The Influence of Thermal Expansion on the Design of Silicon Carbide Bearings

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1. Introduction

Ceramic slide bearings made of Silicon carbide (SiC) in sealless pumps have become accepted as a standard. The advantage of SiC over conventional material is that it has good corrosion- and high temperature resistance as well as a high hardness. All of this allows the use with hot and abrasive delivery media. From the design view there are difficulties in combining the physical very different materials ceramic and metal, particularly due to the different thermal expansion coefficients (17*10-6 1/K for stainless steal und 4*10-6 1/K for SiC) and due to the brittleness of the SiC, which only allows low tensile stresses.

Therefore in this paper the already existing design solutions will be researched and descript. The aim of this research is to list the available technical solutions in order to evaluate the quality of these solutions in the second part of the paper. In the first step of the search a patent search is carried out and in the second step internet resources and magazines are considered.

2. Patent search

The patent search was carried out by means of DEPATISnet. DEPATISnet is provided by the "Deutsches Patent- und Markenamt" (DPMA) and it enables the online search for patent publications all over the world. The patents are sorted in categories in the database. These categories are listed in the International Patent Classification (IPC).

In this search all patents in the IPC category F16C 17/22 were considered. Additionally searches with the terms in the Table 1 were carried out in the category F16C 17/24. The meaning of the both considered categories are listed in Table 2. Further all patents which were cited in the found patents were considered.

Search terms	
• Gleitlager Siliziumkarbid	• Slide bearing
• Wärmeausdehnung	• Thermal Expansion
• Maschinenwelle	• Shaft steel
• Lagerhülse	• Journal bushing
• Dehnungsausgleichselemente	
• Lagerspiel	• Journal play

Table 1: Used search terms

IPC Categories	
Description	**Meaning**
F16C 17/22	Sliding bearing with arrangements for the reconciliation of the thermal expansion
F16C 17/24	Sliding bearing with safety devices against unusual and unwanted conditions, e.g. against overheating

Table 2: Considered IPC classes and its meaning [4]

In the following chapter patents which are significant for this search are described.

2.1 Disk Springs (EP 0 771 957)

The invention of the KSB AG features that tensile loads on the bearing ring are avoided by a radial play between bearing bush and shaft. That's why expansions of the shaft don't act on the bearing ring. The radial centring of the bearing sleeve is realized over cone surfaces at the bearing sleeve and at two retainer rings over a large temperature range. The retainer rings exert pure compression stresses on the bearing sleeve. The different axial expansion of shaft, bearing sleeve and retainer rings, during a temperature rise, would lead with missing reconciliation to a gap becoming larger with rising temperature between bearing and retainer ring. This gap at the cone surfaces is compensated by spring elements, which are arranged between one of the retainer rings and an axial attachment. The spring element exercises an axial force on the retainer ring, which causes the retainer ring to shift on the shaft against the bearing sleeve. Thereby an actuated connection between the retainer rings and the bearing sleeve remains. The retainer ring connections exhibit close sliding fits, so that the bearing sleeve is centred by the cone surfaces in relation to the retainer rings and thus also in relation to the shaft. The reconciliation of the axial expansion can be defined by the dimensioning of the cone surfaces and the spring elements and is applicable over a great temperature range. As spring elements preferably disk springs are used [17].

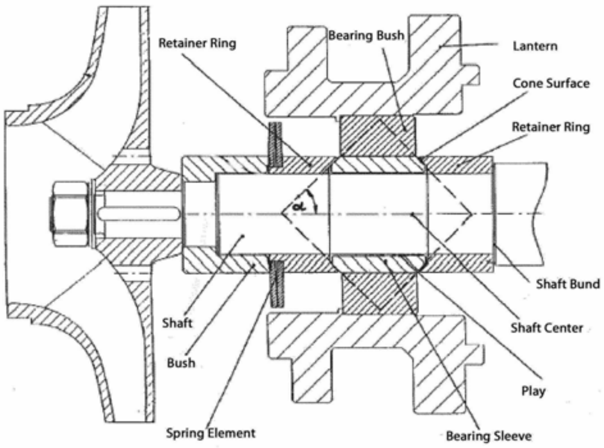


Figure 1: Representation of an execution of the invention as floating bearing [17]

Figure 1 shows a possible design of the connection described above. Here two disk springs are used as spring elements. These push the retainer rings and the bearing sleeve against a socket serving as axial attachment. The outer bearing ring is pressed into the housing (lantern) [17].

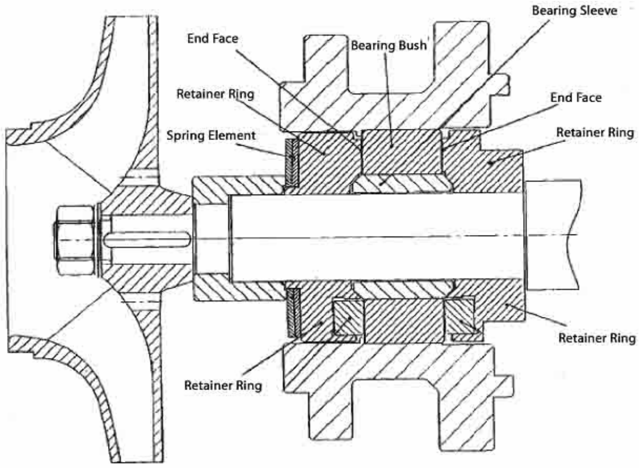


Figure 2: Representation of an execution of the invention as fixed bearing [17]

Contrary to the sliding bearing arrangement represented in Figure 1, which ensures only a radial positioning of the shaft relative to the housing (moveable bearing), the bearing arrangement shown in two alternative designs in Figure 2 allows a radial and an axial positioning of the shaft (fixed bearing). This is made possible by the fact that the retainer rings have sliding surfaces on their axial sides. The retainer rings represented in the top of Figure 2 have sliding bearing surfaces at the front, which cooperate with the axial surfaces of the bushes, and thus form an axial sliding bearing. The axial plays between the axial surfaces have to be designed in the kind that also at the maximum temperature arising during operation a shift of the retainer ring and the bearing is possible for the compensation of the thermal expansions. In order to be able to take up axial forces in both directions as well, the spring action of the spring element is dimensioned so that the force exerted by it is larger than the axial force which can be transferred.

2.2 Disk Springs 2 (EP 0 771 956)

In the KSB's patent from 1996, the invention described in the previous chapter is extended to a complete arrangement of bearings.

