The optimization of the geometries and flow conditions of contactless gaskets

by

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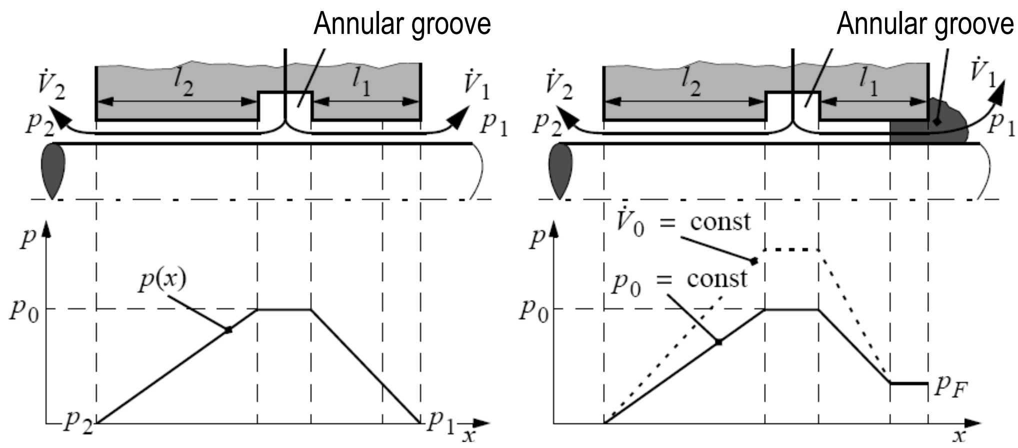
1 Introduction

The purpose of this paper is to demonstrate ways to optimise the geometries and the flow- conditions of a contact-less gasket, which is operated with a liquid media. One of the most important points is the geometry of the feeding-lines and the annular groove which conducts the sealing- media into the annular gap, which is in direct contact with the media which is to seal. In addition to the different measurements the current of the sealing- media is also essential. The flow of the sealing- media which flows in longitudinal direction of the annular gap toward the blocking area is also influenced by the relative movement of the shaft and housing. It will be shown that the different flow-effects and velocities of the sealing- stream are important to generate a homogeneous flow. The flow- conditions inside on the annular- gap will be described to get an optimised flow and optimised measurements of the gab. The better homogeneous the flow is the better is the seal-effect, respectively the generation of the barrier-layer which works against the infiltration of the media inside of the apparatus into the annular gap respectively the gasket-system. The argumentation based on the fundamental equations, to give the designers hints to control the construction regarding the points mentioned above.
The description is focused on the application inside of a vertical assembled mixer. Finally the text is focused on describing ways to optimise the measurements of the annular gap of the contact-less gasket to minimise the necessary sealing media flow. (17) The contents of the text can be separated into a part which describes the pressure loss inside gaps generally and two approaches to describe the flow conditions and of the corresponding calculation of the pressure drop. The approaches regarding the necessary sealing media velocity differ in the dilution of the content of the solid particles. The first approach describes a single particle the second approach describes the flow respectively the removing (fluidization) of a packed bed (solid deposits). By this separation the complete range of dilution, respectively number of solid bodies are displayed.

1.2 Description of the Pressure- conditions inside of the flushing-gap

In the diagrams below, the conditions regarding the pressure- profile inside of the annular gap is portrayed. In the middle of each diagram an annular- groove is show. This groove has the purpose to distribute the sealing-media as even as possible around the periphery around the shaft. In the area around the influx into the annular- groove a pressure (p_0) is generated, which caused the media flow through the gap into both directions. Affected by the flow the pressure drops down to the ambient pressures ($p_{1/2}$) at the ends of the both sides of the gap. When one side of the gap is charged by a media with a certain pressure, this pressure and impulse of the media effects an infiltration into the gap. The media infiltrates into the gap until the local pressure of the sealing- media equals the pressure of the media which is to seal. When the point of contact between both media is ahead the inlet- feeding-point (right side of the annular groove) the sealing- system is tight.



In the case of a further increasing media- pressure up to the blocking- pressure in the gap $p_{(x)}$ the media penetrates against the sealing- media as far as the annular- groove of the flow of the sealing-media would carry the other media to the leakage- side (left side of the annular groove). Then the blocking-system will be operated with a constant blocking- pressure p_0, in the case of gaseous media, the characteristics of the pressure-curves are not influenced by the admission of the media which is to seal. When the blocking- system will be operated with a constant blocking- volume- flow V_0, the blocking-pressure p_0 would rise, because the gap is partly blocked by the media. This partly blocking caused an additional pressure on the right side of the diagrams above and consequently effects an increase of the blocking- pressure p_0. The operation of the system with the volume- flow as control- variable caused a self-energizing sealing-effect. The outlined sealing system has some advantages and disadvantages in comparison to labyrinth- sealing – systems.

The advantages are:

The system is independent of the position, because the blocking- method does not generate an influx which has to be absorbed and leads back with the help of gravity.

The construction- volume in radial direction is small, because a chamber for separation is not necessary.

The system is insensitive in respect of fouling, because the system does not contain e.g. runback-canals which could be blocked by deposits.

The sealing- effect can be adjusted by the blocking- pressure p_0 in the annular- groove as function of the admission of the pressure p_F.

The disadvantages are:

In the case of a break-down of the blocking- flow, the only sealing effect bases on the gab, because it works as a throttle. Thus a permanent monitoring- system of the blocking-flow or pressure is necessary.

The leakage- flow V_2 which flows into the left direction in the diagram has to be reduced by the design. One interesting possibility is the application of e.g. PTFE- lips.

The blocking – flow- system needs little measurement of the gap, because the pressure- drop which is generated by the longitudinal flow through the gap is important for the pressure- generation in the annular groove and for the distribution (which will be described in a later paper) of the flow of the sealing- media. The disadvantage of such a little gap is the possibility that solid deposits can cause abrasion and the destruction of the system.

The costs for the supply of the additional media- flow and the generation of the necessary pressure-gradient are also negative points.

The essential advantage off the blocking- flow- system is the simplicity of the design the principle and the experiences which are available.

1.3 Pressure loss

The pressure loss is the basic to do statements about the flows and the effectiveness of the system.
The calculation of the pressure loss is been given by the following equation:

$$\Delta p = \lambda \frac{l}{D_h} \frac{\rho}{2} c^2$$

For the calculation of the pressure loss and consequently for the calculation of the volume flow which streams through the gap, the equivalent diameter (D_h), length (l) and the friction factor (λ) are essential. These factors describe the duct and therefore the basis to do studies concerning the influence of the geometry of the gap on the sealing- media- flow. In the case of a laminar flow the friction factor (λ) is been given as a function of the Reynolds number (Re) and a factor C which depends on the shape of the channel. (The restriction to describe streams inside of ducts up to a diameter of 1 [mm] makes it highly probably that the stream of the media has the stage of the laminar flow.)(3)

$$\lambda = \frac{C}{\mathrm{Re}}$$

In the case of a circular pipe the factor has the constant value of 64, in the case of e.g. a rectangular shaped pipe, scientists have also developed values which were also constant, but different from 64.
A lot of investigations have been made in the case of micro channels with the principle result, that the rules which are in use for macro channels are also applicable in the case of micro channels. (7)
The C- factor in the case of gaps varies with the quotient of the inner- diameter of the gap (D_i) and the outer diameter (D_a) in the case of a narrow gap the C- factor tends to the constant value of 96. (12)

1.3.1 Flow through micro ducts

Some investigations have been made by researchers with different results concerning the transition of the media flow from the laminar flow to the turbulent flow. The investigations on water in micro-channels with hydraulic diameters ranging from 50 µm up to 300 µm and Reynolds- numbers ranging from 50 up to 1500 have shown that the common equations are applicable down to a hydraulic diameter of 100 µm. This range covers the common range of sealing- gaps respectively the equal- diameter, which is between 60 µm up to 200 µm partly. (9) Rough pipes are characterized by a lower critical Reynolds-number than smooth pipes. Roughness (k) give additional disturbances in the laminar flow thus the traditional calculation of the critical Reynolds- number for circular pipes has to be modified. In the case of a relative roughness of $\frac{k}{D} = \varepsilon_k \leq 0{,}7$ % it is possible to calculate the transition area with two equations.

$$\mathrm{Re}_1 = 1160 \left(\frac{1}{\frac{\varepsilon_k}{D}} \right)^{0.11}$$

and

$$\mathrm{Re}_2 = 2090 \left(\frac{1}{\frac{\varepsilon_k}{D}} \right)^{0.0635}$$

Other authors mentioned that this effect is caused by the presence of initial turbulences or disturbances at the flow channel for a Reynolds- number up from a value of approx. 1200. Because of these influences the critical Reynolds-number for circular pipes has been modified as follows.

$$\mathrm{Re}_0 = 754 \exp\left(\frac{0.0065}{\frac{\varepsilon_k}{D}} \right)$$

Experiments generally indicate that the laminar flow regime friction factor is in a good agreement with the Hagen- Poiseuille theory as fare as the Reynolds- number is below approx. 600. For higher values of the Reynolds- number the friction factor is higher. The transition from the laminar to the turbulent regime occurs for Reynolds- numbers in the range of 1900 up to 2500. This transition is in a good agreement with the flow transition in rough commercial respectively macro tubes. (4) Other experiments with stainless steel tubes, characterized by a higher relative roughness have shown an increase of the factor 64 of approx. 15 % for the calculation of the friction number in the laminar stage. Other measurements have shown that the influence of the roughness can increase the pressure drop, in comparison to smooth channels up to 40 %. These measurements indicate the importance of taking the roughness and the ratio between roughness and width of the gap into consideration. (5, 6, 17)

1.3.2 Flow through annular gaps

The flows inside of annular gaps differ from circular tubes. Because the periphery of the duct is not symmetrical to the flow field, thus the maximum velocity is not on the symmetry-line like in pipes. The maxima – in the case of the laminar as well as in the case of the turbulent state- is shifted to the outer diameter of the shaft. But this offset is not crucial in the case of narrow annular gabs, thus the influence can be disregarded.(17) And the distribution of the flow can be accounted as a flow which is symmetric regarding the mean diameter in the middle of the gap. (11)

These results suit to the definition of the Re- number

$$\mathrm{Re} = \frac{\rho\, c\, D_h}{\eta}$$

and the linear proportional influence of the diameter respectively the equivalent diameter (D_h). The equivalent diameter is -as generally known- defined as

$$D_h = \frac{4\,A}{U}$$

with A as cross-section and U as the periphery. In the case of an annular gap the equivalent diameter is defined as follows:

$$D_h = 2s$$

For the consideration of an non- circular ducts like the annular gap or any other geometry, it is common to use correction factors (φ) to take the discrepancies to the circular cross-section into consideration.

