

Hydraulic investigation of a Y-bifurcator

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Abstract

In this paper the model test of a Y-bifurcator of a power plant will be introduced. The model test consists of a 42° bend, a straight pipe from the bend to the Y-branch and two branching pipes with a branching angle of 40°. The secondary flow caused by the bend will be made visible by a Particle Image Velocimetry (PIV) and quantified with the velocity correction factor α . The downscaling of the hydraulic model test is based on the Reynolds law. This causes too high discharge rates in the model test and an extrapolation with a linear polynomial, based on the least square method, is used to get the discharge rate in respect to the head loss. The head losses and the corresponding local loss coefficient ζ_{local} will be presented and compared by either a linear or polynomial extrapolation.

INTRODUCTION

Considerable efforts are undertaken to reduce hydraulic losses in pipe systems of a hydro-power plant. Beside the friction loss of a straight pipe the local losses due to built-in components like a junction or confuser are also of interest. The first important measurements for the local loss coefficient ζ_{local} in a bifurcator was done in Munich from 1928 to 1931 and are also known as the *Munich experiments* (Vogel [1928], Petermann [1929] and Kinne [1931]). Vogel [1928] represented results for right-angled bifurcators with diameters from Ø15 mm to Ø45 mm. The form of the intersection edges of the bifurcator were either sharp-edged or rounded. Vogel [1928] mentioned also the proportional relationship between the pressure losses and the quadratic discharge rate. Petermann [1929] and Kinne [1931] repeated the test series of Vogel [1928] with a branching angle of 40° and 60°, respectively. McNown [1954] repeated the results of the *Munich experiments* with more accurate measurements at the hydraulic lab in Iowa. The branching angle of the test series in Iowa was 90° and the pipe diameters were from 12.7mm to 50.8mm.

Abundant data for loss coefficients of Y-bifurcators are presented in Miller [1990]. The parameter for the loss coefficients in Miller [1990] are the ratio of area for the main and branching pipe, discharge ratio, different branching angles and different forms of the intersection edge (rounded or edged). Kavianpour [2003] determined loss coefficients for two asymmetrical Y-bifurcators in a hydraulic model test made of plexiglass pipes. Kavianpour [2003] used for his investigation different stiffener inside the Y-bifurcator and also mentioned the important fact that no secondary flow should reach the turbine. Klasinc et al. [1992] reported the loss coefficient for an asymmetrical Y-bifurcator. For this model test the scaling law of Reynolds was used. This leads to too high discharge rates in the hydraulic model test and polynomial extrapolation methods are used. Based on the hydraulic model test of Klasinc et al. [1992] further investigations for a symmetrical Y-bifurcator are shown in this paper.

INSTALLATION OF THE HYDRAULIC MODEL TEST

The hydraulic model test is downscaled with a factor of 8.13, based on the Reynolds law, where the ratio of the inertia to the viscous force is the same in prototype and the model. In the hydraulic model test the Y-bifurcator consists of a main pipe with a inner diameter of 246mm and two branching pipes with a inner diameter of 172mm, respectively. The branching angle of the symmetrical Y-bifurcator is 40° and a stiffener is used for statical reason. 18 diameters (18-D) upstream of the Y-bifurcator is located a bend with an angle of 42° and after this bend exists a 12-D long pipe which ends in a flow conditioner (see Figure 1). Downstream of the Y-bifurcator is a 10-D long pipe with a inner diameter of 123mm and a confuser for each branch.

The flow conditioner consists of a pipe bundle to smooth the incoming flow and to get a fully developed turbulent flow before the flow enters the bend. The bend itself consists of 5 segments each of them rotated by 8.4°. The Y-bifurcator consists of three cone-shaped volumes and a stiffener. After the confuser in the hydraulic model test two pelton turbines are located in the prototype. Instead of the pelton turbines a 10-D long pipe is installed in the hydraulic model test to get a fully redeveloped turbulent flow.

