{1}{2\pi} \sqrt{\frac{k + k_w}{m + m_w}}, \quad (5.6) $$

and the ratio is thus
\[
\frac{f_{0,\text{water}}}{f_0} = \sqrt{\frac{1 + k_w/k}{1 + m_w/m}}. \tag{5.7}
\]

because the added rigidity (hydraulic stiffness) \( k_w \) due to buoyancy on the submerged part of the gate is found via Hooke’s law: \( k_w = \rho g W D \). The error made by neglecting damping here is less than 2%. Numerically, \( m_w \) near a wall in still water was computed in a potential flow model with a finite difference method by Kolkman (1988), and studied many times since then in the context of gates (e.g. Anami et al., 2012). A very universal approach is the observation in Kolkman (1984) that the total kinetic energy of the fluid can be expressed as

\[
E_{\text{kin}} = \frac{1}{2} m_w \dot{z}^2. \tag{5.8}
\]

Summing the velocity magnitude over all computational nodes should yield a value for \( m_w \), assuming that the object velocity is known.

5.5 Measurement conditions and variation of parameters

The measurement procedure was to vary \( V_r \) and \( a/D \), which was done mostly by changing suspension stiffness and gate height and occasionally by changing the pump discharge and height of the outflow weir of the flume. The present study deals with the critical gate opening range for cross-flow vibrations: 98% of the measurements lie in the interval 0.48 \( \leq a/D \leq 1.50 \). During the experiment the stiffness was varied between and 19.3 N/mm and 967 N/mm, corresponding to a range in achieved undamped natural frequency in air of 5.33 Hz \( < f_0 < 37.7 \) Hz for the original gate with closed bottom section. The gate submergence \( C_s = (h_2 - a)/D \), with \( h_2 \) the water depth measured downstream from the gate was in the range 4.2 \( < C_s \leq 6.2 \), for 98% of the data. This means flow conditions were close to fully submerged with minor free surface fluctuations. Discharge coefficient \( C_0 \) is estimated from the measured pump discharge to be on average 0.80 with standard deviation 0.11 for the original gate and on average 0.83 with standard deviation 0.10 for the ventilated gate. The achieved \( V_r \) ranges were 1.2 \( < V_r \) < 11.6 for the original gate and 1.8 \( < V_r \) < 9.5 for the modified gate. The Reynolds number defined as \( \text{Re} = UD/\nu \) again with \( U = (2g\Delta h)^{0.5} \), was in the range 3.2 \( \cdot 10^4 \) \( < \text{Re} < 1.3 \cdot 10^5 \). The mass ratio, defined as \( m_r = (m + m_w)/(\rho D^2 W) \), is computed for the original gate as 12.3 \( \leq m_r \leq 12.7 \), and for the ventilated gate as 12.3 \( \leq m_r \leq 12.8 \).

Damping was monitored throughout the experiment in 32 free vibration tests in still water, where the gate was set in motion by a manual tap on the top. For each test, the logarithmic decrement of damping \( \delta \) was determined over the first ten periods. At small damping levels, we have \( \delta \approx 2\pi\zeta \). This formula is used to compute values of the damping ratio \( \zeta \). For the original gate, \( \zeta \) was on average 0.013 and had a standard deviation of 0.0065. For the ventilated gate, the average value of \( \zeta \) was 0.020 with a standard deviation of 0.0061. The damping values contain no trends related to water depth or stiffness and did not change over time during the experiment. However, there is some inherent variability caused by the manual excitation force not being the same each time. Also, it makes a difference which part
of the decaying free vibration is used for determining $\delta$. The ventilated gate showed a small number of deviant values of higher damping at small openings $a/D < 0.7$, these are not included in the given ranges. The highest recorded damping ratio in this particular situation was $\zeta = 0.093$. A possible explanation is that flow resistance inside the chambers between the inflow and outflow slots is more pronounced at small gate openings due to higher relative velocities. Furthermore, even though damping was not higher at larger water depths in still water, the observed leakage through the side seals at high hydraulic heads (that existed for a small part of the measurements) may have had a significant effect on damping due to skinplate friction. This was not quantifiable, since the free vibration tests were only feasible in stagnant water.