In that case it is possible to use the following equation to calculate the friction factor for the laminar-stage:

$$\lambda = \varphi \frac{64}{\mathrm{Re}}$$

The pressure- drop characteristics of a fully developed duct flow are normally given as a function of the relation between the inner- diameter (Di) and the outer diameter (Da) and leads to the form- factor φ in the case that

$$D_i \cong D_a,\ \varphi = 1{,}5 \quad \text{(3, 17).}$$

1.3.3 Pressure loses generated by the inlet and outlet of gap

The shapes of the inlet- and outlet areas of the gap have also to be taken into consideration, especially when the length of the duct is short. The flow of the media in these areas are characterised by a deflection influenced by the shape of the duct. The profile of the flow is influenced, up to a length like the following equation shows.

$$la \approx 0{,}0575\,\mathrm{Re}\,D$$

After reaching la the flow changed to the normal laminar flow without any disturbances. The disturbances generate additional losses (impetus losses) which are composed into the drag coefficient ζ. (7) In the case of the laminar flow that factor is different to the turbulent flow. The inlet drag factor ζ_I in the case of sharp-edged inlet is between 1 up to 1,2, instead of approximately 0,5 in the turbulent case. The outlet generates a drag factor ζ_O of 1, which is caused by the losses of the kinetic energy (6). By the consideration of the inlet- and outlet losses the calculation of the pressure loss has to be extended as shown.

$$\Delta p = \left\{ \lambda \cdot \frac{l}{D} + \zeta_I + \zeta_O \right\} \frac{c^2 \cdot \rho}{2} = \left\{ \zeta_D + \zeta_I + \zeta_O \right\} \frac{c^2 \cdot \rho}{2}$$

The hydrostatic pressure

It is necessary to take the height h, respectively the pressure which is caused by the weight (water-column) of the contents of the e.g. mixer into concideration. This pressure - the so called hydrostatic pressure- can be calculated as follows:

$$p_{Hydrostatic} = g\ \rho_{Water}\ h$$

The pressure of the atmosphere is not important- except a pressure inside of the vessel- because the difference- pressure is the only interesting one, because the pressure at the surface of the e.g. water column is approximately the same as the pressure around the pump.
The hydrostatic pressure has to be added to the pressure drop to get the complete pressure at the gap.

$$\Delta p_{complete} = \Delta p + p_{hydrostatic}$$

In the case of only a low amount of solid parcels the weight of these ingredients are not important. In a later part of the text, another approach takes these ingredients also into consideration regarding the pressure and the pressure-loss.

1.4 Calculation of the sealing media flow

By the use of the extended version of the equation, for the calculation of the pressure losses it is possible to calculate the velocity of the media flow like shown below.

$$c = \sqrt{\frac{\Delta p}{\left(\lambda \cdot \frac{l}{D} + \zeta_I + \zeta_O \right) \frac{\rho}{2} + g\, \rho\, h}}$$

And with the equation of continuity the calculation of the volume flow as follows.

$$V = c\ A$$

This form does not take the different influences as mentioned into consideration. The friction factor λ has to be modified because of the impact of the roughness. To do that the roughness factor f_R has been introduced. This factor based on experiments and represents the increase of the influence of the surface in the case of micro canals.

$$\lambda = f_R \frac{96}{\mathrm{Re}}$$

The roughness factor can be expected in a range from $f_R = 1{,}15$ up to a value of $f_{R\,Max.} = 1{,}40$.

1.5 Taylor- vortices the axial- and radial Reynolds number

The system shaft- housing principally represents two concentric arranged cylinders with a relative movement to each other. In the case of a rotating inner cylinder the generation of Taylor- vortices in the gap between both cylinders can happen. The centrifugal- force leads to a movement of the media from the inner- area towards the outer-area of the gap, the viscosity counteracts to that movement. After the excess of a critical revolution the laminar flow changed to a flow with secondary vortices.
The change to another state of the flow will commonly be described with the Taylor- number.
The Taylor- number is defined for this problem by Chandrasekhar as:

$$T = \frac{2\,R_1^{\,2}\,d^4}{R_2^{\,2} - R_1^{\,2}}\left(\frac{\Omega_1}{\nu}\right)^2$$

Ω represents the angular velocity of the inner cylinder. The critical Taylor- number

$$T_c = 1708 \; or \; \sqrt{1708} = 41{,}3$$

An additional way to calculate respectively to estimates the change of the state of the flow is possible with a modified Reynolds- number. Then the circumference Reynolds- number exceeds the critical Reynolds- number the flow is characterised by the appearing of single – or twin- vortices.

$$d = r_a - r_i$$

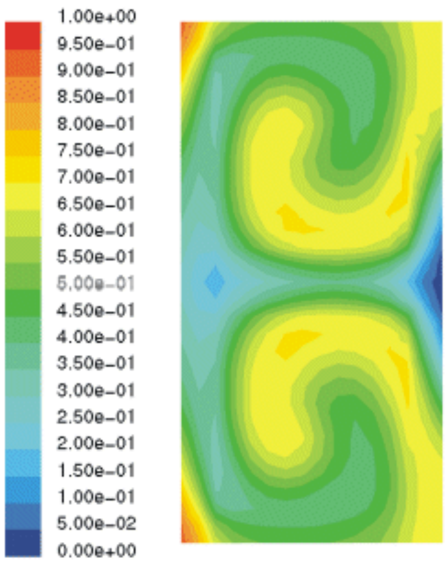
$$\mathrm{Re}' = \frac{\Omega\, r_i\, d}{\nu}$$

The critical circumference Reynolds number can be calculated with the following approximation:

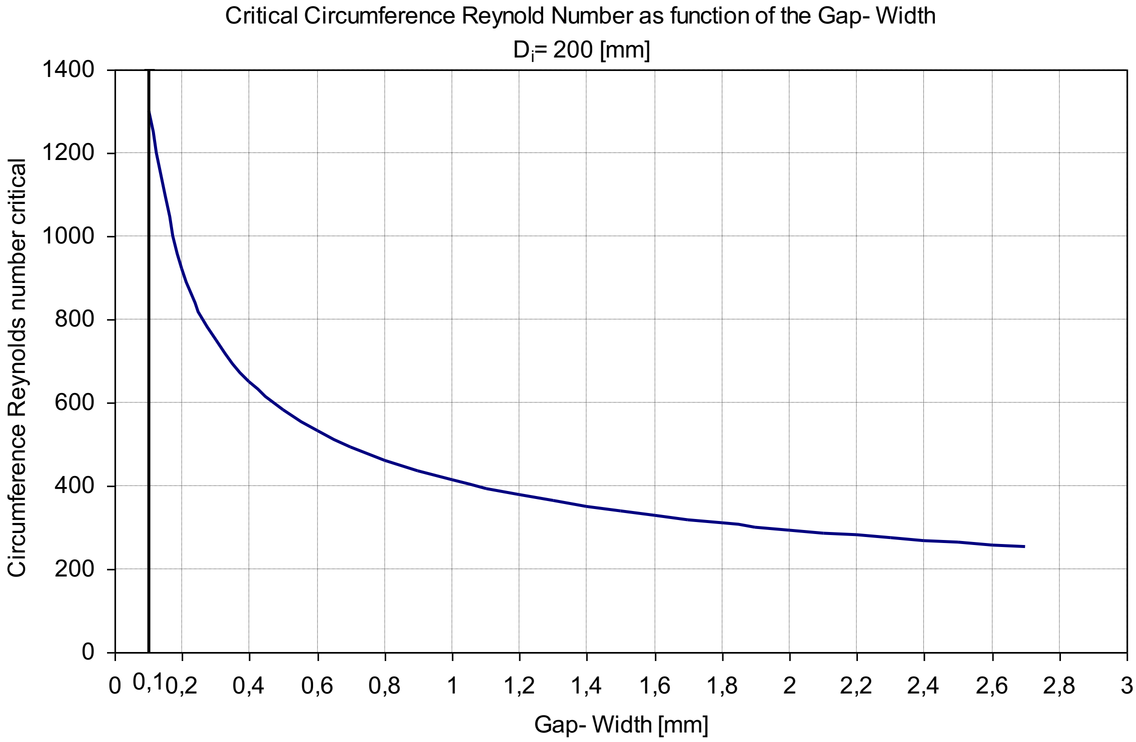
$$\mathrm{Re}_{critical}{}' = \frac{\Omega\, r_i\, d}{\nu} = \left(41{,}1 + 14{,}5\,\frac{d}{r_1}\right)\left(\frac{r_1 + r_2}{2\,d}\right)^{0{,}5}$$

with

$$d = r_a - r_i$$



The interesting point is the term on the right side, because this term contains only geometrical data. Following on these dependences it is possible to describe the influence of the width of the gap regarding the appearance of the critical circumference Reynolds- number. The following diagram shows this influence.



The diagram also shows the a gap- width of 0,1 [mm], because this is a common measurement for the gap. By the use of the mentioned equation

$$\mathrm{Re}_{critical}{}' = \left(41{,}1 + 14{,}5\,\frac{d}{r_1}\right)\left(\frac{r_1 + r_2}{2\,d}\right)^{0,5} = 1300{,}48$$

, the necessary width can be defined to avoid the reaching and of course the exceeding of the critical circumference Reynolds number $(\mathrm{Re}_{critical}{}')$. Otherwise it is possible to take the angular velocity into consideration to optimise the measurement. (20) It is necessary to point out that these measurements do not take the pressure gradient or the surface condition into consideration. Following on that the result of the calculations can only be an approximation. The so called aspect ration

$$\Gamma = \frac{L}{d}$$

(L=length of the gap, d= width of the gap) has no effect on the appearance of the Taylor vortices.
The transition of an axial flow occurs when the axial Reynolds- number reached a value of approximately 2000. If the value of Ta = 1708 (41,3) is reached before the Reynolds number reaches it the critical value, then the transition is to vortex flow. If the axial Reynolds number, exceeds 2000 while the Taylor number is still less than 1708 (41,3) then the transition is directly to the turbulent flow. (23)
Other sources (24) give other magnitudes of the critical axial Reynolds number. One important effect is the roughness, because in that case the critical axial Reynolds number can drop down to below 600 (chapter 3). On the other hand the axial Reynolds number can rise up to 4000 in best circumstances. This big value can only be reached by very smooth surfaces of the gap. The following chart shows the different possible categories of the axial Reynolds number respectively the state of the flow

$$
\begin{aligned}
&Lamiar: && \mathrm{Re} \leq 500 - 2000 \\
&Transitional - area: && 2000 \leq \mathrm{Re} \leq 4000 \\
&Turbulent: && \mathrm{Re} \geq 4000
\end{aligned}
$$

(24)

1.5.1 The effects of the axial flow regarding the Taylor- vortices

The effect of super-positioning on the rotational flow and an axial flow has an additional effect. The axial