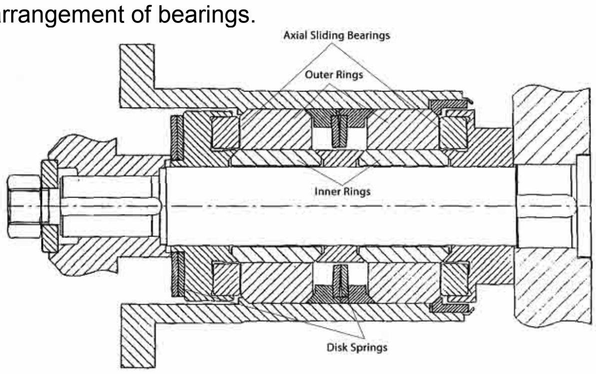


Figure 3: Representation of the complete bearing configuration [16]

In Figure 3 an arrangement of bearings in accordance with this invention is depictured. Here both the inner bearing rings as well as the outer bearing rings are positioned on cone surfaces. Thereby both the different radial as well as the different axial expansions between shaft and inner rings respectively between the outer bearing rings and the housing can be compensated [16].

2.3 V-groove with circlip (DE 2 000 601)

A similar solution with cone surfaces as described in the two previous chapters is also the basis of the patent of Leyland Gas Turbines Limited out of the year 1970 [6]. Here the cone surfaces, between which the inner bearing ring is wedged, are arranged so that they build a V-shaped groove, which opens radially outward. Due to this arrangement one of the cone surfaces can be implemented as part of the shaft. On the opposite side then however the cone surface must be part of a removable ring, in order to be able to install the bearing. Figure 4 shows a possible design of this invention [6].

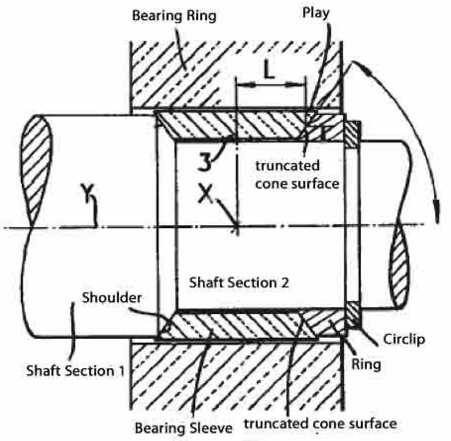


Figure 4: Representation of the bearing sleeve [6], strutted axially between two cone surfaces

The patent further describes that the removable ring should be made of a physically similar material as the shaft. Besides the tips of both cone surfaces of the bearing ring are supposed to lie in the point x on the rotation axis of the shaft. Since the shaft expands with a temperature rise in axial and radial direction, the angle of the cone surfaces to the rotation axis remains constant with this geometrical arrangement and this material selection. Due to this arrangement both the shaft with the removable ring as well as the inner bearing ring can expand without causing inadmissible tensile stresses in the bearing ring [6].

2.4 V-groove with nuts (FR 1 343 721)

The patent of the Commissariat A L'Énergie Atomique out of the year 1962 plans that both bearing rings exhibit a trapezoidal cross section, whereby the insides of the rings exhibit shorter axial lengths than the exterior rings. With the conical front surfaces resulting from this the bearing rings are each wedged between two check rings and thus fixed in axial and radial direction. Furthermore the invention plans that each bearing ring as well as the associated check rings are strutted by groove nuts against a shaft respectively a housing shoulder, in order to create a friction conclusive connection (see *Figure 5*) [20].

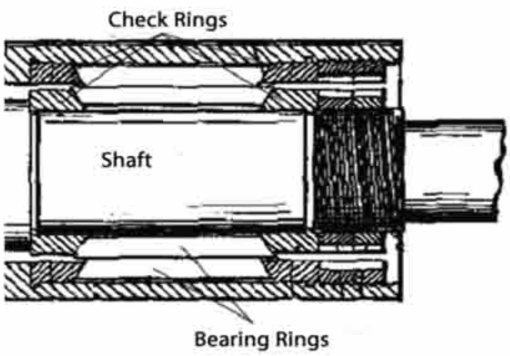


Figure 5: Representation of the bearing rings [20], strutted between the check rings

In the arrangement shown above the two bearing rings always remain centred to the surrounding structure, if the angle of the cone surfaces to the rotation axis is larger than the angle of the friction cone of the used material combination [20].

2.5 V-groove with pin (DE 9 102 933)

A solution similar to the two previous is described in the patent of the Rheinhütte GmbH & Co out of the year 1991 for a shaft made of plastic. Here the different expansions resulting in between the ceramic case and the plastic of the shaft compensate themselves by the fact that the bearing ring again is placed between conical surfaces. A nut is used to place an initial load on the arrangement in axial direction. At rising temperature the axial load remains constant due to the already known affect of the cone surfaces [8].

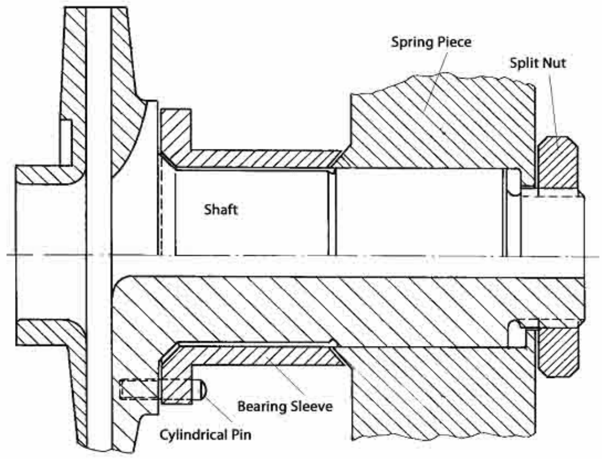


Figure 6: Representation of the clamped case [8]

From Figure 6 it can be seen that additionally to the frictionally engaged contact at the cone surfaces a pin connection ensures a reliable rotatory connection between the bearing ring and the parts with the conical surfaces.

