

Improved control valve sizing for multiphase flow

$$\phi = \sqrt{\frac{\frac{v_{l1}}{v_{g1}}}{\left[1 + \dot{x}_i \left[\left(\frac{v_{g1}}{v_{l1}} \right)^{\frac{1}{6}} - 1 \right] \right] \left[1 + \dot{x}_i \left[\left(\frac{v_{g1}}{v_{l1}} \right)^{\frac{5}{6}} - 1 \right] \right]}}$$

$$\left(\frac{p_{vc, crit}}{p^1} \right)^2 + (\omega^2 - 2\omega) \left[1 - \left(\frac{p_{vc, crit}}{p^1} \right) \right]^2 + 2\omega^2 \ln \left(\frac{p_{vc, crit}}{p^1} \right) + 2\omega^2 \left[1 - \left(\frac{p_{vc, crit}}{p^1} \right) \right] = 0$$

$$N = \left(\dot{x}_i + cp_{l1} T_{l1} p^1 \left(\frac{v_{l1} - v_{g1}}{\Delta h_{vl}} \right)^2 \ln(1 - x_{crit, N=1}) \right)^{\alpha}$$

$$k_{vs} = \sqrt{\frac{\Delta p_0}{\Delta p}} \frac{1}{\sqrt{\rho_0 \rho^1}} W \frac{1}{Y_{MP}} \quad \omega_{N=1} = \frac{\dot{x}_i v_{g1}}{v_{l1}}$$

$$\frac{W}{A_{vc}} = \frac{\sqrt{\omega \ln \left(\frac{p^1}{p_{vc}} \right) - (\omega - 1) \left(1 - \frac{p_{vc}}{p^1} \right)}}{\left[\omega \left(\frac{p^1}{p_{vc}} - 1 \right) + 1 \right]} \sqrt{2 \frac{p^1}{v_{l1}}} \phi$$

Special print from
 "Hydrocarbon Processing"
 March 2005

By:
 Ralf Diener, BASF AG
 Dr.-Ing. Jörg Kiesbauer, SAMSON AG
 Jürgen Schmidt, c/o BASF AG

Improved control valve sizing for multiphase flow

Ralf Diener, BASF AG

Jörg Kiesbauer, SAMSON AG

Jürgen Schmidt, c/o BASF AG

Control valves are frequently used in chemical and petrochemical plants to control medium mixtures which comprise a gas/vapor phase as well as a liquid phase as they enter the valve. The IEC 60534-2-1 Standard is primarily applied to calculate the required valve flow capacity (C_v or k_v coefficient) for single-phase media in the form of gases or liquids. However, this standard does not apply to multiphase flow. A standardized sizing model does not exist even in literature. As a consequence, plant operators and manufacturers employ various published methods according to their own experience, which may lead to considerably different results depending on the operating and medium data involved. This article proposes a new approach to size control valves that handle multiphase mixtures consisting of gases/vapors and liquids. It takes into account how the change in density of the mixture flowing through the valve affects the flow capacity of the valve by using an expansion factor similar to the standard for gaseous media. It provides a high level of predictive accuracy when compared to test data. This expansion factor is easy to integrate into the IEC 60534-2-1 Standard.

Previous prediction methods

The IEC 60534-2-1 Standard specifies that the required flow coefficient (k_v) can be calculated using the following equation, provided the medium is not highly viscous and the pipeline upstream and downstream of the valve is straight [1, 2]:

$$k_v = \sqrt{\frac{\Delta p_0}{\Delta p}} \left(\frac{1}{\sqrt{\rho_0 \rho_1}} \right) W \left(\frac{1}{Y} \right), \quad \Delta p \leq \Delta p_{max} \quad (1)$$

The pressure difference between the inlet and outlet of the valve Δp_0 as well as the density ρ_0 apply to the reference condition of 1 bar and 1000 kg/m³. Y is the expansion factor and is equal to 1 for liquids, whereas it is related to the pressure difference for gases and vapors. Choked flow occurs when the pressure difference Δp_{max} is reached. Even if the pressure downstream of the valve is further reduced, the maximum, so-called critical mass flow rate cannot be increased. The pressure difference varies in compressible and non-compressible fluids [1, 2].