The water supply for the model test is done by a reservoir in the laboratory. The water level in the reservoir is 13m above the symmetry line in the Y-bifurcator. With this water head level a maximum discharge rate of 0.12 (m³/s) is possible. Using pumps a discharge rate of 0.2 (m³/s) is possible and the maximum allowable pressure of 1.3 (bar), due to the strength of the plexiglass-pipe, is achieved in the installation. The water, after passing the model test, enters another

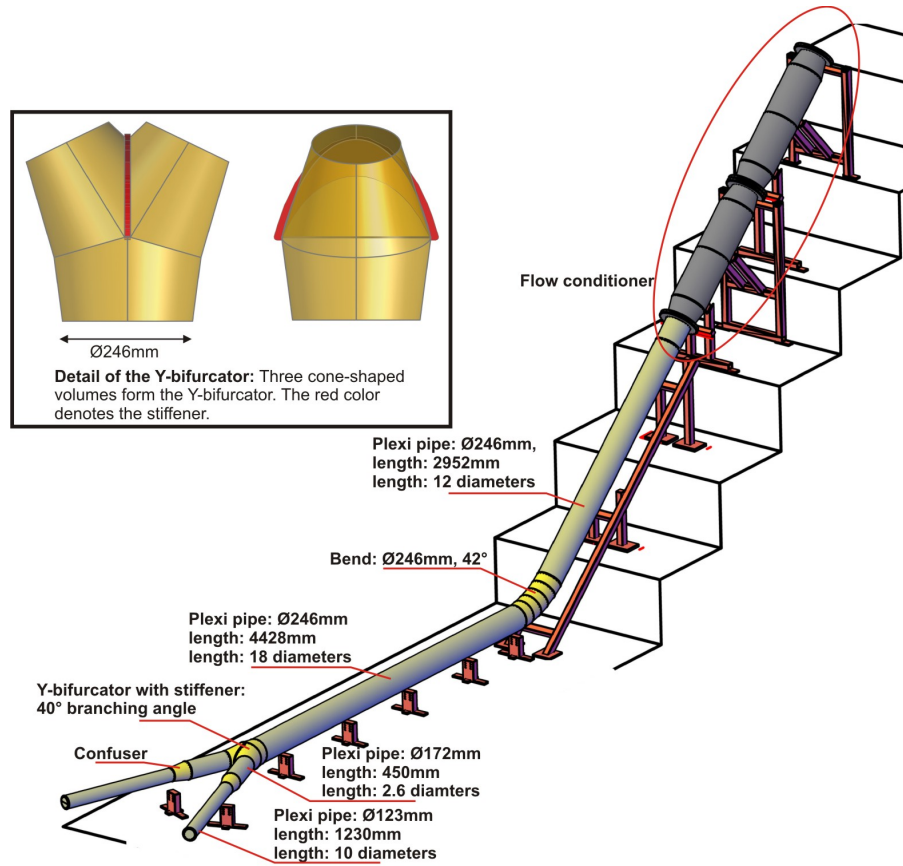


Figure 1: View of the whole plexiglass-pipe installation and a detail of the Y-bifurcator (top-left).

reservoir which is allocated below the model test.

Data acquisition

For the measurement of the loss coefficient ζ_{local} the pressure differences and the discharge rates are needed. The temperature needs not be recorded due to the constant temperature in the laboratory of $\pm 1^\circ C$.

The pressure difference is measured from the control section M1 in Figure 3 to every other control section, that is, from M2 to M6, L1 to L6 and R1 to R6 respectively. Each control section consists of 8 pressure-holes, regularly arranged around the plexiglass-pipe. The pressure-holes are burr-free and have a diameter of 1mm each. All eight bore holes are connected together to a single ring-line to get an average value of the head level in the control section. The pressure-sensor for the head level difference measurement has a maximum inaccuracy of $\pm 0.5\%$ of the full range. For each measuring campaign the pressure-probe is calibrated against a constant water head level. The ring-line is connected with the pressure-sensor by a air-free plastic tube. The signal (4-20mA) from the pressure-sensor is sent to a measurement amplifier and then recorded and stored by the software LabView.

The discharge rates are measured by three electromagnetic flow meters, one in the main pipe and one at each branching pipe. The measuring principle of the flow meter is based on the electromagnetic induction. The discharge rate can be continuously varied by two Howell-Bunger valves (see Figure 2) with an electric motor. The accuracy of the flow meters are $\pm 0.5\%$ of the discharge rate.