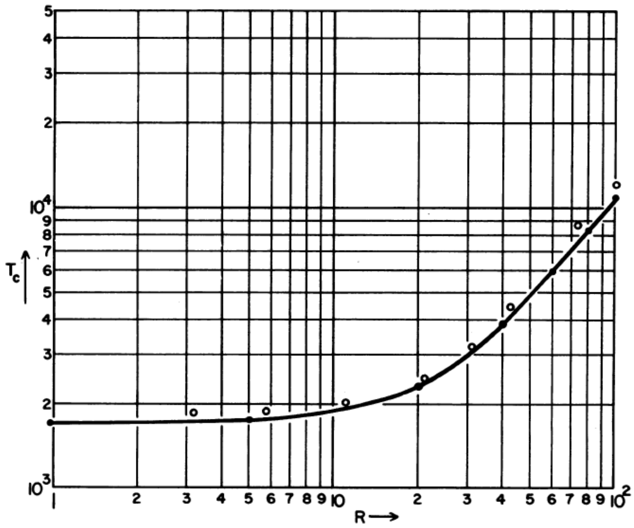
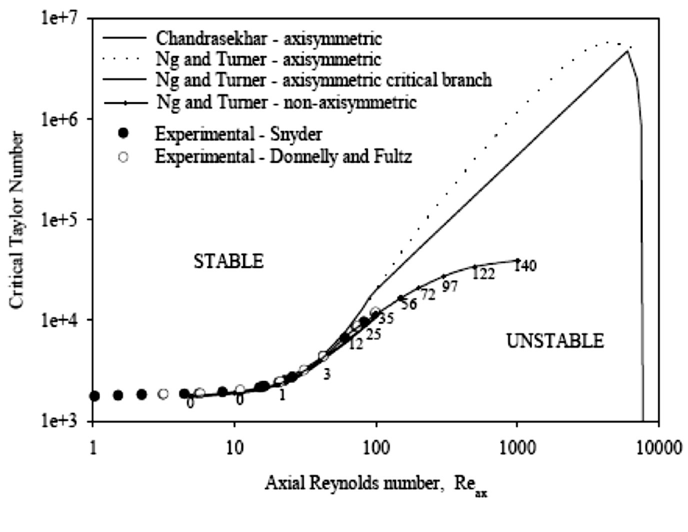


FIG. 1.—Plot of the critical Taylor number for rotation as a function of the Reynolds number for the axial flow. Solid line and solid circles are from Chandrasekhar's calculations. Open circles are experimental points.

flow caused a delay of the onset of the instability. The influence of the axial flow is shown in the diagram above. The critical Taylor- number increases as a function of the axial Reynolds- number. This Reynolds- number is calculated with the mean axial flow. Following on that, the avoiding of Taylor-vortices can be influenced be the volume respectively by the mass- flow of the sealing media. (19, 22)
The figure below shows the run of the ratio of the Taylor- number and of the axial Reynolds- number



(22)

up to a Reynolds- number of nearly 10000. The higher the axial Reynolds- number the higher is the critical Taylor number up to the mentioned range of approximately 10000.

It is generally possible to find 4 different flows:

The absolute laminar flow during a broad range of axial velocities (that range is characterised by the Reynolds number) and a restricted range of the peripheral velocities of the shaft (that range is characterised by the Taylor- number).

A laminar flow with Taylor- vortices in a range with low values of Reynolds numbers up to an area with height Taylor- numbers.

An absolute turbulent state mainly based on height values of the Reynolds- number.

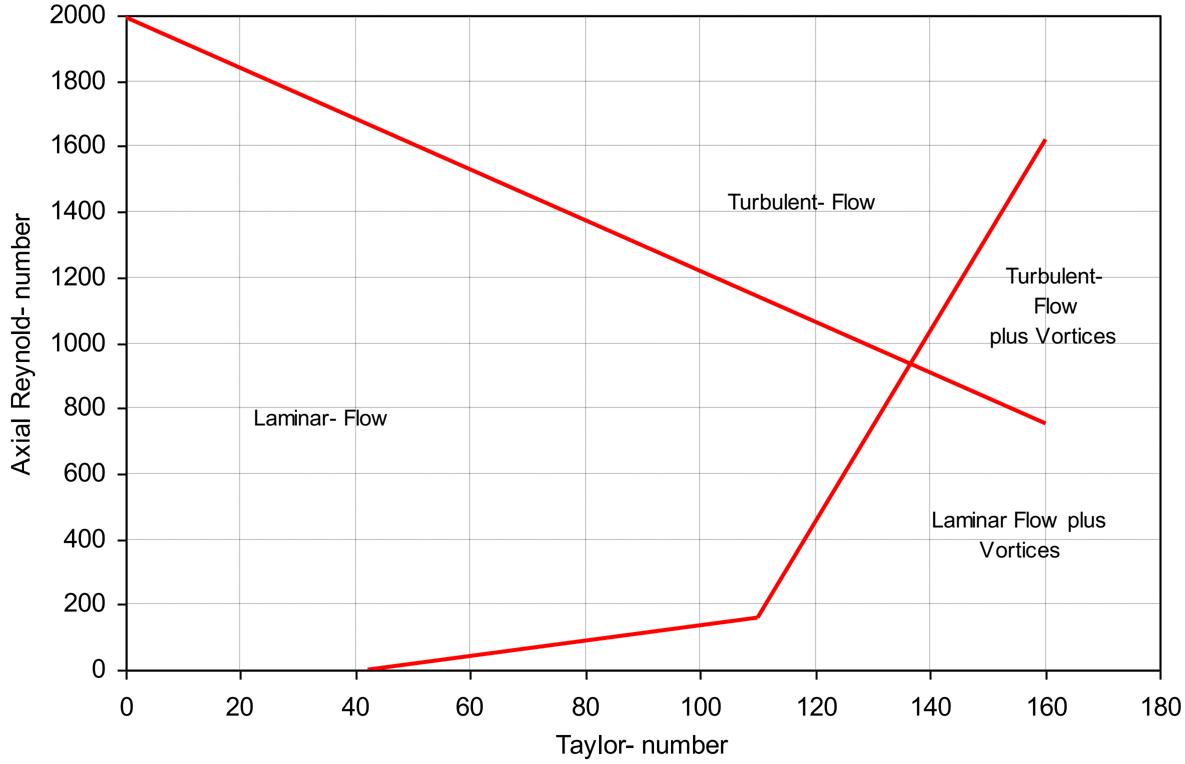
And an absolute tubular state with Taylor- vortices: This state is characterised by big Reynolds numbers and great Taylor numbers. (25)

Focused on the rotation of the inner cylinder, it is axiomatically possible to say that the turbulences respectively the turbulence character of the flow increases depending on the rotation of the inner cylinder.
The following diagram shows the experimental results of flow studies for a relatively wide gap.

$$\left(\frac{b}{r_m} = 0{,}037\right)$$

It visualises the four separate regions of flows. The measured value of the critical axial Reynolds number for zero speed of rotation and for the transition from the laminar to the turbulent flow is 2000 and agreed with other similar observations.

The different regions of flows in gaps
as function of the Taylor and Reynolds- number



It is necessary to quote the used modified Taylor number, to make the application of the showed diagram possible:

$$T = \Omega \, r_m^{\,0,5} \frac{b^{\frac{3}{2}}}{\upsilon}$$

$$b = r_0 - r_i$$

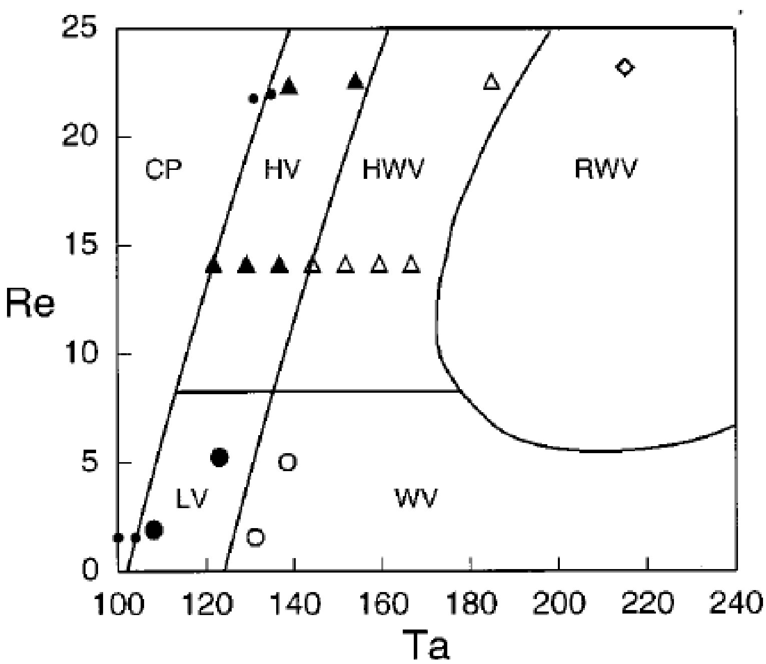
$$r_m : mean\ raduis\ of\ the\ gap$$

$$\upsilon : kinematic\ velocity$$

Some effects haven not been taken into consideration or were not known:

The phase of the development of the media-flow.
The roughness respectively the smoothness of the surfaces.
The axial oscillation and radial oscillation of the inner cylinder or the outer cylinder.

Investigators have recognised that then the rotation is gradually reduced, the Taylor vortices persist down to lower speeds than the calculated critical speed shows. This phenomenon is similar to the persistence of turbulences for the media flow in round pipes. Other investigators found that the Taylor vortices can exist at velocities which are several hundred times the critical speed found by the use of the equation by Taylor. The shown diagram represents the general run of the different curves of the different states of flow. Publications which are later published have shown a similar curve progression, but often in a very little range regarding the Taylor- and the axial Reynolds number like shown below.
The laminar region is limited by a maximal Reynolds number of 25 and a maximal Taylor number of 140.