In cases where a multiphase flow comprising of gases and liq-

uids arises at the valve inlet the density ρ_1 depends on gas/vapor void fraction and can no longer be uniquely defined. Mixture property data are no definite physical properties and therefore they can be defined in several ways.

The addition model is the simplest method. It deals with each individual phase separately, calculating the flow coefficient of each phase according to the above equation. Both flow coefficients are then added together to result in an overall flow coefficient [2].

The correction factor proposed by Sheldon and Schuder [3] which is dependent on the volume share of the gas or vapor phase at the valve inlet raises the flow coefficients which are calculated too low in the addition model.

Another prediction method arises from the concept that assumes both phases are equally mixed, i.e. homogeneous, and that they flow at a constant velocity. In this greatly simplified homogeneous model, the density ρ_1 at the valve inlet is calculated, as demonstrated in Table 1 using $\rho_1=1/v_1$ [4] and then applied to the above equation (1).

The results of such simple model types deviate considerably from the measured data as stated in [2, 5].

A new approach based on more accurate physical assumptions is introduced in detail in [2, 5] which provides a much better prediction accuracy compared to the simple empirical methods. This method includes the following assumptions:

- The change in the flow condition between the valve inlet and vena contracta is assumed to be frictionless and with no wall heat transfer (isentropic change of state).
- In contrast, the flow in the area downstream of the vena contracta up to the valve outlet is viewed to be afflicted by losses.
- Depending on the valve location, different flow patterns are assumed for calculation the mixture density (homogeneous density and momentum density at various flow velocities of the phases by using a special slip model).
- The local mass flow quality of the flow is calculated from the changes in isenthalpic or isentropic state depending on the area of flow in the valve and applied to calculate the mixture density.
- A non-equilibrium factor N is used to take into account the boiling delay and the thermodynamic non-equilibrium between the individual phases.

The model requires an iterative solution of the mathematical equations and involves property data dependent on pressure and temperature (density, enthalpy, and entropy), which are only feasible in practice in certain cases. Therefore in this paper a simply applicable model based on the homogenous non-equilibrium Diener/Schmidt method (HNE-DS method) for the sizing of throttling devices in two-phase flow is proposed. The HNE-DS method extends the ω -method, originally developed by Leung, by adding a boiling delay coefficient to include the degree of thermodynamic non-equilibrium at the start of the nucleation of small vapor mass fractions upstream of the fitting. The additional introduction of a slip correction factor to take account of hydrodynamic non-equilibrium (slip) also makes it possible to calculate reliably the flow rate through control valves and orifices in both, flashing and non-flashing flow.

In this paper the HNE-DS method is adapted to the equations of the IEC 60534-2-1 Standard. The essential calculation steps are outlined in the following section.

New method for the IEC 60534-2-1 Standard using an expansion factor

The new HNE-DS method basically assumes that the flow pattern in the control valve is homogeneous but gas and liquid are not in equilibrium – a non-equilibrium factor N represents the boiling delay. Compared to the more complex integration method by Diener [5] it is easier to apply in practice.

The assumption concerning the homogeneous flow and the introduction of the non-equilibrium factor allow for a calculation of the mass flux (mass flow rate W based on the area of the vena contracta A_{vc}) without any iteration (see [6, 7] for exact derivation).

$$\frac{W}{A_{vc}} = \frac{\sqrt{\omega \ln\left(\frac{p_1}{p_{vc}}\right) - (\omega - 1) \left(1 - \frac{p_{vc}}{p_1}\right)}}{\left[\omega \left(\frac{p_1}{p_{vc}} - 1\right) + 1\right]} \sqrt{2 \frac{p_1}{v_1}} \quad (2)$$

The compressibility coefficient ω contains operating (stagnation) conditions and property data as well as the non-equilibrium factor N :

$$\omega = \frac{\dot{x}_1 v_{g1} + c p_{l,1} T_1 p_1 \left(\frac{v_{g1} - v_{l1}}{\Delta h_{v1}}\right)^2}{v_1} N \quad (3)$$