2.6 Memory Metal (EP 0 264 661)

With the invention described in the patent of the Friedrich Krupp GmbH out of the year 1987 the occurrence of inadmissible tensions in the bearing ring due to different thermal expansions is prevented by dividing the shaft case into different parts along the circumference. The individual shaft cases are fixed to retainer rings by cone surfaces at their axial ends. To avoid a loosening of the shaft case due to a larger thermal expansion of the shaft in comparison to the shaft case, expansion balancing elements made of MEMORY metal are placed between the axial ends of the shaft cases and the retainer rings. A MEMORY metal can "remember" it's original form before a transforming process and pushes back into this old form when a certain temperature is exceeded. The change point of the MEMORY metal can lie between ambient temperature and 400°C [12]. Since due to the radial thermal expansion of the shaft the extent of the gap between shaft and bearing ring becomes larger, a loosening of the shaft cases in circumferential direction would arise. In order to prevent this, the invention provides compensating strips, which likewise consist of MEMORY metal, between the shaft case sections. A possible design of this invention is represented in Figure 7 and Figure 8. Here a three-divided shaft case is represented. The patent however plans arbitrary divisions [12].

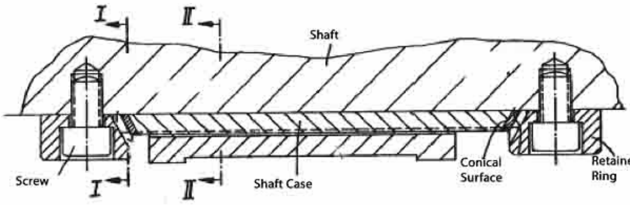
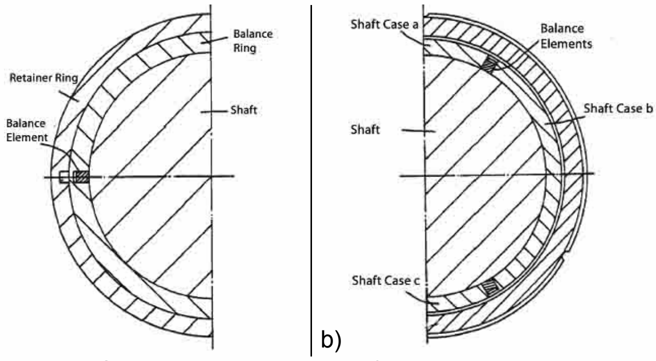


Figure 7: Partial profile by the shaft, the shaft case and the bush [12]



a) b)

Figure 8: Cross sections by the shaft and the retainer ring in the place Ii (a) and place ii-ii (b) in Figure 7 [12]

In the patent it further is suggested to make these compensating strips in axial direction longer than the shaft case, so that the supernatant ends reach into appropriate slots in the retainer rings and thus prevent a twist of the shaft case relative to the shaft [12].

2.7 Ring Webs (DE 1 068 516)

Caro-Werk GmbH's patent from 1959 aims off to create an outer ring of a sliding bearing with which the form of the sliding surface remains cylindrical at all conditions. This is to be achieved by the fact that the outer bearing ring is pressed not directly but indirectly over two back-up rings into the housing. The connection between bearing ring and the back-up rings is reached by the fact that the bearing ring has a ring web on each axial end which reaches over a coaxial positioned ring web of smaller diameter, which is part of the back-up ring. Between the webs of the bearing and those of the back-up rings intermediate rings are intended. These are made of a material with very high thermal expansion coefficient, so that they adjust the larger thermal expansion of the bearing compared to the housing and the back-up rings, which results from the stronger heating up of the bearing in comparison to the housing. The bearing ring can expand freely and a safe connection between bearing and housing is given at all temperatures. In Figure 9 a possible realization of this invention is represented [5].

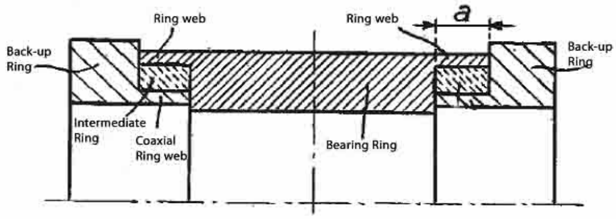


Figure 9: Representation of the bearing ring with the back-up rings, ring webs and the intermediate rings [5]

If the expansion of the sliding surface carrier is supposed to occur evenly, but in smaller extent than the natural thermal expansion, then the bearing ring has to be connected to a torus with a small coefficient of thermal expansion. This torus decreases the thermal expansion of the bearing ring. In accordance with the invention, this torus is to be put around the bearing and stand beyond the bearing ring in axial direction in order to form the ring web described above. Due to this arrangement it is possible to manage the expansion of the bearing ring by a suitable choice of the materials for the supporting body, the bearing ring, the intermediate ring and the torus [5].

2.8 INVAR case (US 2 590 761)

The in the year 1952 announced patent (General Electric) also deals with the expansion-fair construction of a ceramic bearing [30].

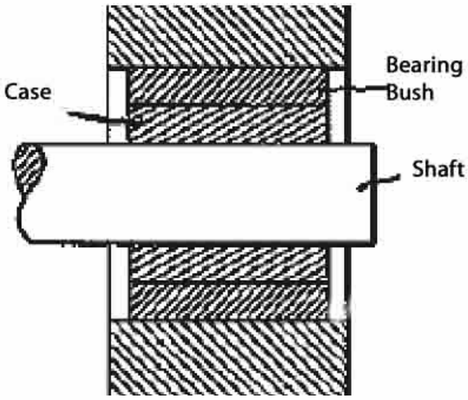


Figure 10: Representation of the shaft from steel with the case from an Invar case and the bush from ceramic(s) [30]

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Figure 10 shows a possible design of this invention with a shaft made of steel, on which a case from an Invar alloy with very low coefficient of thermal expansion is pressed. The inner ceramic ring is pressed on the outer circumference of the INVAR case. Due to the different materials the shaft exhibits a larger thermal expansion coefficient than the Invar case and the inner bearing ring. This leads to the fact that the shaft expands in a greater extend than the case during a temperature rise and thus stretches the case additionally. According to the invention the Invar case is selected in such a way that the resulting expansion of the outside diameter of the Invar case, which consist of the thermal expansion and the stretch caused by the shaft corresponds, to the thermal expansion of the inner bearing ring. With the stretching of the Invar case by the shaft the yield strength of the used Invar case may not be exceeded however, since otherwise after cooling down a plastic stretch of the Invar case would remain [30].

2.9 Carbon fiber coat (DE 69 412 616)

A similar solution, as described in General Electric's patent (see chapter 2.8), is pointed out in the patent of the Bofors AB from the year 1993. This invention plans that the thermal expansion of a light alloy shaft is limited by a coat of carbon fiber reinforced epoxy resin, which exhibits a very small thermal expansion coefficient. The coat however does not have to cover the entire exterior surface of the shaft. In *Figure 11* a hollow shaft with a carbon fiber reinforced epoxy resin coat and a ball bearing pressed on the shaft is represented [11].