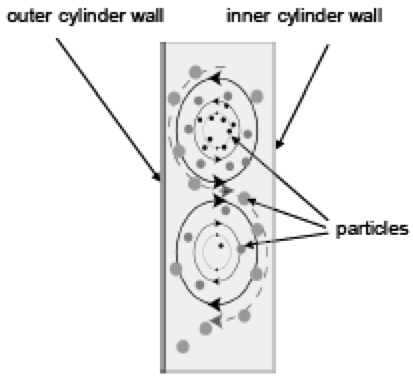


(26)

The use of the diagram in association with the modified Taylor number makes the finding of the flow conditions possible. The knowledge of the flow condition is essential which will be shown in a later chapter. (25)

1.5.2 The effects of the Taylor- vortices regarding particles

The figure below shows the mechanism inside of the Taylor- vortices in respect of solid bodies which are inside of the gap. The centrifugal forces causes the endurance movement of particles due to the rotation of the inner cylinder concentrates particles near the wall of the outer cylinder and the larger particles tend to be located more outside in the vortex- cells. Owing to the effect of the bypass- flow, the larger particles disperse more quickly in the axial direction. (21)



(21)

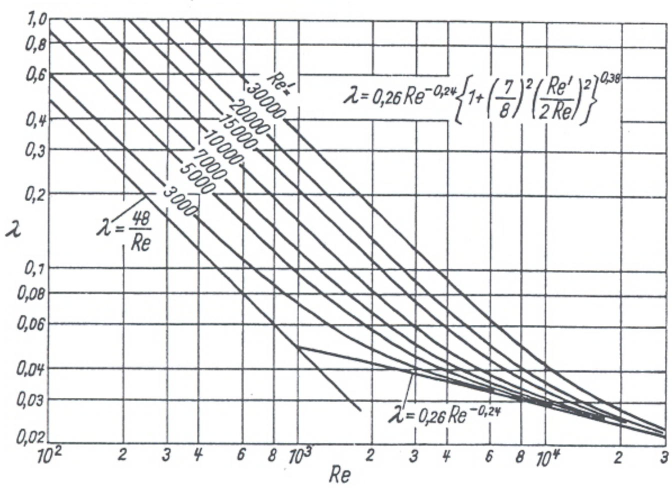
Furthermore some bypass- flows can counteract to the prevailing axial flow (22)

1.6 The effects of the axial flow regarding the friction factor

The superposition of the axial- and radial flow does not influence the friction factor λ when the flow state is laminar. (16) Based on the mentioned it is interesting to get a formula which combines the influences of the axial Reynolds number (Re) and the circumference Reynolds number (Re').
Mr. Yamada developed that –also based on the theoretical part of his work – in the case of a laminar axial flow the friction factor (λ) is unchanged when the rotation is characterised by the circumference Reynolds number and a value of approximately 3000 doesn't exceed. After reaching this value of Re' the value of the friction factor (λ) rises strongly. Based on the theoretical and practical researches he develops the following formula for the tubular stage:

$$\lambda = 0{,}26\ \mathrm{Re}^{-0{,}24}\left[1+\left(\frac{7}{8}\right)^2\left(\frac{\mathrm{Re}'}{2\ \mathrm{Re}}\right)^2\right]^{0{,}38}$$

From a value of $\mathrm{Re}' \leq 10000$ the value of the friction factor (λ) is nearly independent from the geometrical conditions. The following diagram shows the run of the curves which based on the quoted equation.



(17)

2 Determination of the necessary velocity of the sealing media (the fist approach)

The maximal velocity is limited by the appearance of abrasion by the solid bodies. The effect of abrasion increases with the 3rd up to the 4th exponent of the velocity. (13)
Based on that, the knowing of the necessary velocity which is needed to keep media or in this cases particle from infiltrating the gap is fundamental.
The calculations imply the computation of the flow resistance of particles. The equation of the force which is generated by the flow resistance shows the basic phenomenon.

$$F_W = \zeta_W \frac{\rho}{2} c^2 \; A$$

The force depends on the drag- coefficient ζ_W, the spec. weight ρ of the particle, the velocity of the particle c and A the cross-section vertical to the direction of the movement.
The drag- coefficient depends on the Reynolds number of the flow and the shape of the particles.
To determine of the drag- coefficient of the particles the shape of the solid bodies are considered to be spherical.
In the case of a no- spherical shape an equivalent diameter, the so called "Sauterdurchmesser" can be calculated as follows: (18)

$$D_{ST} = \frac{6\,V}{A}$$

In the case of spherical solid bodies the diameter equals the equivalent diameter.
Some formulas are quoted to calculate the drag- coefficient of spheres, the formula of Mr. Martin represents the characteristics in the range of $0{,}2 \le \mathrm{Re} \le 10^4$ with the particle Reynolds number

$$\mathrm{Re}_p = \frac{\rho\; c\; D_{ST}}{\eta}. \text{ (18)}$$

$$\zeta_W = \frac{1}{3}\left(\sqrt{\frac{72}{\mathrm{Re}_p}} + 1\right)^2$$

But based on the non- uniform curve of the drag- coefficient it is appropriate to apply different formulas for the several ranges of the Reynolds number.
Range $0 \le \mathrm{Re} \le 10$

$$\zeta_W = \frac{24}{\mathrm{Re}_p} + 2$$

Maximal relative error $\pm 4\%$

Range $0 \le \mathrm{Re} \le 100$

$$\zeta_W = \frac{24}{\mathrm{Re}_p} + 1$$

Maximal relative error +14 % up to -20 %

Range $0 \le \mathrm{Re} \le 10^5$

$$\zeta_W = \frac{24}{\mathrm{Re}_p} + \frac{1}{2}$$

Maximal relative error +32 % up to -40 %

Range $0 \le \mathrm{Re} \le 10^3$

$$\zeta_W = \frac{24}{\mathrm{Re}_p} + \frac{4}{\mathrm{Re}_p^{\frac{1}{3}}}$$

Maximal relative error +7 % up to -10 %

The value of the drag factor drops with an increasing Reynolds number, thus the flow resistance is generally stronger in the stage of a laminar flow. Not until reaching a Reynolds number greater than 10^6 the factor is constant $\zeta_W \approx 0{,}44$. (14, 15)

Consequently, the laminar flow generates the best flow conditions to transport solid bodies.

2.1 Determination of the necessary velocity of the sealing media to move spherical particles

Causing by the sealing media flow in the area of the outlet of the gap an entrainment of the particles - which are in the media - occurs. The movement of the media and the particles differs in the form of a relative velocity. The following formulas do not take the accumulation of particles into consideration. In the case of an accumulation in the exit or around it (seen from the movement of the sealing media) the approach similar to fluidised- bed can be used. This description based on the assumption of a low concentration of solid materials and a low amount of number of solid materials.

The assumption based on the aim to avoid the accumulation from solid materials by an adequate velocity of the sealing media flow. Because the depositing of solid materials would effects the damaging of the sealing system.

The method to calculate the necessary velocity of the sealing media based on the calculation of the sink-rate of spherical solid bodies. The velocity of the particle will be put on a level with 0, thus the mentioned relative- velocity equals 0. The velocity of the sealing media has to be the same like the sink-rate of the solid particles.

$$c_0 = c_f = \left[\frac{4}{3} \frac{(\rho_P - \rho_F)}{\rho_F} \frac{g\, D_{ST}}{\zeta_W} \right]^{0,5}$$

With the insertion of the calculation from Martin the formula can be written as follows:

$$c_0 = \left[\frac{4}{3} \frac{(\rho_P - \rho_F)}{\rho_F} \frac{g\, D_{ST}}{\left(\frac{1}{3} \left(\sqrt{\frac{72}{\mathrm{Re}_p}} + 1 \right)^2 \right)} \right]^{0,5}$$

The equation can be used in the wide range in respect of the Reynolds number ($0{,}2 \le \mathrm{Re}_p \le 10^4$) and consequently in a wide range of the velocity of the sealing media.

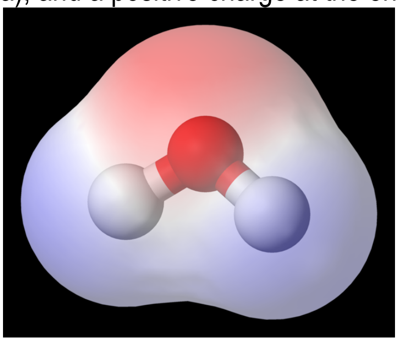
Because of the interdependency of the velocity $c_f ; c_0$ and the Reynolds number it is necessary to solve the equation iteratively.

The effectiveness of this approach is, like already mentioned dependants on the stage of the flow. Another considerable point is the movement of the particles regarding the direction of the media flow. Because the solid bodies tend to move in that way that only the minimal cross- section – of the particles – will be blow by the media flow. This leads to a higher value of the necessary velocity. After the beginning of the lifting a relative velocity will be generated, thus the calculation shows a too low amount of the velocity. Another flow- phenomenon is the “Magnus- effect”. This effect based on the rotation of the solid bodies the rotation causes an “additional” relative velocity. That “additional” velocity affects a lighter remove of the particles and assists the main flow.

2.2 Influencing-variables

The real flux can differ significant from the calculated flux. For a precise investigation the following influences- variables and losses also should be taken into consideration:

1. A change of the flow- profile direct in front of the gap.
2. Acceleration- losses, which are generated by the reaching of a homogeneous velocity- profile near the entry into the gap.
3. The contraction of the flow (contraction-losses)
4. The approach-section which is necessary to generate the velocity- profile.
5. The eccentricity of the gap.
6. The movement of one or both limitation- surfaces.
7. The wall-roughness
8. The losses of the kinetic- energy after leaving the gap.
9. The change of the flow- profile after leaving the gap.
10. The change of the viscosity caused by decompression (gaseous media).
11. The change of the viscosity caused by the increase of the temperature based on the conversion of energy (friction losses)
12. An alteration of the gap- measurement caused by an alteration of the temperature and an alteration of the pressure.
13. A blocking caused by polarised molecules.
    A commonly-used example of a polar compound is water (H2O). The electrons of the hydrogen atoms of water are strongly attracted to the oxygen atom, and are actually closer to oxygen's nucleus than to the hydrogen nuclei; thus, water has a relatively strong negative charge in the middle (red area), and a positive charge at the ends (blue area).



14. An alteration of the density caused by a change of the pressure. (17)

2.3 The influence of the eccentric position of the shaft.

Another important point regarding the design is the influence of the eccentric position of the shaft caused by tolerances. In the case of the laminar flow the influence of the eccentric position can be described as follows:

$$f_V = 1 + \frac{3}{2}\frac{e}{h}$$

The alphabetic character e represents the space of both centres; *h* represents the width of the gap in the concentric arrangement. The factor f_V tends to a minimum in the case of concentric position. In the case of a eccentric position the flow respectively the factor tends to $f_V = 2{,}5$.

Thus the volume- flow tends to 2,5 times of the theoretical flow. The formula is applicable up to a range of $\frac{r_i}{r_a} \leq 0{,}5$. Following on that the equation is applicable in the range of gaps which are applied to seal.

The influence of the concentric positioning is the strongest influence-able influence by the design next to the roughness. (25) Therefore it is essential to take this influence into consideration to reach a minimal mass- respectively volume- flow of the sealing media.

3 The second approach

Another approach to describe the removal of particles out of the area of the gap, respectively the outlet of the sealing media, is to look upon single solid bodies as a packed-bed (PB).

The analyses in this way can be seen as the maximal possible or in other word as worst case conditions.

The crucial fact to describe the processes in this way is that this description is nearer to the common conditions then the way to describe the removal for only one single particle.