The area of the vena contracta A_{vc} can be expressed using the flow coefficient as is usual in the IEC 60534 Standard:

$$A_{vc} = \sqrt{\frac{\rho_0}{2 \Delta p_0}} F_L k_v \quad (4)$$

The pressure ratio p_1/p_{vc} increases as the pressure difference between inlet and outlet increases. The mass flow reaches its maximum and remain constant when the ratio of inlet and outlet pressure falls below the so-called critical pressure ratio. The critical pressure ratio ($p_1/p_{vc, crit}$) can be determined by resetting the first derivation of W/A_{vc} according to (p_1/p_{vc}):

$$\left(\frac{p_{vc, crit}}{p_1}\right)^2 + (\omega^2 - 2\omega) \left[1 - \left(\frac{p_{vc, crit}}{p_1}\right)\right]^2 + 2\omega^2 \ln\left(\frac{p_{vc, crit}}{p_1}\right) + 2\omega^2 \left[1 - \left(\frac{p_{vc, crit}}{p_1}\right)\right] = 0 \quad (5)$$

Table 1 contains an approximated equation for values of $\omega \geq 2$ in which an iterative solution of the above equation is avoided. The non-equilibrium factor N is approached on the basis of the measured data in [5] as a power law function, where $\dot{x}_e(p_{vc, crit})$ is to be regarded as the vapor content in equilibrium state in the vena contracta for a critical flow. In [6, 7] an exponent of $\alpha=0.6$ (travel < 25 mm) or $\alpha=0.4$ (travel \geq 25 mm) for control valves is proposed:

$$N = \left[\dot{x}_e(p_{vc, crit})\right]^\alpha \quad (6)$$

Based on the mass flow quality \dot{x} at the valve inlet, the increase of the equilibrium factor between the stagnation condition upstream of the valve inlet and the vena contracta can be determined with the assumption of an equilibrium state:

$$N = \left[\dot{x}_1 + cp_{l,1} T_1 p_1 \left(\frac{v_{g,1} - v_{l,1}}{\Delta h_{v,1}^2} \right) \ln \left(\frac{p_{vc, crit}}{p_1} \right) \right]^\alpha \quad (7)$$

As a result, equations are provided which allow the mass flow rate or flow coefficient to be calculated. As already mentioned, the boiling delay which affects the mass flow rate the most, is taken into account by the factor N for two-phase flashing flow. In [6, 7] the authors additionally propose using a slip correction factor ϕ for the hydrodynamic non-equilibrium of both phases, i.e., both phases have different flow velocities and are not homogeneously distributed.

$$\frac{W}{A_{vc}} = \frac{\sqrt{\omega \ln \left(\frac{p_1}{p_{vc}} \right) - (\omega - 1) \left(1 - \frac{p_{vc}}{p_1} \right)}}{\left[\omega \left(\frac{p_1}{p_{vc}} - 1 \right) + 1 \right]} \sqrt{2 \frac{p_1}{v_1}} \phi \quad (8)$$

with

$$\phi = \sqrt{\frac{\frac{v_1}{v_{l,1}}}{\left\{ 1 + \dot{x}_1 \left[\left(\frac{v_{g,1}}{v_{l,1}} \right)^{\frac{1}{6}} - 1 \right] \right\} \left\{ 1 + \dot{x}_1 \left[\left(\frac{v_{g,1}}{v_{l,1}} \right)^{\frac{5}{6}} - 1 \right] \right\}}} \quad (9)$$

To summarize, the following applies for the flow coefficient:

$$k_v = \sqrt{\frac{\Delta p_0}{\Delta p}} \frac{1}{\sqrt{\rho_0 \rho_1}} W \frac{1}{Y_{MP}} \quad (10a)$$

or

$$W = \sqrt{\frac{\Delta p}{\Delta p_0}} \sqrt{\rho_0 \rho_1} k_v Y_{MP} \quad (10b)$$

including the new expansion factor Y_{MP} for multiphase flow:

$$Y_{MP} = \frac{\sqrt{\omega \ln \left(\frac{p_1}{p_{vc}} \right) - (\omega - 1) \left(1 - \frac{p_{vc}}{p_1} \right)}}{\left[\omega \left(\frac{p_1}{p_{vc}} - 1 \right) + 1 \right]} \phi \frac{F_L}{\sqrt{x}} \quad (11)$$