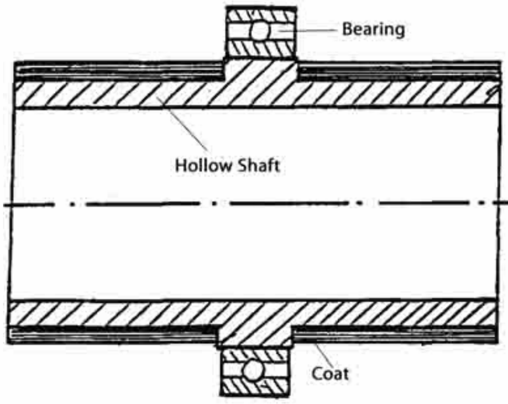

Figure 11: Representation of a hollow shaft with a coat from carbon fiber reinforced epoxy resin and a pressed on ball bearing [11]

From *Figure 11* it is evident that the coat is interrupted in the place, in which the bearing is pressed on. Yet the coat reduces the radial thermal expansion of the shaft also in the place of the ball bearing [11].

2.10 Tight fit outside (DE 9 419 709)

In the patent of the Sihi GmbH & CO KG out of the year 1996 an arrangement is described for centring a case made of a tension-sensitive material (ceramics) on a shaft [9]. On the shaft, which exhibits a shaft collar and a thread, the case is placed between two centring rings, which are manufactured from the same material as the shaft. This arrangement is axially strutted against the shaft collar over an intermediate ring by the nut present on the thread. Between the outside diameter of the shaft and the inside diameter of the case there is a clearance in cold condition. The case possesses cylindrical parts at both ends. These cylindrical parts have smaller outside diameters than the rest of the case. The centring rings exhibit a drilling. The parts of the case with smaller outside diameter are placed inside these holes. The diameters are chosen in such a way, that the there is a tight fit at the lower end of the temperature range. The centring rings are centred on the shaft. In the cold condition the centring of the ceramic ring is caused by the centring rings. If the temperature rises, then the ceramic ring expands in a lower degree than the shaft and the centring rings. If the inside diameter of the centring rings is in the relaxed condition equal to the appropriate outside diameter of the ceramic ring, then a clearance between these surfaces will occur during a temperature rise, while the clearance between the ceramic ring and the shaft decreases. If the ceramic ring is pressed with a tight fit into the centring rings, the tightness of this fit will decrease during the temperature rise. But the clearance between shaft and ceramic case still decreases and becomes a tight fit. Due to this the bearing ring can be centred by the centring rings at the lower end of the temperature range and by the shaft at higher temperatures. The turning entrainment of the ceramic ring is in all cases secured via axial wedging caused by the nut [9].

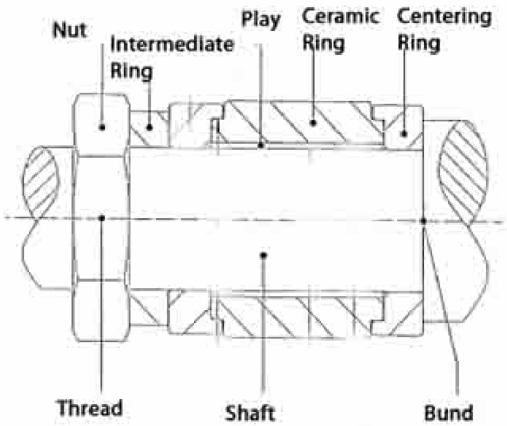

Figure 12: Representation of the bearing configuration in accordance with this invention [9]

In order to adapt the sum of the thermal expansions in axial direction to the axial expansion of the shaft an intermediate ring with a high coefficient of thermal expansion is placed between the nut and one centring ring [9].

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2.11 Case with springs (NL 1 027 464)

In the Ceratec Technical Ceramics B.V.'s patent from 2004 the problem examined here is solved with the help of a bearing ring carrier. This carrier is a cylindric part made of stainless steel, into which lengthwise several springy elements are integrated, which compensate both axially and radially the different coefficients of expansion of the steel shaft and the ceramic sliding bearing [28]. A closer description or a meaningful drawing is not contained in the patent.

2.12 Flexible case (EP 0 345 214)

A similar concept is described more detailed in the patent of the Gebrüder Sulzer Aktiengesellschaft from 1989 [14].

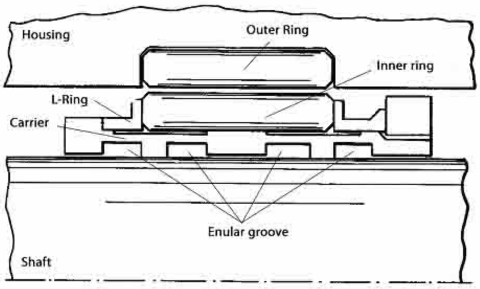

Figure 13: Representation of shaft, housing, sliding bearing and flexibly implemented carrier of the inner ring [14]

As represented in Figure 13, this invention plans a cylindrical bearing carrier, which exhibits circumferential grooves on the inside. Due to these grooves the carrier gets flexible and can compensate the larger radial thermal expansion of the shaft in comparison to the inner bearing ring, without exerting large tensions into the bearing ring. Axially the bearing ring is fixed by L-shaped rings, which compensate the different axial expansion of bearing ring and bearing carrier by an axial pre-loading [14].

2.13 Thin-walled case (CH 0 341 362)

Caro Werk GmbH's swiss patent from 1956 also describes such a flexible span. Here this span serves as carrier for the outer bearing ring, in order to prevent that the inside diameter of the outer bearing ring and the radial clearance are reduced with rising temperature. Nevertheless this solution can also be used for the inner ring [1].

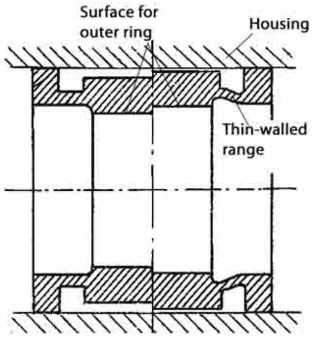

Figure 14: Representation of the bearing ring carrier left in the undeformed, right in the deformed condition [1]

As it is recognizable in Figure 14, a circular body is concerned here, which has at its axial ends ranges of larger outside diameter, which have an actuated connection to the housing. Axially inward adjacent to these ranges, a range with very thin wall thickness is intended. This part makes it possible that the middle range of the carrier, which takes up the outer bearing ring can expand radially relative to the range with increased outside diameter and thus also relative to the housing [1].