Similar to the other approaches to calculate the pressure- loos in pipes, it is possible to use identification- numbers like the Reynolds number to describe the pressure- loss and the flow- conditions inside of fills. The problem for the calculation of the pressure drop inside of fills is the number of influencing variables:

1. Velocity
2. Measurements of particles
3. Shape
4. Arrangement of particles
5. Physical characteristics of the media

A big problem is the consideration of the laying of the particles, *for the reason that* the positioning has a direct impact to the porosity (Which will be described later). Because that has a direct influence on the pressure- drop. (32)

The complexity of the flow condition can be also described by the listing of the appearing forces:

1. Mass- inertia
2. Pressure- forces
3. Gravitational forces
4. Toughness- forces

The following chapters display a way to calculate the maximal pressure drop and the minimal necessary velocity of the sealing media to avoid the infiltration of solid bodies (PB) into the gap. (33)

3.1 The Packed bed

A packed bed is a fixed layer of small particles or objects arranged in a vessel to promote intimate contact between gases, vapours, liquids, solids, or various combinations thereof; used in catalysis, ion exchange, sand filtration, distillation, absorption, and mixing.
In this case the packed bed represents an accumulation of solid partials respectively a very high amount of solid particles inside of a liquid inside of e.g. a mixer.

3.2 The Fluidization

The processing technique of fluidization has the purpose to generate a suspension or fluidization of small solid bodies in a vertically rising stream of fluid.
The fluidization principle can be described as follows:
A passing fluid flows upwards through a packed bed of solids particles and generates a pressure drop due to the fluid-drag. When the fluid-drag force is equal to the bed weight –this is reached by a continuously growing media- flow- the particles no longer rest on each other, this is the point of fluidization. This velocity is known as the “minimum fluidization velocity”. If the fluid velocity increases further the pressure drop does not significantly increase – it remains equal to the bed weight, but the bed may expand. In this stage the packed bed is weightless, caused by the equilibrium of forces between the weight- force of the bed and the force generated by the media flow. Based on that, the behaviour of the bed respectively particles tend to the behaviour of a liquid. A further increase of the media velocity caused that the behaviour of the fluidized bed turns into to boiling liquid. The pressure drop is still constant during this stage of Fluidization. The volume of the fluidized bed rises gradually, caused by the increasing distances between the different particles. This stage is characterised by the generation of “media-bubbles”, which arise toward the surface of the fluidised-bed. At a certain velocity of the media flow the distances between the different solid bodies rises to a magnitude that the stage of transportation of particles out of the bed is reached. With the reaching of this velocity and consequently the discharge of material the maximal media velocity is reached. (27)

3.3 The Velocity inside and the porosity of fills (PB)

The velocity inside of fills is dependent on the porosity. This velocity can be calculated with the velocity in the clear cross- section c_{f0} and the porosity ε of the fill. The porosity represents the ratio of the vacuity of the fill V_c, the volume of all particles V_p and the total volume of the fill V_f. Thus the porosity represents the space between the particles caused by the clearance between the solid bodies.

$$\varepsilon = \frac{V_c}{V_f} = 1 - \frac{V_p}{V_f}$$

The value of the velocity inside of the fill can be calculated as follows:

$$c_p = \frac{c_0}{\varepsilon}$$

In the case of spherical particles the value of the porosity, in the state of motionless is approximately $\varepsilon \approx 0{,}44$.

3.4 The Internal surface of the fill

The value of the internal surface of the fill is necessary to do additional calculations. Normally the surface is important to calculate the mass- transfer inside of the fill, but in this case it is necessary to calculate the pressure drop.

$$a_{ph} = \frac{(1-\varepsilon)6}{D_{ST}}$$

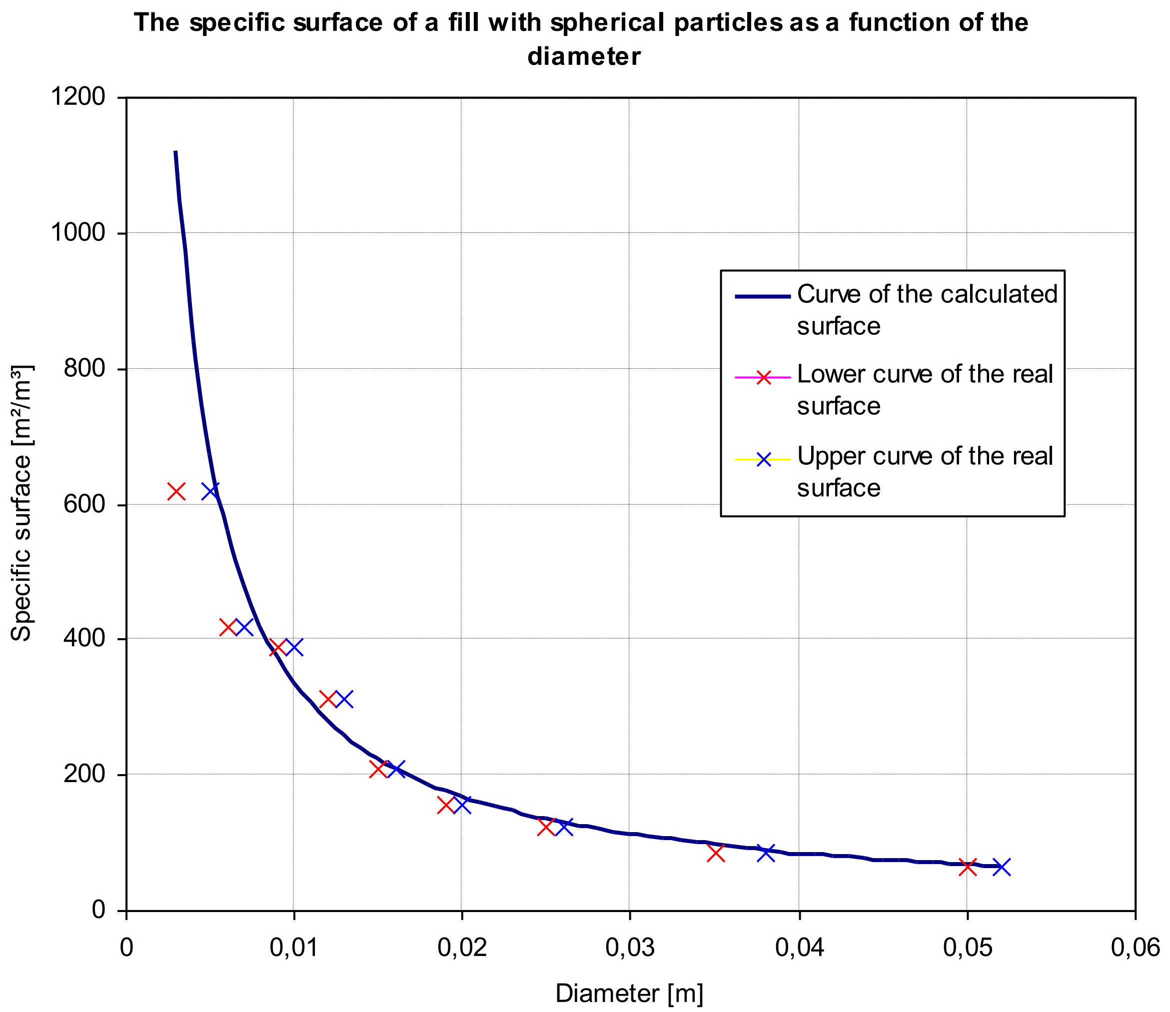
The result a_{ph} represents the specific surface of the fill in the SI- units m² per m³.

By the use of the value of $\varepsilon \approx 0{,}44$ it is possible to simplify the formula as follows:

$$a_{ph} = \frac{3{,}36}{D_{ST}}$$

The formula shows the influence of the diameter (equivalent diameter) on the inner surface.(29)

The following diagram displays the comparison between the calculated specific surface and the real surface of a sphere-fill. The diagram shows the good correlation between the real existing and the approximation. It is necessary to point out that the value of ε has been calculated with a constant figure of 0,44, because the real value fluctuates. (28)



3.5 The reaching of the state of a fluidized bed

The state of a fluidized bed is reached when the media velocity has caused a force which lifts the bed up. This relaxation velocity can be calculated as follows:

$$c_R = 7{,}19(1-\varepsilon_f)\frac{\eta}{\rho}a_{ph}\left\{\left(1+0{,}067\frac{\varepsilon_f^{\ 3}}{(1-\varepsilon_f)^2}\frac{(\rho_p-\rho)g\rho}{\eta^2 a_{ph}^{\ 3}}\right)^{\frac{1}{2}}-1\right\}$$

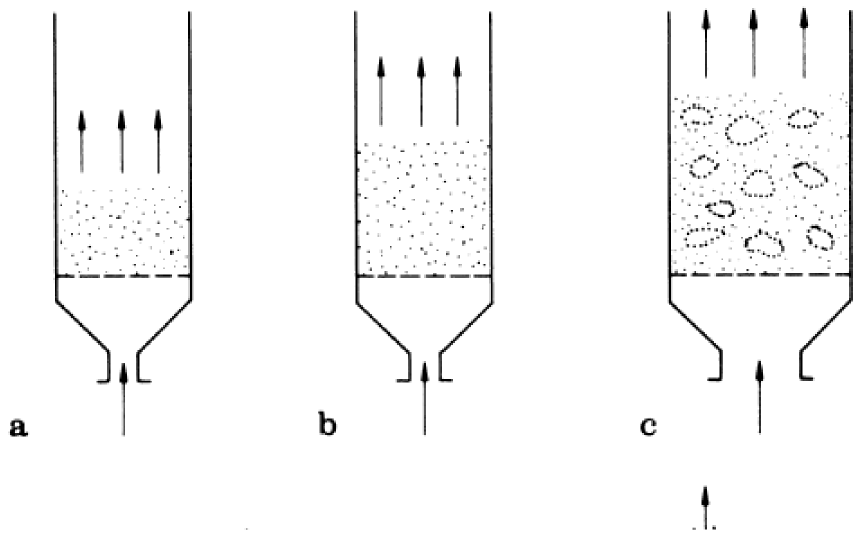
The important difference in comparison to the other formulas is the porosity ε_f , which represents the porosity at the beginning of the lifting the fill. It is generally possible to say the following regarding the measurements of particles:
The smaller the particles the larger the internal surface and the larger the pressure- drop. (31)
The result of the equation represents the lower limit of the necessary velocity, consequently the value has to be increased for the use in applications and therefore both values of the porosity are set on the same value $(\varepsilon_f = \varepsilon = 0{,}44)$. When additional information are available it is important to try to optimise the basic inputs regarding the mentioned equation and especially concerning the estimated porosity. The following estimation can optimise the precision of the mentioned formula. The porosity of a fill can be calculated within a range of $\mathrm{Re}_P = 10^{-2}...20$ with the following formula (27):

$$\varepsilon_f = 2{,}23\left(\frac{\mathrm{Re}_p}{Ar}\right)^{\frac{1}{4}}$$

The dimensionless Archimedes number can be interpreted as the ratio of lifting forces and the friction forces and is defined as follows:

$$Ar = \frac{D_{ST}^{\ 3}\Delta\rho}{\upsilon^2\rho}g$$

The variable υ represents the cinematic viscosity [m²/s] and $\Delta\rho$ represents the differences of the spec. weights $\Delta\rho = \rho_p - \rho$. The solution, respectively a result which is nearer to the reality can be reached with a few iterative steps. The first result of c_R has to be used to calculate ε and after this step the calculated porosity has to be taken to calculate the first corrected result of c_R. The picture on the next side shows the first three stages of the bed/ fluidized bed, the other stages are not interesting in this approach, because the media velocity has a bigger value when in the stage of fluidisation respectively lifting (b).