Figure 2: Howell-Bunger valve to vary continuously the discharge rate.

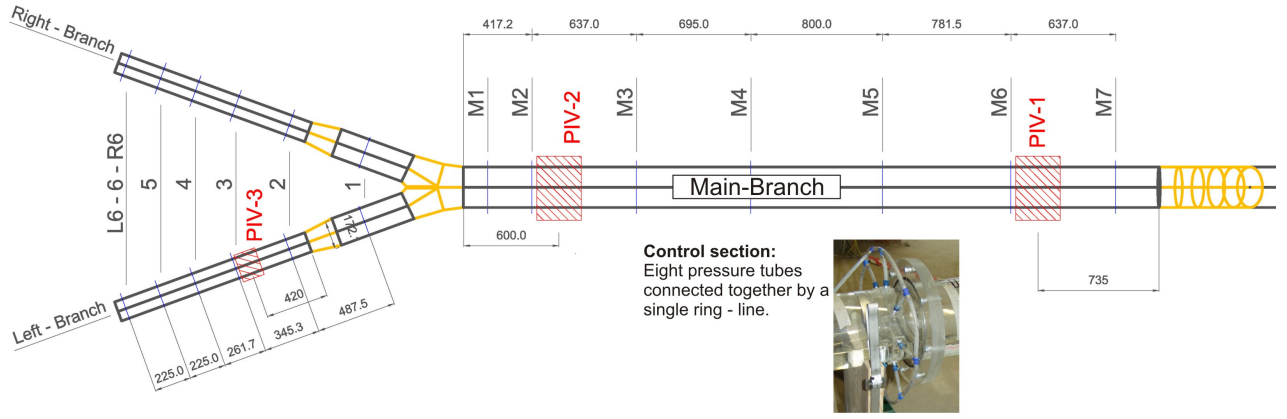


Figure 3: Plan view of the model test: Control section M1 to M7, L1 to L6 and R1 to R6. M, L and R stands for Main, Left and Right respectively. Position for PIV-measurements are the red hatched areas (two in the main pipe, one at the left-branch.)

Particle Image Velocimetry (PIV)

PIV is a non-intrusive device to measure velocities at a 2D-plane within the pipe installation. The basic parts of a PIV-measurement are a double pulsed laser which expands to a 2D laser light sheet, a high speed camera and a special PIV-box to avoid the astigmatism-effect due to the curved surface of the plexiglass-pipe. The natural particles in the flow ($\varnothing 0.1\text{mm}$) are used as seedings. As can be seen in Figure 4, for each PIV-box 4 planes are used to measure the velocity vector fields. The physical recording time of the flow is 1 second and the sampling rate 1 kHz. With this set-up the axial secondary flow as well as the roughness of the plexiglass wall can be measured. The software to calculate the velocity is DynamicStudio 2.20.18 from DantecDynamics.

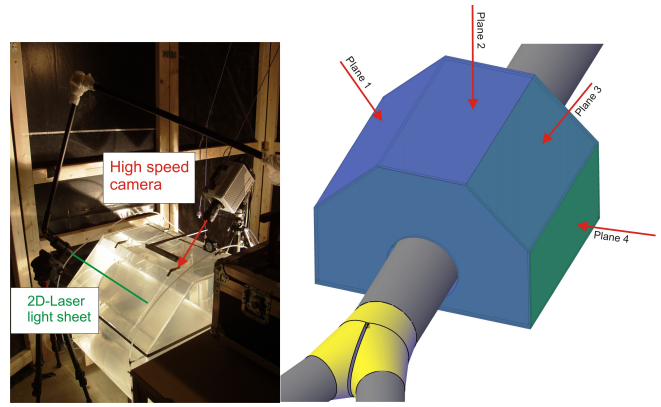


Figure 4: PIV - Set up: Figure left shows the laser, the PIV-box filled with water to avoid the astigmatism-effect and the High Speed Camera. Figure right shows the 4 investigation planes for the PIV-measurements.