2.14 Laser Manufactured Spring Elements (EP 0 280 030)

Likewise a flexible element in form of a case is planned in the invention described in the Forschungszentrum Geesthacht's patent from 1987. This case is arranged between the shaft and the inner bearing ring. In order to obtain the springy effect, several u-shaped slots are intended (see Figure 15) over the circumference of the case. These slots let springy tongues develop. Furthermore in places outside of the springy tongues, flattened ranges on the outside surface are intended, so that the case exhibits a smaller outside diameter there. This leads to the fact that the inner ring is only in contact with the tongues. On the inside of the case, a backing off is intended. Within this range the case exhibits an increased inside diameter. Background of this design is that the shaft and most parts of the case can expand at rising temperature, while the springy tongues exhibit a constant outside diameter and thus only little tensile stresses are caused in the bearing ring [13].

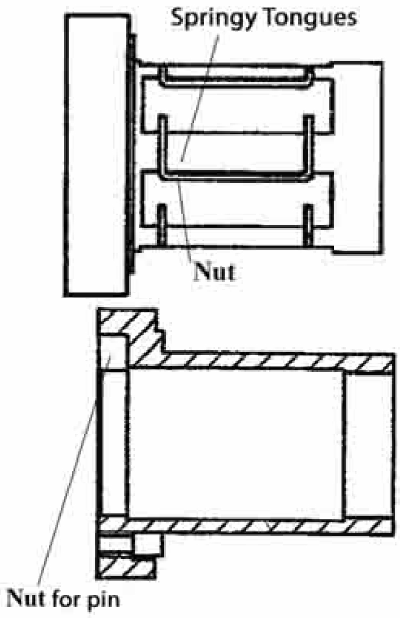

Figure 15: Representation of a case with springy tongues [13]

In order to guarantee furthermore a driving of the case with the shaft, still another groove is intended in the case, which produces a form-fit connection with a pin in the shaft [13].

2.15 Fluid as span (DE 10 045 301)

Allweiler AG's patent right from 2000 refers to the bedding of a shaft for centrifugal pumps, in which the bearings are subjected with a rotating load, which increases with rising number of revolutions. In order to make a centring of the inner ring possible without causing inallowable tensions in a large temperature range, it is intended to place a flexible element between shaft and inner bearing ring. Here a viscous medium is used. This medium opposes an appropriate centrifugal force to the rotating force on the outside, which provides for a centring of the bush relative to the shaft. In order to keep the medium in the gap between shaft and inner bearing ring sealing elements are used. These are also used for the centring at low numbers of revolutions. As represented in Figure 16, O-rings-seals can be used for this [10].

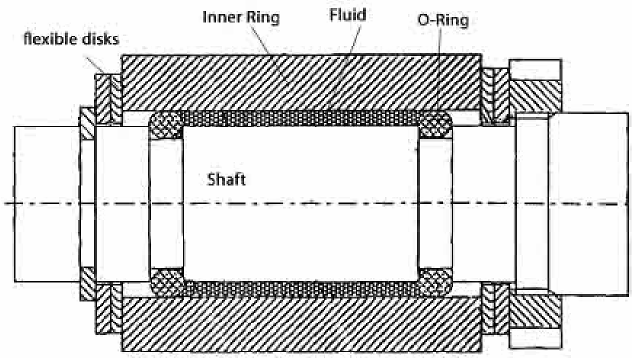

Figure 16: Representation of the inner ring with O-rings and viscous medium [10]

The adjustment in axial direction and in circumferential direction is reached by axial pre-loading. Flexible disks are used to keep the axial preload constant [10].

2.16 Tight fit outside with span (EP 0 563 437)

The invention described in Feodor Burgmanns Dichtungswerke GmbH & CO's patent from 1992 intends to centre the inner ring radially from the outside in order to create a sufficient play between shaft and inner ring. This play serves to compensate the different thermal expansions of shaft and inner ring. Outside at the bearing ring, rings made of a material with a thermal expansion coefficient similarly to that of the inner ring are planned. These rings are pressed firmly into carrying rings made of steel, which are connected with the shaft. The tight fit between the rings must be so firm that also at high temperatures no loosening arises due to the different thermal expansion [15].

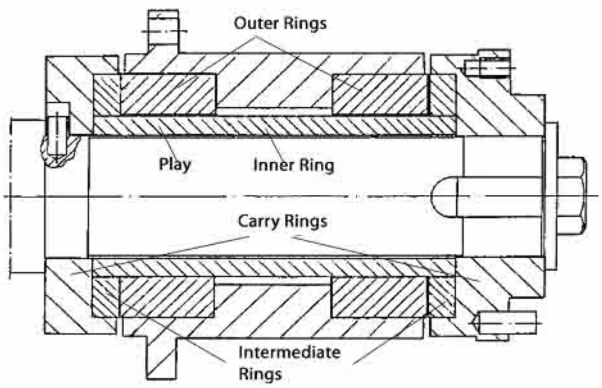

Figure 17: Representation of the bearing configuration in accordance with this invention [15]

In the arrangement of bearings represented in Figure 17 the two intermediate rings exhibit sliding surfaces, which form axial sliding bearings with the outer bearing rings. Due to this design also axial forces can be transmitted [15].

3. Internet and literature search

The Internet and magazine search was done with the help of the Internet search machine Google to find sources in the Internet [22] and in the DOMA data base of the Fachinformationszentrum Technik e.V. for the location of sources in books, theses, technical periodicals, conference reports and research reports. The DOMA data base supplies bibliographic referring to the German and international technical literature of the machine and equipment construction, to the production and processing of materials and the finishing technique [19]. The searches were done in the Internet and in the literature data base with the technical terms from Table 1. However no usable realizations could be found, since none of the companies publishes detailed constructional solutions.