Stages of a fluidized bed are characterized by:

a	packed state
b	loosening state
c	bubbling state

3.6 The influence of rotation regarding fluidized beds

The rotation of the shaft influences the fluidisation of the bed like also the media-flow.
The increasing rotation of the bed affects an increase of the pressure- drop after reaching the state of fluidization. It has also been observed a generation of macro- circulation zones.
Sciences believe that the circulation causes concentration peaks regarding the media on the surface of the bed. The circulations can cause a back- mixing of the particles and thus an effect which is very important to avoid. (35)

3.7 The pressure drop in fluidized beds

The pressure drop can be calculated in the stationary state until the reaching of the loosing state of the bed with the following formula:

$$\Delta p = \Psi \left[\frac{(1-\varepsilon)}{\varepsilon^3} \right] c^2 \, \frac{\rho h}{D_{ST}}$$

The flow- resistance is heavily dependent from the properties of the fill. For the calculation of a homogeneous fill it is necessary to put the following factor into consideration:

$$\Psi = \left(\frac{150}{\mathrm{Re}_p} \right)(1-\varepsilon) + 1{,}75$$

And the particle Reynolds number:

$$\mathrm{Re}_p = \frac{\rho \, c \, D_{ST}}{\eta}$$

It is necessary to limit the value of h, because of the influence on the pressure drop and consequently on the pressure- drop based on the bed. The maximal value of the height is reached when the height of the bed equals the height of the e.g. water column above the gap. This hydrostatic pressure $p_{Hydrostatic} = g \, \rho_{Water} \, h$ has to be added to the pressure drop generated by the formula for the bed, less the displaced volume of the solid bodies (PB).

$$\Delta p_{complete} = \Delta p + p_{hydrostatic} \, . \text{ (30)}$$

4 Suggestions regarding the cooperation with customers

The dimensioning of the contact-less-sealing-system like the described is complex, effects by the height amount of influences. The multiplicity of influences makes a close corporation with the customer unavoidably. The physical properties of the particles are not the only important factors. The following composition represents the most important facts to reach an optimised system:

Medial size of solid bodies
- Maximal size of solid bodies
- Minimal size of solid bodies
- Spec. weight of particles

Porosity of the PB
Spec. surface of the PB
Height of the PB
Abrasion effect of the particles
Height of the water column
Pressure inside of the vessel
Measurement of the gap
- Length in longitudinal direction
- Width of the pap

Based on the classification by Geldart it is possible to separate the dissimilar solid bodies in four different classes. By the use of this classification the cooperation with the customer can by simplified.

Group A
The group A represents small particle sizes which are easy to reach the state of fluidization. The bodies have a strong tendency to absorption, little tendency to pass through, only a relatively low velocity of the sealing media is necessary to reach the state of fluidization.

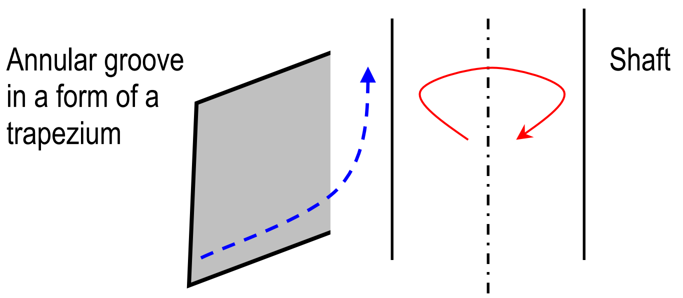
Group B
This group represents sandy particles which enable an easy pass through with only a little tendency to absorption. The group B represents solid bodies which need a low amount of velocity of the sealing media and consequently they are easy to reach the state of fluidization.

Group C
This group represents very small ($Ca(OH)_2$ 0,0025 …0,004 [mm]) particles with a cohesive tendency. These particles a have additionally electrostatic forces, thus it is very difficult to reach the state of fluidisation. An additional negative effect can occur in the form of bridge-building-effects and this can cause solid caking. One corrective technique can be the use of vibrations and the application of flexible elements made from e.g. elastic materials.

Group D
This group represents large sizes particles with a high density, which enables a very easy pass through and a high sink rate. This characteristic caused a high velocity of the sealing media and also the collision between the bodies, because the collusion affects a deceleration especially caused by the mass-inertia. (34)

5 Conclusions regarding the flow conditions

The basics of a media flow through a gap have been described to show the most importance influencing variables. The purpose has been to display the basic physical effects and demonstrates of the importance to take these influences into consideration. The pressure loss has been one of the important basics to do statements about the flows through gaps. It has been shown that the C- factor - in the case of gaps- tends to the constant value of 96. (12) Rough pipes are characterized by a lower critical Reynolds- number than smooth pipes. Roughness (k) gives additional disturbances in the laminar flow thus the traditional calculation of the critical Reynolds- number for circular pipes has to be modified. Experiments generally indicate that the laminar flow regime friction factor is in a good agreement with the Hagen- Poiseuille theory as fare as the Reynolds- number is below approx. 600 or based on other sources 500. Other measurements have shown that the influence of the roughness can increase the pressure drop, in comparison to smooth channels up to 40 %.



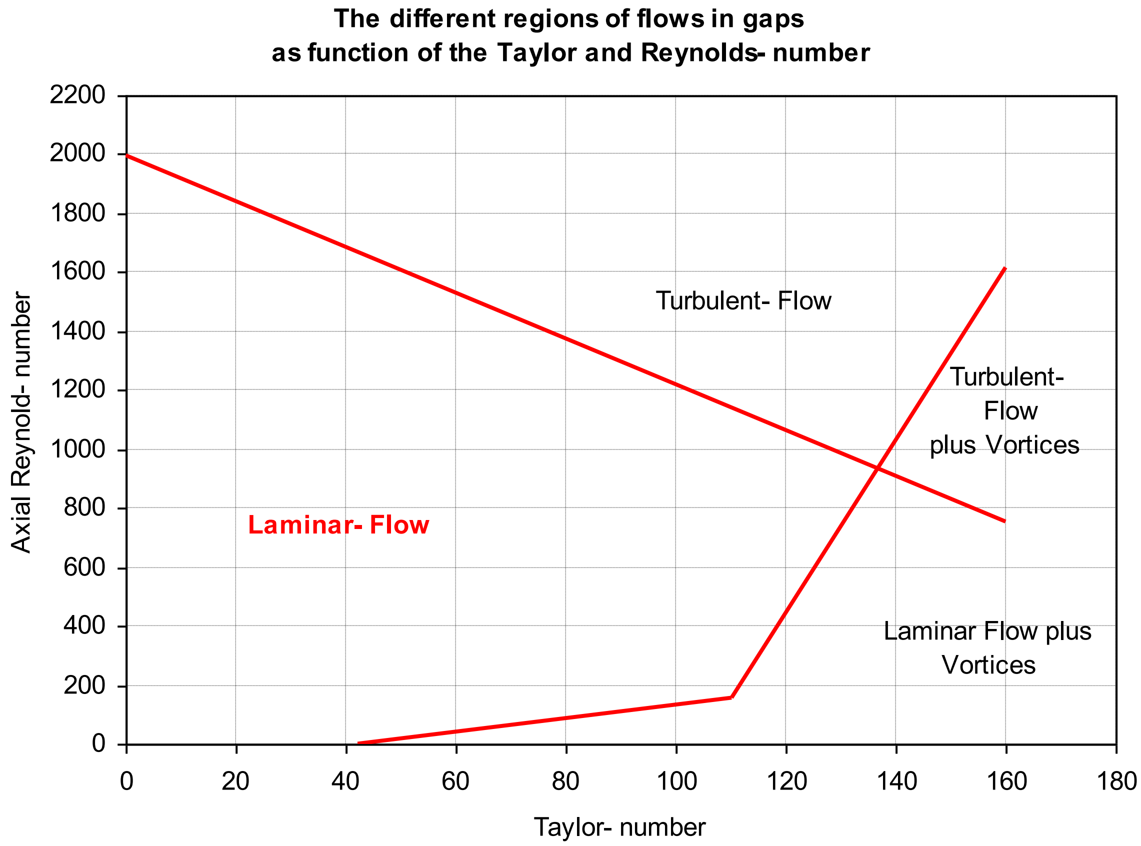
These measurements indicate the importance of taking the roughness and the ratio between roughness and width of the gap into consideration. (5, 6, 17) The surfaces between the gap have to be as smooth as possible to reduce the pressure loss and to move the generation of turbulences to higher regions in respect of the Reynolds number. The shapes of the inlet- and outlet areas of the gap have also to be taken into consideration, especially when the length of the duct is short. The flow of the media in these areas are characterised by a deflection influenced by the shape of the gap. In the case of the laminar flow that factor is different to the turbulent flow. The inlet drag factor ζ_I in the case of sharp-edged inlet is between 1 up to 1,2, instead of approximately 0,5 in the turbulent case. The outlet generates a drag factor ζ_O of 1, which is caused by the losses of the kinetic energy (6). Based on that, the inlet of the gap has to design with the aim to minimise the generation of turbulences. This can be reached with a rounded outlet-area of the annular groove. A further way for the reduction of the pressure- drops it to design the annular groove in a form of a trapezium (This is displayed on page 22). This shape generates an angle less than 90°in the area of the outlet of the groove. The other positive effect: The angle is bigger than 90° in the direction of the undesirable volume- flow V_2 (page 1), because this shape causes a bigger drag- factor. The avoiding of the volume-flow V_2 can be optimised by the use of e.g. additional PTFE- lips, because of the very low amount of the friction loss of these lips. It is possible, by the use of the modified formula of Taylor to identify the state of the flow- conditions by the use of the diagram which is shown below:

$$T = \Omega \, r_m^{0,5} \frac{b^{\frac{3}{2}}}{\upsilon}$$

$$b = r_0 - r_i$$

$$r_m : mean\ raduis\ of\ the\ gap$$

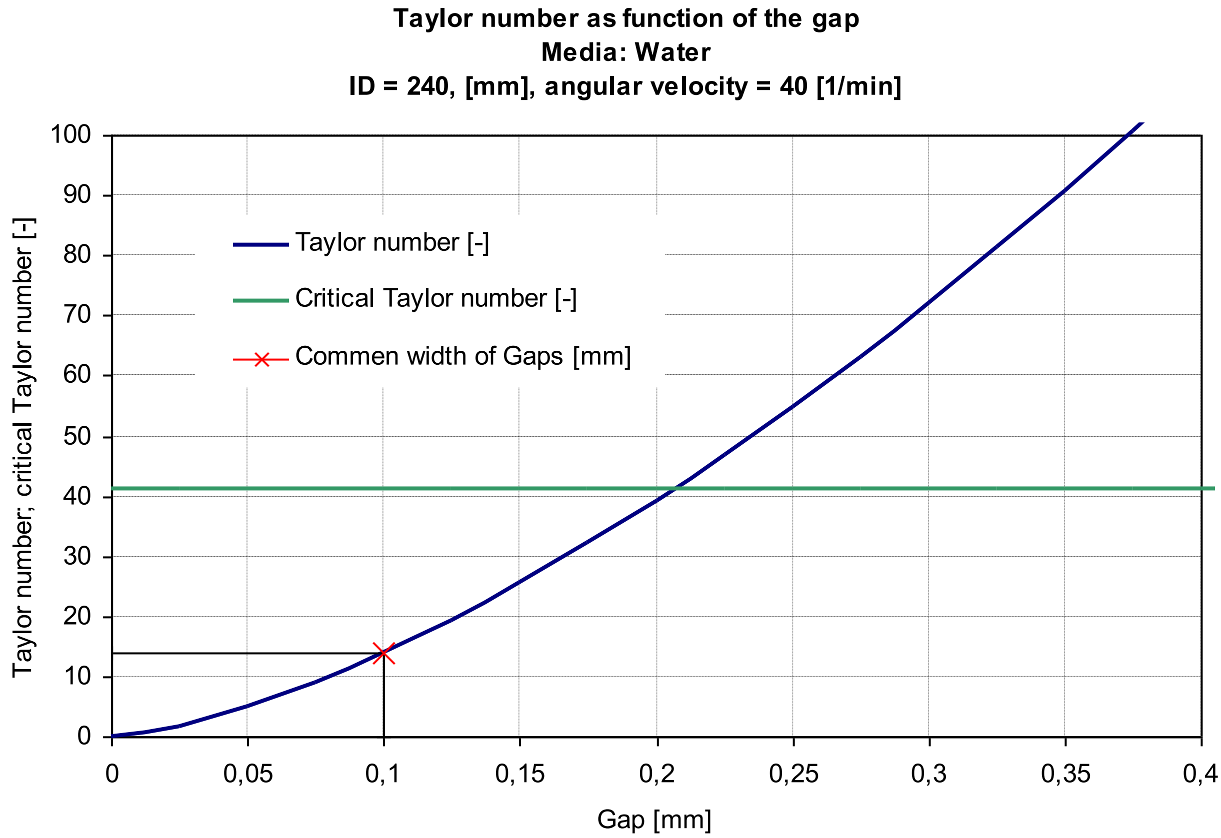
$$\upsilon : kinematic\ velocity$$



The use of the diagram in association with the modified Taylor number and the axial Reynolds number

$$\mathrm{Re} = \frac{\rho \, c \, D_h}{\eta} = \frac{\rho \, c \, 2s}{\eta}$$

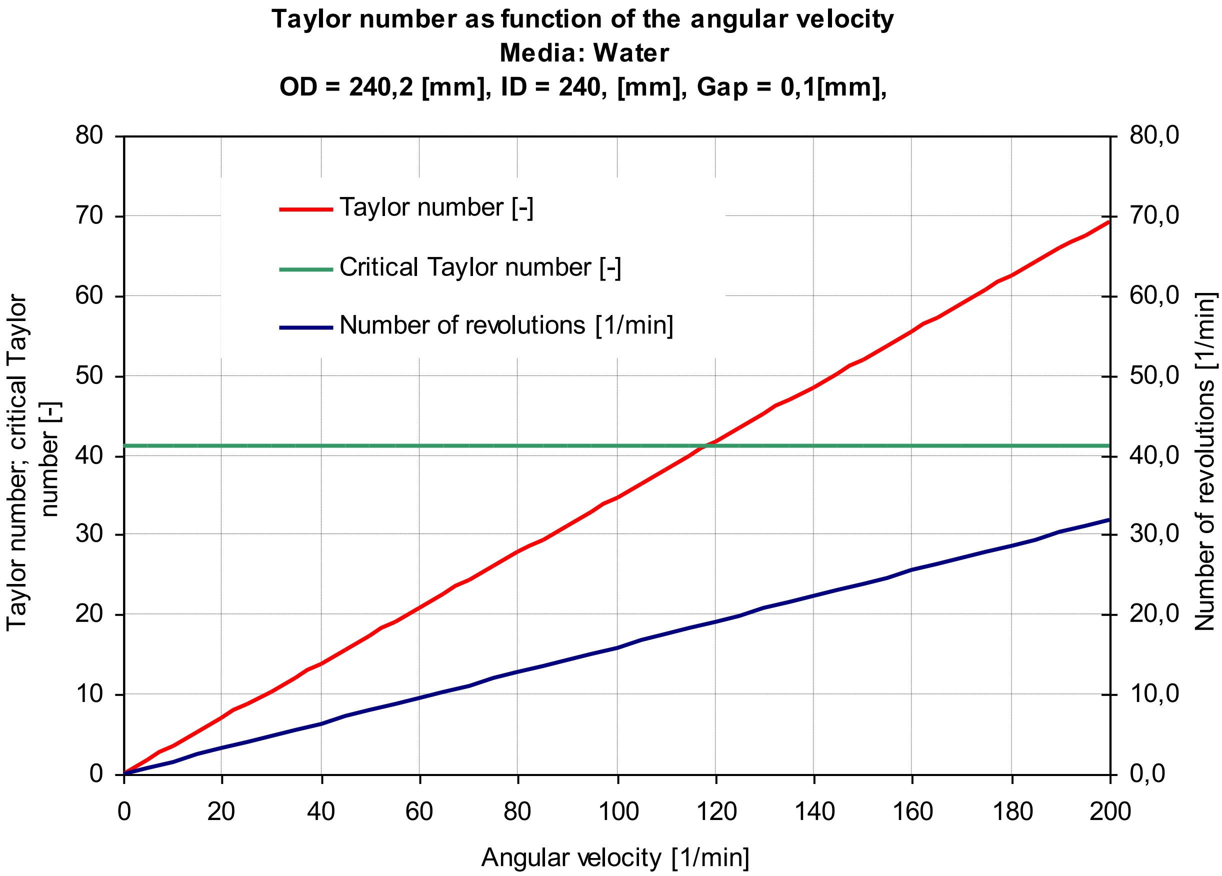
makes the finding of the optimised flow conditions and geometrical conditions possible. That is represented by the laminar area (Laminar- Flow) on the left side on the diagram.



The shown diagram displays the strong influence of the measurement (width) of the gap regarding the generation of Taylor vortices and the common width of 0,1 [mm].
The diagram shows –in associate with the formula above- the strong effect of the width regarding the generation of Taylor- vortices. This influences is characterized by the exponent of 1,5. The text has shown the importance of avoiding this phenomenon. In the case of single particles it is possible to calculate the necessary velocity of the sealing media to lift of the solid bodies as follows**:**

$$c_0 = \left[\frac{4}{3} \frac{(\rho_P - \rho_F)}{\rho_F} \frac{g\, D_{ST}}{\left(\frac{1}{3} \left(\sqrt{\frac{72}{\mathrm{Re}_p}} + 1 \right)^2 \right)} \right]^{0,5}$$

The equation can be used in the wide range in respect of the Reynolds number ($0,2 \leq \mathrm{Re}_p \leq 10^4$) and consequently in a wide range of the velocity of the sealing media. In the case of a PB (packed bed) the following formula makes the calculation of the necessary velocity possible:



$$c_{R(real)} > c_R = 7{,}19\left(1-\varepsilon_f\right)\frac{\eta}{\rho}a_{ph}\left\{\left(1+0{,}067\frac{\varepsilon_f^{\;3}}{\left(1-\varepsilon_f\right)^2}\frac{\left(\rho_p-\rho\right)g\rho}{\eta^2 a_{ph}^{\;3}}\right)^{\frac{1}{2}}-1\right\}$$

This necessary velocity is also known as the “minimum fluidization velocity”. The smaller the particles the larger the internal surface and the larger the pressure- drop. (31) The rotation of the shaft influences the fluidisation of the bed like also the media-flow. The increasing rotation of the bed affects an increase of the pressure- drop after reaching the state of fluidization. A generation of macro-circulation zones has also been observed. Scientists believe that the circulation causes concentration peaks regarding the media on the surface of the bed. The circulations can cause a back- mixing of the particles and thus an effect which is very important to avoid. (35)

6 Considerations regarding the pressure conditions

The next part of the work is focused to describe the maximal possible velocity of the sealing media. The maximal velocity is characterised by the appearance of cavitation. Cavitation has been a familiar phenomenon for a long time particularly in shipping. In 1917, the British physicist Rayleigh was asked to investigate what caused fast- rotating ship- propellers to erode quickly. He discovered that the affect of cavitation, already proved in experiments by Reynolds in 1894, was the source of the problem. Leonard Euler has predicted the generation of cavitation, based on theoretically consideration 1754. [36]
Furthermore, cavitation can arise in hydrodynamic flows when the pressure drops. This effect is, however, regarding to be a destructive phenomenon for most part. In addition to pump-rotors, control-valves are particularly exposed to this problem since the static pressure at the “vena contracta” even at moderate operating conditions can reach levels sufficient for cavitation to start occurring in liquids.