Step-by-step determination of the expansion factor Y_{MP}

Table 1 shows the necessary input parameters and each step required to calculate the expansion factor as well as the mass

flow rate W or the flow coefficient k_v . It is essential that the data \dot{x} , $v_{g,1}$, $v_{l,1}$, $\Delta h_{v,1}$, $cp_{l,1}$ related to the (stagnation) condition at the valve inlet are available.

Similar to the method described in the IEC 60534 Standard, the differential pressure ratio x and the critical pressure ratio x_{crit} ($\Delta p_{max} = p_1 x_{crit}$) are included in the new method. The generally unknown pressure p_{vc} can be replaced with known parameters. As a result, the expansion factor is then derived directly from the differential pressure ratio x .

The first step in the interim calculations is to determine the compressibility coefficient ω without the non-equilibrium factor N (i.e., $N=1$), and following this, x_{crit} and N and then the final compressibility coefficient ω . The critical pressure difference ratio x_{crit} can be directly determined for $\omega \geq 2$, in other cases, just in an iterative way using the second equation.

Prediction example

The following example in Table 2 illustrates a prediction to explain the procedure more clearly.

Accuracy of the new approach

The average predictive accuracy of the models is based on the values obtained for the variance of the logarithmic deviations between the experimental and calculated values, Table 3. The advantages of using this parameter are discussed in [5] and showed in the past to allow for a balanced description of the merits of each correlation. (f defines the independent number of variables in the model and can be neglected if n is greater than 500).

In two-phase flow the real mass flow rate can be determined only by experiments, for example measurements made at the Technical University of Hamburg-Harburg, in the department of Prof. Dr. L. Friedel, and by SAMSON AG. There, the mass flow rate was measured when mixtures of steam and boiling water were passed through control valves. The test valves used had nominal diameters of 25, 50 and 80 mm and had different types of valve plug (V-port plug, parabolic plug and perforated plug). The test setups and the measuring methods and technology employed are described in [4]. In Fig. 1 the mass flow rates calculated by the complex method are plotted against the measured mass flow rates. The deviations are almost symmetrically distributed on the diagonal. The variance of the logarithmic deviations amounts only 11 %. In opposite to this model allows the addition model and the model by Sheldon and Schuder no reasonable predictions, Fig. 2 and Fig. 3. The deviations are distributed asymmetrically along the diagonal in a range from more than 100% underprediction up to even 100 % overprediction. The variance of the logarithmic deviations amount to more than 40 %.

The new approach leads to a considerable improvement in

Input parameters	
Inlet pressure	p_1
Inlet temperature	T_1
Outlet pressure	p_2
Pressure difference	$\Delta p = p_1 - p_2$
Mass flow rate	W
Flow coefficient	or k_v
Mass vapor or gas content [-]	\dot{x}_1
Specific volume of vapor or gas phase at the inlet ($1/\rho$)	v_{g1}
Specific volume of liquid phase at the inlet ($1/\rho$)	v_{l1}
Heat of vaporization in relation to p_1 and T_1 (liquid-vapor mixture only)	Δh_{v1}
Specific heat capacity of liquid phase in relation to p_1 and T_1 (liquid-vapor mixture only)	cp_{l1}