4. Evaluation and Selection

After in the preceding chapters some possible solutions for the connection of a steel shaft with a sliding bearing made of ceramics were pointed out, these solutions are to be evaluated now, in order to select some suitable solutions for closer examination.
The following requirements are to be considered for the connection:

- Low tensile stress in the inner bearing ring caused by different thermal expansions
- Safe rotatory connection between the bearing ring and the shaft
- Good radial centring of the bearing ring
- Good resistance against abrasive and aggressive pump media
- Good heat resistance
- Safe and simple construction of the connection
- Low manufacturing costs

Before the evaluation it is important to identify solutions that do not fulfil any one of the requirements at all, since otherwise a useless solution might receive a good total evaluation due to other "strengths". Looking at the field of solutions with respect to the criteria it becomes obvious that the solutions "Carbon fiber coat" (see chapter 2.9) and "Fluid as span" cannot fulfill the demanded resistance (see chapter 2.15) due to the used materials (carbon fiber, rubber). Thus these two solutions are no further pursued. Likewise the solution "Case with springs" (see chapter 2.11) will not be considered any longer due to the low degree of concretion of the patent.
The criteria consulted for the evaluation have to be independent and must also point out actual differences between the solutions. Since the rotatory connection, the centring of the bearing ring and the resistance of the used materials against abrasive and aggressive pump media as well as high temperatures are fulfilled by all solutions in case of a safe and simple construction, these requirements will not be directly considered as evaluation criteria. The chosen evaluation criteria are specified in Table 3. The demand "Low manufacturing costs" is partitioned here into the criteria "Few components" and "Inexpensive components" in order to simplify the evaluation.

Evaluation criteria	
1.	Good technical properties
1.1	Low tensile stress
1.2	Safe and simple design with respect to the requirements posed
2.	Low manufacturing costs
2.1	Few components
2.2	Inexpensive components (standard parts)

Table 3: Evaluation criteria

In the following the remaining solutions are evaluated individually regarding each evaluation criterion with a point spectrum between 0 and 10. The value "0" does mean "not suitably" here, while the value "10" indicates "a very good suitability". In Table 4 the entire evaluation is summarized. The selection of the solutions which can be further pursued is done via addition of the individual points of each solution and the sum is divided by the number of evaluation criteria to determine the average score of a solution. The criteria are not weighted thou.

4.1 Good technical Properties

4.1.1 Low tensile stress in the bearing ring

Regarding this criterion all solutions, where, due to the arrangement, no tensile stresses in the bearing rings arise, are evaluated with 10 points. Exemplarily the solution "Disk Spring" (see chapters 2.1and 2.2) is named here, because in these solutions the cone surfaces are arranged in such a manner, that only compression stresses in the bearing ring develop.
The solutions, where tensile stresses in the bearing ring are produced either by flexible elements or cone surfaces, are evaluated with 5 points. For the better ones of these solutions in the next step it has to be determined whether the permissible tension in the case is exceeded or not. The solution "V-groove with pin" (see chapter 2.5) is evaluated with 7 points, since in contrast to the other solutions the cone surface here only serves to position the case relative to the shaft. Thus smaller normal forces at the cone surfaces are necessary, which again leads to smaller tensile stresses in the bearing ring.

4.1.2 Safe and simple design with respect to the requirements posed

Regarding this evaluation criterion the solutions using materials with special characteristics, like MEMORY metal and INVAR, get low scores, since it is questionable whether a construction of such solutions with consideration to the demanded resistance is possible at all. Therefore the solutions "Memory Metal" (chapter 2.6) and "INVAR case" (chapter 2.8) are only given 2 points each. Better evaluations receive solutions using materials with a larger coefficient of thermal expansion than steel. These solutions are given 5 points, since a smaller uncertainty than with the solutions described before exists whether a construction is possible in the desired way. Likewise with 5 points are those solutions evaluated, which use spring elements between the bearing ring and the shaft, since due to the large tensions necessary for the deformation of the elements, relaxation must be considered and thus the design process is more difficult. Additionally a force pointing radial outward must be applied, to guarantee a reliable rotatory connection between the bearing sleeve and the shaft. Thereby however tensile stresses in the case are produced, so that this radial force is limited to a small size. During construction of the solutions "V-groove with circlip" (chapter 2.3) and "Tight fit outside with span" (chapter 2.16) the possible yielding of the material must be considered. Since with these solutions certain forces, not certain tensions, as with the solutions with spring elements, are desired, the risk of yielding can be decreased by enlarging the cross-section areas. Thus these two solutions are evaluated with 7 points here. With the solution "V-groove with nuts" (chapter 2.4) exists also the danger that yielding arises, however this can be compensated here by tightening the nuts. Therefore this solution gets 8 points here. The two solutions "Disk Springs" (chapters 2.1& 2.2) and "V-groove with pin" (chapter 2.5) are evaluated here with 10 points each, since in the first solution possible yielding is compensated automatically with the help of the diaphragm springs, while in the second solution due to the form closing substantially smaller forces are needed, and thus yielding must not be considered during the construction.

4.2 Low manufacturing costs

4.2.1 Few components

With the help of this criterion it is evaluated, how many parts additional to shaft and bearing ring are needed for the attachment of the inner bearing ring to the shaft. Since in the represented solutions at least one additional construction unit is needed, such a solution receives 10 points. For each further necessary part one point is deducted. The assigned points are summarized in Table 4.

4.2.1 Inexpensive Components (standard parts)

Concerning this evaluation criterion those solutions using materials with special characteristics (INVAR, MEMORY metal) receive only 1 point each, due to the expectedly high prices of the materials. Also materials with a large coefficient of thermal expansion, which exhibit the demanded resistance, might have a comparatively high price. Thus solutions, which need such materials, are evaluated with 5 points. Likewise 5 points receive solutions, which use spring steel, since high prices have to be encountered here also. Exceptions form this are made for the solutions "Laser Manufactured Spring Elements" (chapter 2.14) and "Disk Springs" (chapters 2.1& 2.2). The first one is evaluated with 3 points only, since high production costs for manufacturing the slots in the case have to be added to the high material costs. In contrast to it the second solution is evaluated with 7 points, since it stands to expect that diaphragm springs with the demanded characteristics might be available at lower costs.
With 9 points the remaining solutions are evaluated here, since standard parts and construction units with simple geometry made of a similar material as the shaft material are predominantly used here.

4.2.3 Summary of the evaluation

In Table 4 the whole rating is summarized. The five best solutions, according to this rating, are highlighted in yellow.