The dimensionless Scruton number combines the mass and damping ratios and is, for low damping, defined by $Sc = 2m_r \delta = 4\pi m_r \zeta$. It was found that for the original gate $1.0 \leq Sc \leq 2.3$ and for the ventilated gate $0.9 \leq Sc \leq 3.7$. The variation in the Scruton number is almost completely due to the discussed variation in damping. The outliers for the ventilated gate at small openings are not included in the Sc ranges. Table 5.1 gives additional information on measurement conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Min</th>
<th>Max</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>pump discharge $Q_{pump}$</td>
<td>13.0</td>
<td>50.0</td>
<td>l/s</td>
</tr>
<tr>
<td>gate opening $a$</td>
<td>22.5</td>
<td>100</td>
<td>mm</td>
</tr>
<tr>
<td>upstream water depth $h_1$</td>
<td>0.297</td>
<td>0.656</td>
<td>m</td>
</tr>
<tr>
<td>downstream water depth $h_2$</td>
<td>0.120</td>
<td>0.352</td>
<td>m</td>
</tr>
<tr>
<td>head difference $\Delta h$</td>
<td>0.063</td>
<td>0.355</td>
<td>m</td>
</tr>
</tbody>
</table>

Parameter ranges given in Table 1 are based on the combined data set for both gates, consisting of 145 measurements for the original gate plus 85 measurements for the modified gate. The achieved underflow discharge was somewhat lower than the pump discharge as a result of the sideways leakage through the seals. An observed variation in the pump discharge of +/- 0.5 l/s was of little influence since the frequency of this variation was very small compared to signal recording length.

### 5.6 Results of physical experiment

The focus of the experimental data analysis is on determining how dominant force amplitudes in cross-flow direction change with $V_r$ for both gate types. Absolute maximum force amplitudes depend on structural damping of a particular set-up and are of less interest; therefore response amplitudes are presented for different settings relative to the stationary hydrodynamic force $F_0$.

Figure 5.6 shows the dimensionless dynamic force response of the closed gate and the ventilated gate. Judging from the plot, the response may be divided into three different regions: $2 < V_r < 3$ (relatively high stiffness), $3 < V_r < 8.5$ (medium stiffness) and $V_r > 8.5$ (relatively low stiffness). Response values of $\vec{F}_z > 5N$ represent significant, regular oscillations with response frequencies in the range $4.6 \text{ Hz} < f_{\text{resp},z} < 20.2 \text{ Hz}$. Vertical displacement amplitudes estimated from the force amplitude as $\tilde{z} = \vec{F}_z / k$ were overall less
than 0.10. The strongest recorded force amplitude in the high stiffness region was found at $V_r = 2.54$. The maximum response in the relatively low stiffness vibration region occurred at $V_r = 10.16$. The excitation mechanisms associated with these two regions are discussed in section 4.1.

The results show that the vibrations found at low $V_r$ occur quite suddenly. There are steep increases in force amplitude around $V_r = 2$ and $V_r = 3.0–3.5$; most significantly for gate openings less than or equal to $D$. Tests at gate openings smaller than 0.5D were hindered by the risk of the gate hitting the flume bottom. Although the gate opening was not varied over a

Figure 5.6. Overview of vibration characteristics of the original rectangular gate (left) and the new ventilated gate (right): reduced velocity $V_r$ versus dimensionless vibration amplitude $F_z/F_0$ of the main vertical force meter; $a$ is the gate opening, gate thickness $D$ is 50 mm.
large range, the data seems to show that the force response at $V_r \approx 2.5$ occurs at a smaller gate opening than for the response maximum at $V_r \approx 10$. Zooming in on the densely sampled area of relatively high frequency response at $2 < V_r < 3$ gives further insights (Figure 5.7). The measurements were done by making series of about ten data points of different stiffness settings, keeping gate opening and discharge constant.