Interim calculations	
Pressure difference ratio x	$x = \frac{(p_1 - p_2)}{p_1}$
Homogeneous specific volume of mixture	$v_1 = \dot{x}_1 v_{g1} + (1 - \dot{x}_1) v_{l1}$
Slip correction factor ϕ	$\phi = \sqrt{\frac{\frac{v_1}{v_{l1}}}{\left\{1 + \dot{x}_1 \left[\left(\frac{v_{g1}}{v_{l1}} \right)^{\frac{1}{6}} - 1 \right] \right\} \left\{1 + \dot{x}_1 \left[\left(\frac{v_{g1}}{v_{l1}} \right)^{\frac{5}{6}} - 1 \right] \right\}}}$
Compressibility coefficient ω initially used for thermodynamic equilibrium ($N=1$)	
Vapor:	$\omega_{N=1} = \frac{\dot{x}_1 v_{g1} + cp_{l1} T_1 p_1 \left(\frac{v_{g1} - v_{l1}}{\Delta h_{v1}} \right)^2}{v_1}$
Gas ($\dot{x} = \text{constant}$):	$\omega_{N=1} = \frac{\dot{x}_1 v_{g1}}{v_1}$
Critical pressure difference ratio $x_{crit} = 1 - p_{vc,crit}/p_1$ initially used for thermodynamic equilibrium ($N=1$)	$x_{crit,N=1} = 1 - \left[0.55 + 0.217 \ln \omega_{N=1} - 0.046 (\ln \omega_{N=1})^2 + 0.004 (\ln \omega_{N=1})^3 \right]$
Non-equilibrium factor N Vapor:	$N = \left(\dot{x}_1 + cp_{l1} T_1 p_1 \left(\frac{v_{l1} - v_{g1}}{\Delta h_{v1}^2} \right) \ln(1 - x_{crit,N=1}) \right)^\alpha$ $\alpha = 0.6$ control valves with valve travel < 25 mm $\alpha = 0.4$ control valves with valve travel \geq 25 mm
Gas ($\dot{x} = \text{constant}$):	$N=1$
Recalculation of compressibility coefficient ω Vapor:	$\omega = \frac{\dot{x}_1 v_{g1} + cp_{l1} T_1 p_1 \left(\frac{v_{g1} - v_{l1}}{\Delta h_{v1}} \right)^2}{v_1} N$
Gas ($\dot{x} = \text{constant}$):	$\omega = \omega_{N=1}$

Vapor:	
Recalculation of critical pressure difference ratio $x_{crit} = 1 - p_{vc,crit}/p_1$ for thermodynamic non-equilibrium	For $\omega \geq 2$: $x_{crit} = 1 - \left[0.55 + 0.217 \ln \omega - 0.046 (\ln \omega)^2 + 0.004 (\ln \omega)^3 \right]$ For $\omega < 2$: $(1 - x_{crit})^2 + (\omega^2 - 2\omega)(x_{crit})^2 + 2\omega^2 \ln(1 - x_{crit}) + 2\omega^2 x_{crit} = 0$
Gas ($\dot{x} = \text{constant}$):	$x_{crit} = x_{crit,N=1}$

Final results	
Critical pressure difference	$\Delta p_{max} = x_{crit} p_1$
Expansion factor Y_{MP}	For $\Delta p < \Delta p_{max}$: $Y_{MP} = \frac{\sqrt{-\omega \ln(1-x) - (\omega-1)x}}{\left[\omega \left(\frac{x}{1-x} \right) + 1 \right]} \phi \frac{F_L}{\sqrt{x}}$ For $\Delta p \geq \Delta p_{max}$: $Y_{MP} = \frac{\sqrt{-\omega \ln(1-x_{crit}) - (\omega-1)x_{crit}}}{\left[\omega \left(\frac{x_{crit}}{1-x_{crit}} \right) + 1 \right]} \phi \frac{F_L}{\sqrt{x_{crit}}}$
Flow coefficient k_v	For $\Delta p < \Delta p_{max}$: $k_v = \sqrt{\frac{\Delta p_0}{\Delta p}} \frac{1}{\sqrt{\rho_0 \rho_1}} W \frac{1}{Y_{MP}}$ For $\Delta p \geq \Delta p_{max}$: $k_v = \sqrt{\frac{\Delta p_0}{\Delta p_{max}}} \frac{1}{\sqrt{\rho_0 \rho_1}} W \frac{1}{Y_{MP}}$
Mass flow rate W	For $\Delta p < \Delta p_{max}$: $W = \sqrt{\frac{\Delta p}{\Delta p_0}} \sqrt{\rho_0 \rho_1} k_v Y_{MP}$ For $\Delta p \geq \Delta p_{max}$: $W = \sqrt{\frac{\Delta p_{max}}{\Delta p_0}} \sqrt{\rho_0 \rho_1} k_v Y_{MP}$