HYDRAULIC EQUATIONS

The dimensionless coefficient ζ_{local} is defined as

$$\zeta_{local} = \frac{\Delta h_{local}}{V_{main\ pipe}^2 / 2g} \quad (1)$$

whereas Δh_{local} is the local head loss in the Y-bifurcator without friction losses due to pipe friction, $V_{main\ pipe}$ the mean velocity in the main pipe and g the gravity. To calculate Δh_{local} the relationship

$$\Delta h_{tot} = \Delta h_{local} + h_f \quad (2)$$

$$\Delta h + \alpha_1 \frac{V_1^2}{2g} = \alpha_2 \frac{V_2^2}{2g} + \Delta h_{tot} \quad (3)$$

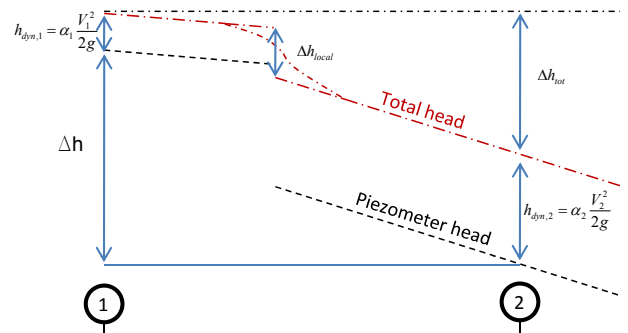


Figure 5: Definition of the local head loss Δh_{local}

is used. The introduced variables in Eq. 2 and Eq. 3 have following meaning (compare also with Figure 5): Δh_{tot} is the total head level difference which includes local (Δh_{local}) and pipe friction losses (h_f). Δh is the with the pressure-sensor measured piezometer head level difference in the hydraulic model test. α is the velocity correction factor for a non-uniform velocity distribution. V_1 and V_2 are the measured velocities due to the flow meters before and after the Y-bifurcator. h_f is the pipe friction due to the Darcy-Weisbach equation (see Eq. 4), whereas L is the length, D the diameter, V the velocity in the pipe and λ the friction coefficient. λ can be calculated with the Colebrook equation (see Eq. 5), whereas Re is the Reynolds number and k the equivalent sand roughness.

Finally, to calculate the friction loss h_f substitution cylinders need to be defined along the pipe installation. Depending on the control section in Figure 3 for the loss coefficient calculation, 5 substitution cylinders are defined to calculate the friction loss of the pipe installation between two control sections and subtract it from the total head level difference (Three cylinders for the Y-bifurcator and one for the main- and one for the branching pipe).

An essential point in the loss coefficient calculation is the determination of the equivalent sand roughness in the plexiglass installation to determine the pipe friction losses h_f . A simple method to calculate the sand roughness is the application of the results of the measured piezometer head level difference where a linear loss gradient appears.

This certain flow behaviour exist between the bend and the Y-bifurcator and downstream of the Y-bifurcator.

The other method to estimate the equivalent sand roughness is the use of the logarithmic law of the wall by von Kármán [1930]. The law of the wall is shown in Eq. 6, whereas u_τ is the friction velocity, κ the von Kármán constant, y the distance normal to the pipe wall, u the corresponding axial velocity for the distance y and k the equivalent sand roughness. Solving Eq. 6 in respect to u_τ a set of equations are defined and with PIV-results close to the wall (wall distance y_i and the corresponding axial velocity u_i , i denotes the step of the discrete points of the velocity vector field) the sand roughness can be calculated (see Eq. 8). With Eq. 6 and Eq. 7 the logarithmic law of the wall, which is valid between $30 \leq y^+ \leq 300$, can be plotted.