Evaluation					
Criteria	technical		financial		
Solutions	Low tensile stress	Safe and simple construction	Few components	Inexpensive components	Sum
2.1 & 2.2 „Disk Springs“	10	10	7	7	8.5
2.3 “V-groove with circlip“	5	7	9	9	7.5
2.4 V-groove with nuts“	5	8	7	9	7.25
2.5 V-groove with pin“	7	10	8	9	8.5
2.6 Memory Metal“	5	2	0	1	2
2.7 Ring Webs“	10	5	7	5	6.75
2.8 INVAR case“	10	2	10	1	5.75
2.10 Tight fit outside“	10	5	7	5	6.75
2.12 Flexible case“	5	5	7	5	5.5
2.13 Thin-walled case“	5	5	10	5	6.25
2.14 Laser Manufactured Spring Elements“	5	5	10	3	5.75
2.16 Tight fit outside with span“	10	7	5	9	7.75

Table 4: Evaluation of the remaining solutions

From the above result it could be read that in the following an arrangement with diaphragm springs, a generalized arrangement with v-groove as well as a tight fit from the outside have to be regarded. With the solutions with diaphragm springs however little optimization is needed, since there is no danger of tensile stresses in the bearing ring and during the construction only the springs have to be dimensioned in such a way that a secure connection between the bearing ring and the shaft is ensured. For this generalized arrangements with v-groove as well as a tight fit from the outside are regarded for closer examination. At the solution with the tight fit from the outside also a transition to an interior centring can be considered if necessary at the upper end of the temperature spectrum.

5. Examining the selected solutions

In this chapter the two selected solutions will be examined closer. For each solution first there will be simple analytic calculations of the forces required. Afterwards these calculations will be verified with more accurate Finite Element Analyses. With the results of these simulations some parameters will be changed, so that the best combination of parameters can be found. But first of all the used materials are described.

5.1 Material Properties

The two mainly used materials in seal less pumps are steel (1.4408) and Silicon Carbide (SiC) [25]. For later use the typical properties of these materials are shown in the tables below.

Material Properties: SiC				
Compressive Strength	σ_d	3541	MPa	[26]
Tensile Strength	σ_z	83.4	MPa	[31]
Modulus of Elasticity	E	417000	MPa	[26]
Poisson's ratio	ν	0.16-0.17		[18]
Coefficient of thermal expansion	α	4.1*10-6	1/K	[26]

Table 5: Typical material properties of SiC

Material Properties: Steel 1.4408				
Yield point	$R_{p\,0.2}$	185	MPa	[29]
Tensile strength	R_m	440	MPa	[29]
Modulus of Elasticity	E	210000	MPa	[23]
Poisson's ratio	ν	0.3		[23]
Coefficient of thermal expansion	α	17*10-6	1/K	[23]

Table 6: Typical material properties of steel 1.4408 (GX5CrNiMo 19-11-2)

Since in both solutions a friction based connection between the bearing ring and the shaft is proposed, also the coefficients of frictions are important for the next step. Therefore for the important material combinations they are listed below.

Coefficients of friction					
Steel	Steel	$\mu_{St\text{-}St}$	Dry	0.15	[23]
			Wet	0.07	[23]
SiC	Steel	$\mu_{SiC\text{-}St}$	Dry	0.3	estimated
			Wet		
SiC	SiC	$\mu_{SiC\text{-}SiC}$	Dry	0.5	[24]
			Wet	0.35	[24]

Table 7: Coefficients of friction for combinations of steel and Silicon carbide

With the help this data the applied forces in each of the selected solutions can be calculated.

5.2 V-groove

First the general assembly with a V-groove will be examined closer. The solution has two rings which are tightly fit to the shaft. The bearing ring is located between the two rings. An axial force presses the rings towards each other and therefore both rings are pressed against the bearing ring. The contacting surfaces of the three rings have a conical shape to allow a greater thermal expansion of the shaft and the two outer rings in comparison to the bearing ring (see chapter 2.3 to 2.5). Since a relative motion between any of the outer rings and the bearing ring would lead to a failure of the whole pump, it is important that the axial force has such a magnitude that a sufficient friction based connection between the outer rings and the bearing is created. In this case the momentum caused by the friction in the bearing has to be transmitted.

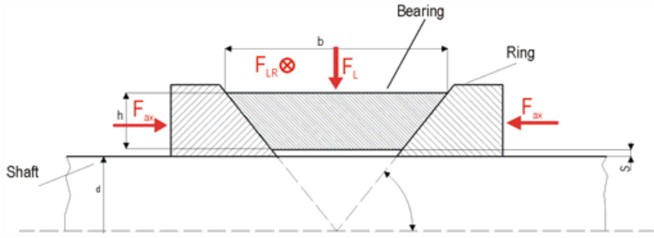

Figure 18: The arrangement with all necessary measures and the forces working on it: The angle α is defined here, in such a way, that the pikes of the two conical faces are supposed to lie in the same spot on the middle axis

This gets clearer, when cutting out the inner bearing ring and locking at the forces working on the bearing ring (Figure 19). On the outside of the inner bearing ring there is the bearing force F_L working, with which the two bearing rings are pressed together.

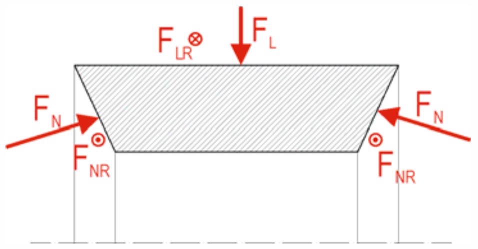

Figure 19: Picture of the bearing ring with the forces working on it

The resulting friction force F_{LR}, which is working on the outside of the inner ring as well, can be calculated with the following formula:

$$F_{LR} = \mu_{SiC-SiC} \cdot F_L \qquad \textit{Eq. 1}$$

Since the bearing ring is not supposed to be accelerated, the sums of all forces and momentums working on the ring have to equal zero (see Eq. 2).

$$\sum M = 0 = F_{LR} \cdot \left(\frac{d}{2} + s + h\right) - 2 \cdot F_{NR} \cdot \left(\frac{d}{2} + s + \frac{h}{2}\right)$$
$$\Rightarrow F_{NR} = \frac{1}{2} F_{LR} \frac{d + 2 \cdot s + 2 \cdot h}{d + 2 \cdot s + h} \qquad \textit{Eq. 2}$$

In this equation d is the diameter of the shaft, s the clearance between shaft and bearing ring and h the thickness of the bearing ring (see Figure 18).
The force F_N working normal on the conical surfaces can be calculated with Eq. 3. With Eq. 4 the required axial Force F_{ax} can be computed.

$$F_N = \frac{F_{NR}}{\mu_{SiC-St}} \qquad \textit{Eq. 3}$$

$$F_{Ax} = F_N \cdot \sin(\alpha) \qquad \textit{Eq. 4}$$

The angle α between the conical surfaces and the rotational axis can be computed with Eq. 5 (see Figure 18).

$$\alpha = \arctan\left(\frac{d + 2 \cdot s + 2 \cdot h}{b}\right) \qquad \textit{Eq. 5}$$

Since only the surface pressure p_L in the bearing is known, the force in the bearing F_L has to be calculated first. The given surface pressure is related to the projected gliding area [25], therefore the bearing load F_L can be computed with Eq. 6.

$$F_L = (d + 2 \cdot h) \cdot b \cdot p_L \qquad \textit{Eq. 6}$$

The clearance s is left out here, since there is none in the examples shown in [27].