Table 1: Parameters and equations used in the new model

prediction accuracy compared to the methods mentioned in Section 2, Fig. 4. It achieves almost the same accuracy as the much more complex method shown in Fig. 1. The deviations are also almost symmetrically distributed on the diagonal and the variance of the logarithmic ϕ deviations amount to almost 17 %. This value is acceptable for the control valve sizing because in most cases a safety margin of min. 20% is applied.

Summary

This article proposes a method to determine the flow coefficient or to predict the mass flow rate in valves controlling non-flashing two-phase flow or flashing two-phase flows. Similar to the existing IEC 60534 Standard for predicting the flow in control valves, the expansion factor applied to gas flows has been

Input parameters	Steam, control valve with $F_L = 1$
Inlet pressure	$p_1 = 10$ bar
Inlet temperature	$T_1 = 182.89$ °C
Outlet pressure	$p_2 = 5$ bar
Pressure difference	$\Delta p = 10 - 5 = 5$ bar
Flow coefficient	10 m ³ /h
Mass vapor or gas content [-]	$\dot{x}_1 = 0.01$
Specific volume of vapor or gas phase at the inlet (1/ ρ)	$v_{g1} = 0.209$ m ³ /kg
Specific volume of liquid phase at the inlet (1/ ρ)	$v_{l1} = 0.001128$ m ³ /kg
Heat of vaporization in relation to p_1 and T_1	$\Delta h_{v1} = 2,019$ kJ/kg
Specific heat of liquid phase in relation to p_1 and T_1	$c_{p1} = 4,400$ J/kg/K

Interim calculations	
Pressure difference ratio x	$x = (10 - 5) / 10 = 0.5$
Homogeneous specific mixed volume	$v_1 = 0.01 \times 0.209 + (1 - 0.01) \times 0.001128 = 0.00321$ m ³ /kg
Slip correction factor ϕ	$\phi = 1.26$
Compressibility coefficient ω initially used for thermal equilibrium ($N=1$)	$\omega_{N=1} = 7.28$
Critical pressure difference ratio x_{crit} initially used for thermodynamic equilibrium ($N=1$)	$x_{crit, N=1} = 0.169$
Non-equilibrium factor N :	$N = 0.1194$ ($\alpha=0.6$, valve travel < 25 mm)
Recalculation of compressibility coefficient ω	$\omega = 1.44$
Recalculation of critical pressure difference ratio x_{crit} for thermodynamic non-equilibrium	$x_{crit} = 0.38$

Final results	
Critical pressure difference	$\Delta p_{max} = 3.47$ bar
Expansion factor Y_{MP}	$Y_{MP} = 0.82$
Mass flow rate W	$W = 8,558$ kg/h

Table 2: Calculation example

extended to mixtures comprising gases/vapors and liquids. Just a small amount of additional information on physical property data is required for the calculation compared to the original method according to the IEC 60534 Standard. The entire steam table is not required. The results achieve a high level of accuracy which is sufficient for most engineering purposes. The authors will endeavor to integrate this method into Part 2-1 of the IEC 60534 Standard on its next revision. The method can, in principle, also be extended for a flow with more than two phases. However, the corresponding measured data are required.