In Eq. 3 the α coefficients need to be defined. Due to the bend in the installation a slight secondary effect still exists shortly before the flow enters the Y-bifurcator. With PIV it is possible to calculate the α value before and after the Y-bifurcator (compare also with the label *PIV-2* and *PIV-3* in Figure 3). The analytical equation for the α coefficient (Preißler and Bollrich [1985]) is shown in Eq. 9, whereas A is the cross section area of the pipe and u the axial velocity due to PIV-measurement.

$$\alpha = \frac{1}{u_{mean}^3 A} \int^A u_A^3 dA \quad (9)$$

EXTRAPOLATION OF THE PIEZOMETER HEAD LEVEL DIFFERENCE

The scaling ratio, based on the Reynolds law, between the prototype and the hydraulic model test is 8.13. This means that a 8.13-times higher velocity would be needed in the hydraulic model test which is not possible to achieve. Therefore, an extrapolation method is needed for the function of the discharge rate in respect to the piezometer head level difference Δh . In the work of Klasinc et al. [1992] a second order polynomial equation was used for the extrapolation due to the quadratic behaviour of the head loss in respect to the discharge rate. In this paper, the extrapolation is done by using a linear function of the **quadratic discharge rate** in respect to the head loss. With *Matlab(R2008a)* it is possible to find either for a linear- or a quadratic polynomial extrapolation the necessary function to approximate the measured head loss and extrapolate it beyond the possible discharge rate in the hydraulic model test. The subroutine in *Matlab* for the extrapolation method is called *polyfit* and is based on the least square method.

RESULTS AND DISCUSSION

Roughness

In Figure 6 the results for the determination of the roughness k due to the Δh_{local} - measurements are presented. The standard value for k of a plexiglass-pipe is 0.0015 (mm). Because of the built-in components (e.g. flanges) the roughness increases to a value for the main pipe of 0.01 and for the branching pipes of 0.019 (mm), respectively. The roughness k is also double checked by a PIV-measurement as can be seen in Figure 7. With Eq. 8 the k coefficient yields 0.0187 (mm) for the first point in the logarithmic layer. Therefore, the chosen roughness for the plexiglass installation is 0.019 (mm) due to the PIV- and Δh_{local} - measurements.

In Figure 8 the velocity profiles for all four planes at the position PIV-2 (compare with Figure 4) are shown. The distorted velocity profile, which is caused by the secondary flow initialized by the bend is still visible. The calculated mean value of the alpha coefficient for all four planes is 1.08. This coefficient is calculated for a symmetrical flow distribution with 0.08 (m³/s) in the left and right branch. For an asymmetrical flow distribution (only left branch with 0.08 (m³/s)) the coefficient increase slightly to 1.10. It can be expected that the α coefficient of 1.08 is in good agreement with the α coefficient of other flow distributions. Therefore the α coefficient of 1.08 is chosen upstream of the Y-bifurcator. Downstream of the Y-bifurcator the measured α coefficient is 1.01 due to smoothing effect of the confuser.

$$h_f = \lambda \frac{L}{D} \frac{V^2}{2g} \quad (4)$$

$$\frac{1}{\sqrt{\lambda}} = -2.0 \lg \left(\frac{2.51}{Re \sqrt{\lambda}} + \frac{k/D}{3.71} \right) \quad (5)$$

$$u^+ = \frac{u}{u_\tau} = \frac{1}{\kappa} \ln \left(\frac{y}{k} \right) + 8.5 \quad (6)$$

$$y^+ = \frac{y u_\tau}{\nu} \quad (7)$$

$$\frac{\frac{1}{\kappa} \ln \left(\frac{y_i}{k} \right) + 8.5}{u_i} = \frac{\frac{1}{\kappa} \ln \left(\frac{y_{i+1}}{k} \right) + 8.5}{u_{i+1}} \quad (8)$$

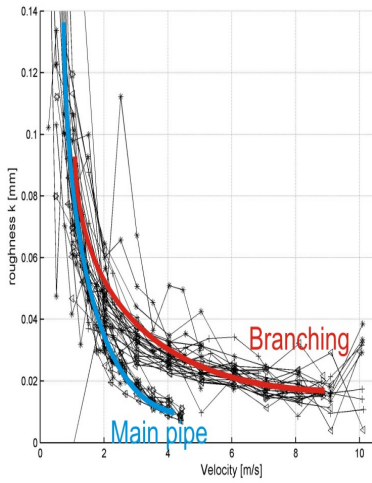


Figure 6: Sand roughness calculated from the Δh_{local} - measurements where a linear friction gradient exists.