The consequences for a control- valve as well as for the entire control- process vary and are often destructive:

- Loud noise
- Strong vibrations in the affected sections of the plant
- “Choked flow” caused by vapour formation
- Change of the fluid properties
- Erosion of the valve components
- Destruction of the control- valve
- Plant shutdown

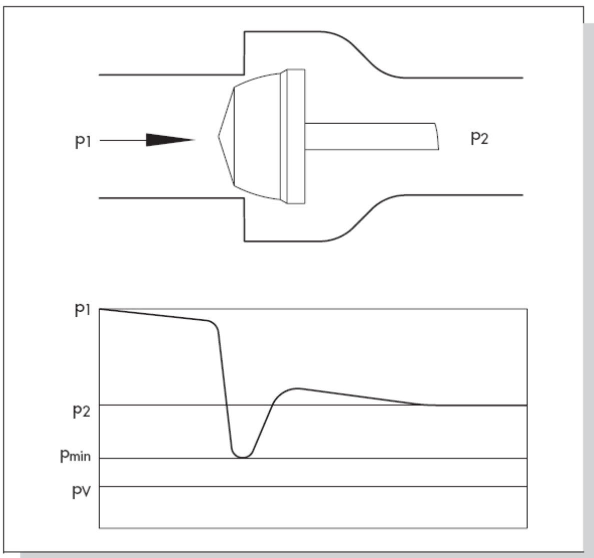
Cavitation shall be generally understood as the dynamic- process of the formation and implosion of cavities in fluids. Cavitation occurs, for instance, when high flow- velocities cause the local hydrostatic- pressure to drop to a critical value which roughly corrosponds to the vapor- pressure of the fluid. This causes small bubbles filled with steam and gases. These bubbles finally collapse when they reach the high- pressure areas as they are carried along by the liquis flow. In the final phase of the bubble implosion, high- pressure peaks are generated inside the bubbles and thei immediate surroundings. These pressure- peaks lead to mechanical vibrations, noises and material erosions of surfases in walled areas.If cavitation is severe, the hydraulic- valve coefficents as well as the fluid- properties change.

6.1 The cavitation coefficient x_{FZ}

In the case of less viscous liquid flows around streamlined bodies, the internal friction compared to the pressure may be frequently neglected. The velocity distribution of these types of flow can be calculated on the basis of the potential theory if the flow conditions are known. The pressure distribution along the body contour is derived from Bernoulli’s equation so that a relationship between the minimum pressure p_{min} and the critical pressure can be *stated* according to the following equation.
(The Bernoulli equation states that for an ideal fluid, without friction losses, an increase in respect of the velocity occurs simultaneously a decrease of the pressure and a change of the potential energy of the fluid.)

6.2 Critical flows in annular gaps (control valves)

In case of stalling flows as these occur in annular gaps (control valves), the potential theory cannot be used to determine the minimum pressure. Instead, the cavitation coefficient x_{FZ} has proven useful.

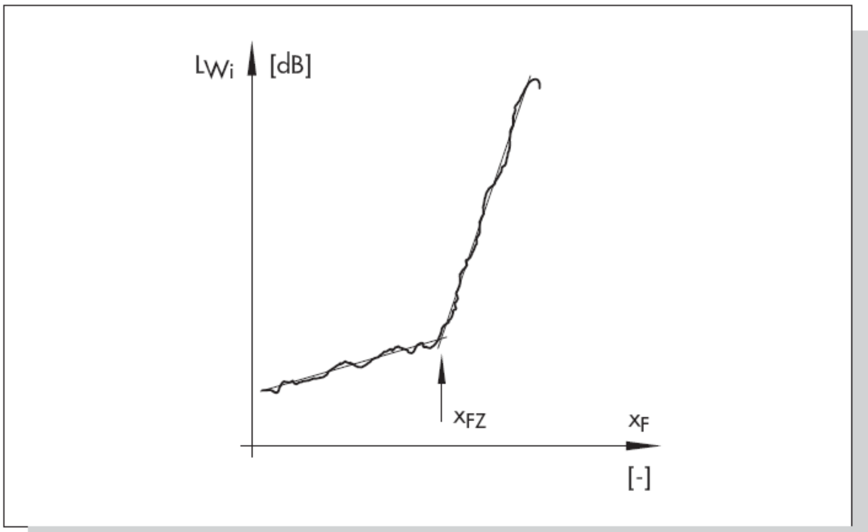


6.3 Distribution of pressure in the annular gap

It is based on the assumption that, in a control valve, the ratio of the external pressure difference (p_1 – p_2) to the internal pressure difference (p_1 – p_{min}) for all cavitation- free operating states equals a gap-specific value x_{FZ} .

$$x_{FZ} = \frac{p_1 - p_2}{p_1 - p_{\min}}$$

Since the minimum pressure occurs in one of the unsteady vortex cores downstream of the restriction, it cannot be determined by direct measurement.

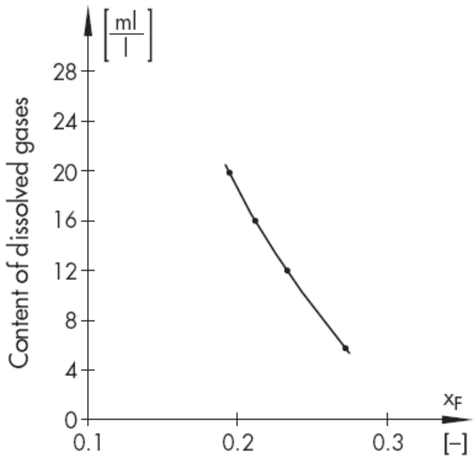


Determining of the cavitation- coefficient x_{FZ}

It is therefore assumed that the minimum pressure p_{min} equals the vapour- pressure p_v of the fluid when cavitation noise begins. Thus the determining of the pressure ratio xFZ as a function of the load by means of noise measurements is necessary.
When the x_{FZ} values are known over the entire range of application, it can be determined in advance for the all operating pressure ratios whether cavitation effects are to be expected.
In the case of an operating pressure ratio $x_F < x_{FZ}$, there is no danger of cavitation occurring, when $x_F < x_{FZ}$ and $x_F = x_{FZ}$, a stationary cavitation zone builds up whose expansion is roughly proportional to the difference (x_F - x_{FZ}).

$$x_F = \frac{\Delta p}{p_1 - p_v}$$

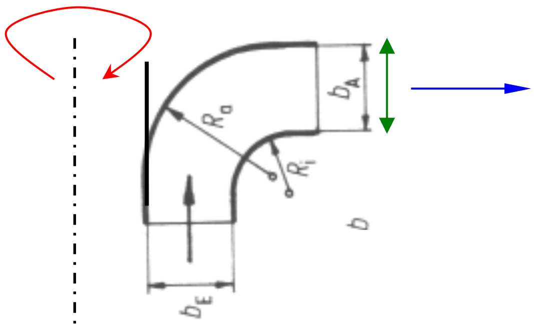
However, since the difference p_v – p_{crit} according to equation is not covered by the operating pressure ration x_F, these relationship can strictly speaking only be applied to media which confirm to the test medium water regarding their nuclei spectrum, surface tension and velocity. Oldernziel clearly showed this by measuring the pressure ratio x_F at the beginning of cavitation as a function of gas content of water.
The experimental x_{FZ} values should therefore be rounded to full five hundredths to account for the accuracy limits of the process.



The Ratio between the pressure ratio x_F and the gas content. [35]

7 Conclusion

The destructive effect of cavitation shows the importance to avoid this range regarding the pressure ratio and velocity. The avoiding of these ranges can be reached with the monitoring of the solid-borne sound respectively the measuring of the sound generally. An additional strengthening effect is generated by the rotation, but this effect has been neglected. The general aim to get a laminar flow is the best way to avoid this flow- condition. The reason to describe this state is the necessity to know the maximal velocity, if the solid bodies are fixed in the annular gap, because the impulse of the laminar flow is not enough to remove the material. The measurement of the cavitation- coefficient x_{FZ} is essential to give the operator –in addition to the minimal velocity - the complete range of working velocities. With the consideration of all described effects an optimised run and construction is possible.
Another advantage of the use of such a sealing system is the avoidance of any wear and consequently any frictional- heat and also a cooling effect caused by the media flow. On the other hand, the internal conditions of the vessel have to be regarding the temperature below the boiling temperature and have to be regarding the pressure above the boiling-pressure to avoid the generation of steam (vapour pressure curve).
Next to the formulas for the calculation of the necessary velocities the equations to calculate the corresponding pressure drop are also given. With the use of the given information an optimisation of this fluid- system should be possible. Based on the different influences it is possible to optimise the outlet of the gap. The unknown flow conditions affected by e.g. a mixer and the unknown exact height of the PB can be bypassed with the modified construction of the outlet. The change is shown below.



The bend is characterized by the radii (Ra, Ri) and the cross-sections (b_E, b_A) all of these measurements represent the modified outlet. The construction caused a deflection of the sealing media into radial direction. The upper part (Ra) shielded the outlet area of the gap against the outer flow and the flow of the solid bodies. Both parts have a similar shape to minimise the pressure loss. An additional positive effect is the use of the centrifugal force which is generated by the rotation of the shaft to keep

the outlet free of solid bodies. The discharge can be constructed as a flexible material or a movable part. This device would close the gap when the sealing- system does not work (green arrow).
Next to the use of the formulas a test should be the next step to get a better solution, because the formulas cannot take into consideration all the different influences.
Especially regarding the avoiding of wear and the cooling effects [37], this system can generally be classified as a future- oriented system.

8 List of abbreviations

Symbol	Description	Unit
a_{ph}	Spec. surface of the fill	m²/m³
Ar	Archimedes number	-
d	Difference of the radii	m
s	Gap	m
h	Height	m
Δp	Pressure drop	N/m²
p_0	Pressure influx into the annular- groove	N/m²
$p_{1/2}$	Ambient pressures	N/m²
$p_{hydrostatic}$	Hydrostatic pressure	N/m²
p_1	Inlet pressure	N/m²
p_2	Outlet pressure	N/m²
p_{min}	Minimal pressure	N/m²
R_1; R_2	Radii	m
r_m	Middle radius	m
Ω	Annular velocity	1/min
Ψ	Factor	-
Γ	Aspect ratio	–
g	Acceleration of gravity	kgm/s²
λ	Friction factor	-
l	Length	m
D_h; $D_{eff;}$ D_{ST}	Diameter; Equivalent diameter	m
ρ	Specific gravity	kg/m³
C	Form/shape factor	-
Ma	Mach- Number	-
Re	Axial Reynolds- number	-
Re’	Circumference Reynolds- number	-
Re_p	Particle Reynolds- number	-
Re_1; Re_2; Re_0	Reynolds- number	-
c	Velocity	m/s
c_p	Velocity in the fill	m/s
c_0	Velocity in the clear cross- section	m/s
k	Roughness	m
ε	Relative roughness	-
ε	Porosity	-
A	Cross section	m²
U	Periphery	m
b	Width of the triangle	m
h	Height of the triangle	m
φ	Form factor	-

La	Length to reach the laminar flow stage	m
ζ; ζ_I; ζ_O; ζ_D	Drag factor	-
V	Volume flow	m³/s
V_C	Volume of the fill	m³
V_P	Volume of the particles	m³
V_f	Total volume of the fill	m³
$f_R, f_{RMax.}$	Additional friction factor	m³
f_V	Eccentricity factor	-
η	Dynamic viscosity	Ns/m²
x_{FZ}	Cavitation- coefficient	-
x_F	Pressure ratio	-
L_{wi}	Acoustic power	dB

9 Literatur

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