Statistical number	Definition
Variance of logarithmic deviations	$S_{in} = e^{\left[\frac{\sum_{i=1}^n X_{i,ln}^2}{n-1} \right]} - 1$ $X_{i,ln} = \ln \frac{W_{i,exp}}{W_{i,calc}}$

Table 3: Definition of statistical number used to characterize the average predictive accuracy of the models

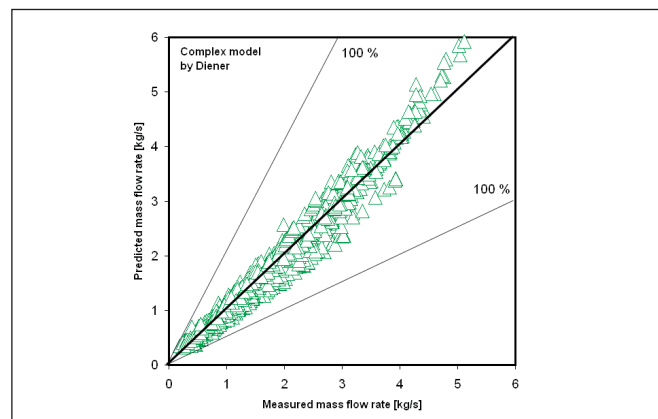


Fig. 1: Accuracy of reproduction of control valve mass flow rates by means of the complex model by Diener for steam/water flow having low vapor content

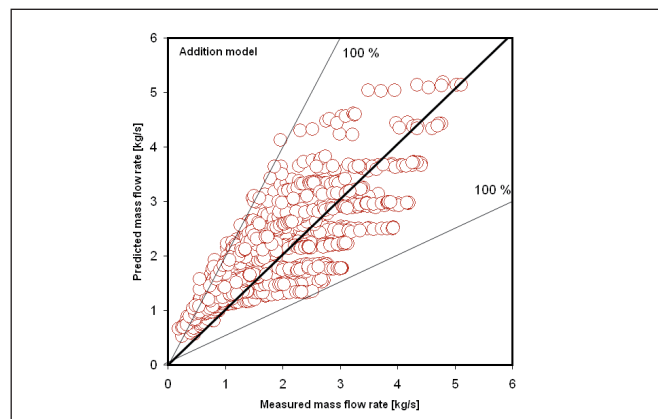


Fig. 2: Accuracy of reproduction of control valve mass flow rates by means of the addition model for steam/water flow having low vapor content

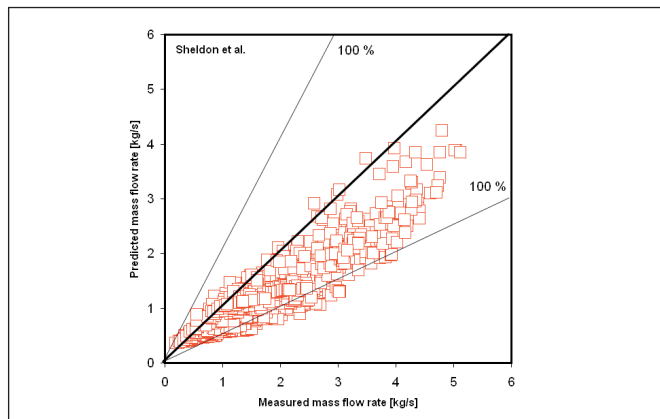


Fig. 3: Accuracy of reproduction of control valve mass flow rates by means of the Sheldon and Schuder for steam/water flow having low vapor content

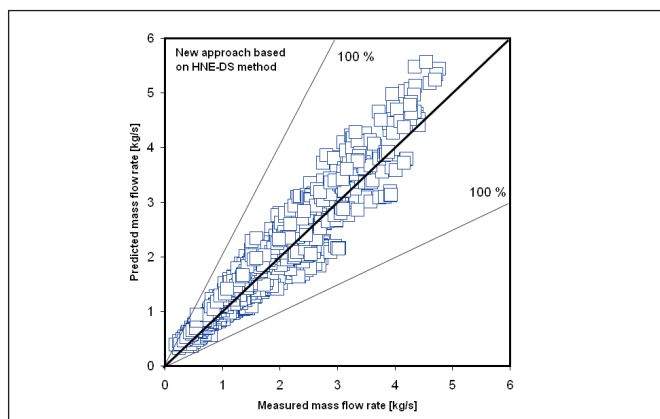


Fig. 4: Accuracy of reproduction of control valve mass flow rates by means of the new approach with the new expansion factor Y_{MP} based on the HNE-DS method with slip correction for steam/water flow having low vapor content