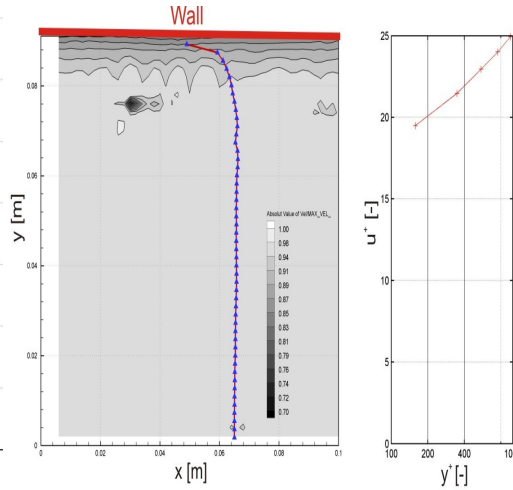


Figure 7: Left: Velocity profile (blue line) and contour plot close to the wall at Position PIV-2 in Figure 3. Right: Evaluation of Eq. 6 and Eq. 7.

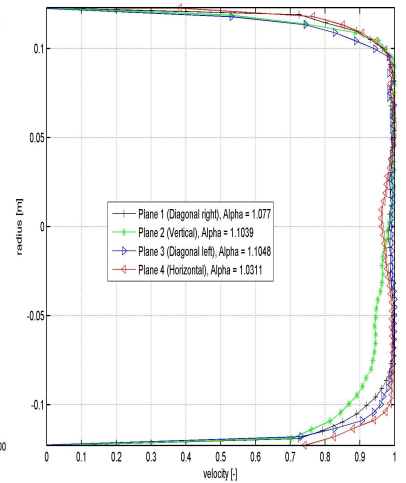


Figure 8: Velocity profiles made dimensionless with u_{max} extracted at the four planes in Figure 4 with their respective coefficient α .

Difference between the linear or quadratic polynomial extrapolation of the ζ_{local} coefficient

In Figure 9 an example of a linear ζ_{local} extrapolation is shown. The red line shows the measured and the blue line the extrapolated values. The flow distribution is symmetrically ($Q_{left} = Q_{right}$). As can be seen in Figure 9 the loss coefficient is nearly independent at a Reynolds number higher than 1×10^6 .

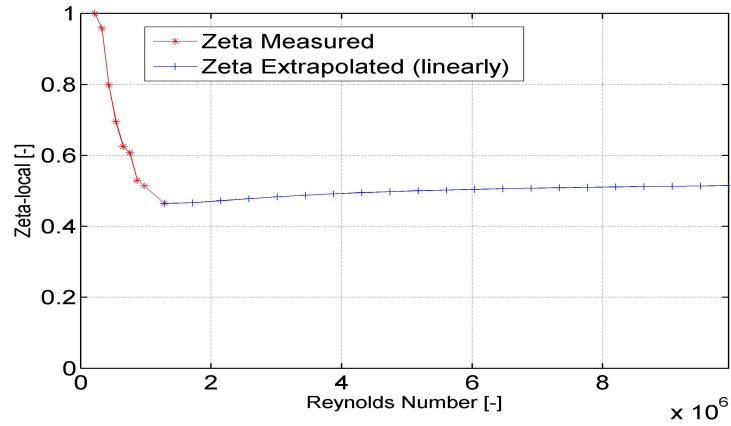


Figure 9: Example of ζ_{local} coefficient extrapolated with a linear polynomial, distribution: $Q_{left}/Q_{main} = 0.57$. Due to data protection the ζ_{local} coefficient is made dimensionless with the maximum ζ_{local} coefficient

In Figure 10 the extrapolated ζ_{local} values for a linear- and a quadratic polynomial extrapolation are shown. All loss coefficients in Figure 10 are in the hydraulic rough area, that is, they are independent of the Reynolds number. The loss coefficients are calculated for four flow distributions (e.g. LC1 is Load case 1 with $Q_{left}/Q_{main} = 1$). The figure to the left in Figure 10 represents the result of the linear and the figure to the right of the quadratic polynomial extrapolation, respectively. Due to the numerous pressure difference measurements it is possible to calculate the loss coefficient for five pair of control sections, that is, control section M1-L3/R3, M3-L3/R3, M4-L4/R4, M5-L5/R5 and M6-L6/R6.