![Figure 5.7. The most significant vibrations at $1.5 < V_r < 4$. Measurements series represent constant gate opening $a/D$ and head difference $\Delta h$ for varying suspension stiffness. Gate submergence relative to downstream level is denoted $C_s$. The solid triangles mark the gate with slots, all other symbols denote the original gate. Responses with $F_z/F_0 < 0.25$ not belonging to these series are left out for clarity.](image)

These results reconfirm that in this reduced velocity region significant cross-flow vibrations occur at gate openings in the range $0.5D \leq a \leq D$. For $1.5 < V_r < 4$, with the strongest force amplitudes around $a/D = 0.5$. From Figure 5.7 it appears that for head differences $\Delta h \approx 1.5D - 1.75D$, vibration maxima decrease with increasing gate opening. The series with high head difference $\Delta h = 283$ mm $\approx 5.7D$ differs from the rest by its higher hydraulic head and higher stiffness settings. Because the still water damping tests proved that damping is not significantly higher for high spring stiffness settings, the most likely explanation of the smaller response of this series is an increased damping associated with the higher hydraulic head. This is possibly related to the observed leakage through the side seals at high hydraulic head. This finding does not influence the comparisons between the two gate types. However, the reported higher damping for the ventilated gate at $a/D < 0.7$, see Section 2.3, suggests that the responses of the ventilated gate series at $a/D = 0.52$ are underestimated.

Despite the smaller data set size of the new gate design, the overall effect of applying ventilation holes seems clear. For very similar conditions at $1.5 < V_r < 3.5$, the force amplitude of the vibration is a factor 3 less for the rectangular gate with added holes. For $3.5 < V_r < 9.5$, the holed gate shows a very low response for similar conditions (openings and head difference) as the original gate. From the available data points at $V_r > 8.5$, the picture of the effect of adding holes is incomplete. More data is needed for $V_r \geq 10$. The available data suggests that the ventilation succeeds in having a mitigating effect on the vibrations in that range as well.
The ventilated gate shows an overall shift towards lower achieved $V_r$-values compared to the original gate. The maximum force response of the new gate lies at $V_r = 2.19$. Also, this maximum is found at a higher gate opening than the maximum of the closed gate ($0.74D$ against $0.48D$). This is the net effect of the altered gate design: a slightly different mass and additional discharge caused by the jet of relatively high velocity through the gate.

### 5.7 Comparison with other experimental results

The measurement data of the original rectangular gate resembles results from previous experimental research. It is important to realise that most other studies measure response in displacement instead of force. In the high stiffness at $V_r < 5$, which has been studied most frequently, the reduced velocity at which highest amplitude is found lies close to $V_r = 2.75$ found by Hardwick (1974) and $V_{r,nat} = 2.5$ found by Thang and Naudascher (1986a,b). The reduced velocity based on the natural frequency $V_{r,nat}$ is somewhat lower than the $V_r$ used in this study. The gate openings of maximum amplitude were $a/D = 0.67$ and $a/D = 0.65$ for these two studies, slightly higher than what is found here. Billeter and Staubli (2000) report maximum force coefficients in $z$-direction at $V_r = 2.75$ and $V_r = 3.0$.

The experiments of Vrijer (1979) covered vibrations in the range $10 < V_r < 80$. Their model dimensions were smaller (gate width of 10 mm). A comparison based on the small overlap of less than ten data points suggests that the lower bound of vibrations in the low stiffness region in the present study is at a slightly lower $V_r$ value ($V_r = 9.0$-$9.5$) than in Vrijer (1979) where amplitudes start to increase for $V_r > 10$.

### 5.8 Summary

This experimental study investigates a new way to reduce cross-flow vibrations of hydraulic gates with underflow. A rectangular gate section placed in a flume was given freedom to vibrate in the vertical direction. Horizontal slots in the gate bottom enabled leakage flow through the gate to enter the area directly under the gate which is known to play a key role in most excitation mechanisms. For submerged discharge conditions with small gate openings the vertical dynamic support force was measured in the reduced velocity range $1.5 < V_r < 10.5$ for a gate with and without ventilation slots. The leakage flow significantly reduced vibrations. This attenuation was most profound in the high stiffness region at $2 < V_r < 3.5$.

### 5.9 Photographs from the experiment

Figures 5.8-5.12 show photographs taken during the physical experiment.
Figure 5.8. Side view of measurement frame in flume. Flow from left to right.

Figure 5.9. Detail of the measurement installation: gate body, frame and main leaf spring.
Figure 5.10. Original closed flat-bottom gate profile (top) and modified gate profile with ventilation holes (bottom).
Figure 5.11. Close-up of gate bottom section with original bottom profile installed. Flow from left to right.
Figure 5.12. Adjusting the stiffness of the leaf spring.