Literature:

- [1] Kiesbauer, J., Meffle, K.: Ein Leitfaden für eine vereinfachte Auslegung eines Stellgerätes auf der Basis von EN 60534 (An introduction to a simplified valve sizing based on the EN 60534 Standard), Automatisierungstechnische Praxis, Issue 8, 2001, Vol. 43, Oldenbourg Verlag, München
- [2] Diener, R., Friedel, L., Kiesbauer, J.: Auslegung von Stellgeräten bei Zweiphasenströmung (Sizing control valves for two-phase flow), Automatisierungstechnische Praxis, Issue 3, 2001, Vol. 42, Oldenbourg Verlag, München

- [3] Sheldon, C.W., Schuder, C.B.: Sizing control valves for liquid-gas-mixtures, Instruments & Control Systems, Vol. 38, January 1965
- [4] Heckle, M.: Zweiphasenströmung Gas/Flüssigkeit durch Drosselorgane. Ein neues Berechnungsverfahren der Zweiphasenströmung in Blenden, plötzlichen Verengungen und Ventilen (Two-phase gas/liquid flow through throttling devices), Fortschrittberichte der VDI-Zeitschriften, 1970
- [5] Diener, R.: Berechnung und Messung der Massendurchsatzcharakteristik von Stellventilen bei Zweiphasenströmung (Experimental and calculated control valve two-phase mass flow characteristic), Fortschrittberichte VDI-Reihe 7, no. 388, 2000
- [6] Diener, R., Schmidt, J.: Sizing of throttling device for gas/liquid two-phase flow. Part 1: Safety valves, Process Safety Progress 23 (2004) 4, 335-344.
- [7] Diener, R., Schmidt, J.: Sizing of throttling device for gas/liquid two-phase flow. Part 2: Control valves, orifices and nozzles, submitted to Process Safety Progress.

Dr.-Ing. Ralf Diener is senior plant manager of the ammonium carbonate plant in the division Inorganic Chemicals of BASF AG. His main fields of work include ammonium carbonate production, process support, and the further development of the products and the production plant. Previously he was involved with the development of calculation methods for safety-related problems.

Address: BASF Aktiengesellschaft, E-CAA/SM - Q404, 67056 Ludwigshafen, Germany
 Phone: +49 0621 60-55889, Fax: -73646,
 E-mail: ralf.diener@basf-ag.de



Dr.-Ing. Jörg Kiesbauer is Director of R&D at SAMSON AG MESS- UND REGELTECHNIK in Frankfurt/Main in Germany. His work experience includes R&D in the field of control valves equipped with electric and pneumatic accessories as well as self-operated regulators (flow and acoustical tests, development and optimization of calculation methods, development and testing of diagnosis tools for control valves etc., development of software tools). Since 1999, he has been involved in the IEC Working Group 65B-WG9 and in the DKE working group 963 as an expert.

Phone: +49 69 4009-1464
 E-mail: drjkiesbauer@samson.de



Dr.-Ing. Jürgen Schmidt works freelance in the Safety Engineering division at BASF AG as well as being active as a lecturer at the Technical University in Karlsruhe, Germany. For the past 11 years, he has worked at BASF AG in Ludwigshafen in the field of plant safety. He has been lecturing since 2002 in the subject of plant safety in the chemical industry and chairs the Dechema working committee for safety-related sizing of chemical plants. His key activities also include consulting companies in production process matters as well as the development of new methods in the field of safety engineering.

Address: BASF Aktiengesellschaft, GCT/S-L511, 67056 Ludwigshafen, Germany
 Phone: +49 621 60-56205,
 E-mail: juergen.schmidt@onlinehome.de





SAMSON AG · MESS- UND REGELTECHNIK · Weismüllerstraße 3 · 60314 Frankfurt am Main · Germany
Phone: +49 69 4009-0 · Fax: +49 69 4009-1507 · E-mail: samson@samson.de · Internet: <http://www.samson.de>