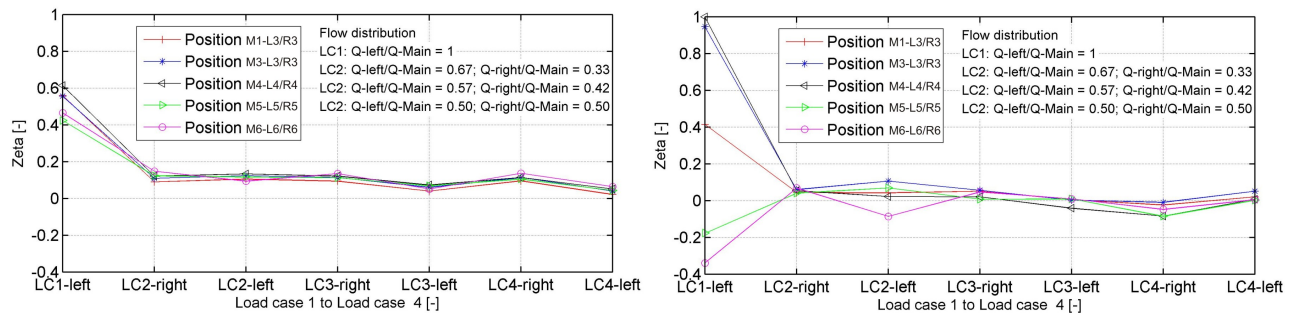


Figure 10: Comparison of the linear- (left figure) and quadratic polynomial extrapolation. ζ_{local} is plotted against the flow distribution (LC1 to LC4) and for five different pairs of control sections (M1-L3/R3 to M6-L6/R6). Due to data protection the ζ_{local} coefficient is made dimensionless with the maximum ζ_{local} coefficient

If one checks the left figure in Figure 10 all 5 five curves (Position M1-L3/R3 to M6-L6/R6) are almost identical, which is reasonable because ζ_{local} does not include any friction loss h_f and thus is independent of the position of the control sections as long as no strong secondary flow occurs in the pipe. If the left picture is considered there can be clearly distinguished unreasonable loss coefficients for LC1-left, where the coefficients have a very high deviation and even negative quantities. Also, all five lines do not collapse to one single line as good as it can be seen in the left figure. In summary it can be said that the linear polynomial extrapolation causes less deviation as the quadratic polynomial extrapolation. The linear extrapolation is therefore more suitable for the determination of the loss coefficient ζ_{local} .

CONCLUSION

Due to the Reynolds law of similarity very high discharge rates would be necessary in the hydraulic model test. With a linear extrapolation of the quadratic discharge rate in respect to the piezometer head level difference the loss coefficient ζ_{local} can be calculated. It can be shown that the linear extrapolation yields more reasonable results than does the polynomial extrapolation of Klasinc et al. [1992]. The roughness calculation either with PIV or with the measured head level differences in the hydraulic model test shows almost the same result. The secondary flow caused by the bend is quantified with the velocity correction factor α and thus it is possible to calculate the loss coefficient ζ_{local} .

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NOMENCLATURE

α	Velocity correction factor (-)
Δh	Piezometer head level difference (m)
Δh_{local}	Local head loss (m)
Δh_{tot}	Total head level difference (m)
κ	von Kármán constant (-)
λ	Friction coefficient (-)
ν	Kinematic viscosity (m^2/s)
ζ_{local}	Local loss coefficient (-)
A	Area (m^2)
D	Inner diameter (m)
g	Gravity (m/s^2)
h_f	Friction loss (m)
k	Equivalent sand roughness (m)
L	Length of a substitution cylinder (m)
Q	Discharge rate (m^3/s)
Re	Reynolds - Number (-)
u	Axial velocity (m/s)
u^+, y^+	Dimensionless distance and velocity (-)
u_τ	Friction velocity (m/s)
V	Mean velocity (m/s)
y	Distance from the center line of the pipe (m)

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