With the help of the drawings in [27] typical dimensions of a pump and the bearing can be estimated. Here the shaft will be modelled with a diameter of d = 25 mm, the bearing ring will have a thickness of h = 6 mm and the axial width b will be 20 mm. For all coefficients of friction the one for wet conditions are chosen here, since the bearing would be destroyed when running in a dry condition anyway. The surface pressure in the bearing will be up to p_L = 2 N/mm^2 [25].
With this data and the above equations the angle α can be calculated to α = 62.68° and the required axial force to F_{ax} = 1297 N. All dimensions and forces are summarized in Table 8.

Dimensions and forces		
d	25	mm
h	6	mm
b	20	mm
p_L	2	N/mm^2
F_L	1480	N
F_N	1020	N
α	62.85	°
F_{ax}	907	N

Table 8: The dimensions of and forces working on and around the bearing

With this information the tensile stress in the bearing ring can be approximated. Therefore first the component of F_N pointing outward in radial direction has to be determined:

$$F_{N,rad} = F_N \cdot \cos(\alpha) = 1020\,\mathrm{N} \cdot \cos(62.85°) = 465.45\,\mathrm{N} \qquad \textit{Eq. 7}$$

Since this force is located on both sides of the bearing ring, it has to be considered 2 times to calculate the approximate tensile stress. This can be done with Eq. 8. The cross sectional area in this equation is about A = 102 mm^2.

$$\sigma_Z \approx \frac{2 \cdot F_{N,rad}}{2 \cdot A} = \frac{2 \cdot 465.45\mathrm{N}}{2 \cdot 102 mm^2} = 4.56 \frac{N}{mm^2} \qquad \textit{Eq. 8}$$

In order to determine the stress, especially the tensile stress, in the bearing ring, more accurately a FEA in Algor is set up with the dimensions and forces shown in Table 8. Since the forces are not axisymmetric, the assembly is modelled in 3D. The bearing force FL and force in axial direction Fax are modelled as area loads like shown in Figure 20. The contact between the bearing ring and the two outer rings and the contact between the two outer rings and the shaft are modelled as surface contacts with friction. The coefficients of friction are chosen for wet conditions (see Table 7). The temperature rise is set up as a part temperature of 600°C.

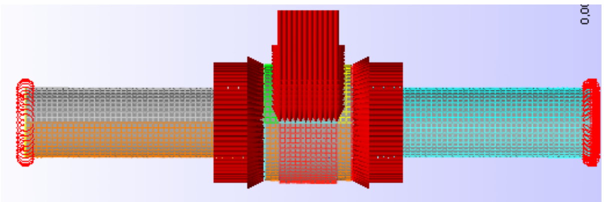

Figure 20: The 3D-FEA-Modell, with the bearing and the axial forces

The results of the analysis are given out as the normal stresses according to Mohr's Circle for the ceramic bearing. This allows differentiating between the very high compressive and the low tensile strength of the silicon carbide. Other theories like the Coulomb-Mohr Theory or the Modified Mohr Theory describe the strength more accurately [1]. But the usage of these theories is not as easy as the Maximum Stress Theory according to Mohr. Therefore the Mohr Theory is used here in order to get quick results.
Since in this case the bearing ring is the weakest part, at first it will be the only one considered. The results of the first 3D-Analyses show that there are hardly any differences between the Analyses with the bearing force F_L and those without it. The stresses in the bearing ring even appear to be smaller in the Analysis without the force. Therefore the parts for the next Analyses are modelled as axisymmetric 2D parts, in order to get quicker results. Also the symmetry of the assembly is considered here. Only one of the two outer rings and half of the bearing ring are modelled.
In a first step only the axial force F_{ax} is put onto the outer ring, in order to determine the resulting stresses. In the next analysis also a temperature rise of 600°C is considered. The result of the first analysis is shown in Figure 21.

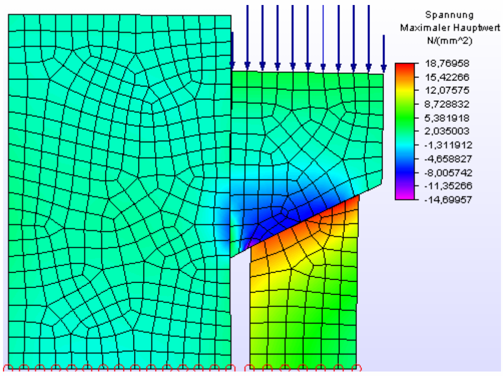

Figure 21: Results of the analysis only with axial force

It can be seen in the picture that the highest stress occurs close to the contact surface between the bearing ring and the outer ring. The maximum normal stress in the bearing ring is about 19 N/mm^2 and therefore much smaller than the tensile strength of SiC. The contact force between the bearing ring and the outer ring is about 1020 N and therefore has about the same magnitude as the needed normal force F_N. About the same results occur, when the temperature rise of 600°C is applied. This shows that the temperature rise has no effect on the stresses developing in the bearing ring.
In order to verify the results of the 3D-Analyses, where it could be seen, that stresses in the bearing ring even get smaller when, adding the bearing force F_L, this force is applied on the assembly for the next analysis. But through to the modelling as axissymmetric 2D parts a force put on the outer edge of the bearing ring, is interpreted as a force equally spread over the whole circumference pointing inward in radial direction. So the resulting stresses only give a figure in which way the stresses in the bearing ring change. In Figure 22 the results of the analysis with temperature rise and bearing force are depicted.
It can clearly be seen, that the stresses in the bearing ring are lower than those in the previous analyses. Therefore in the next steps the bearing forces can be neglected.

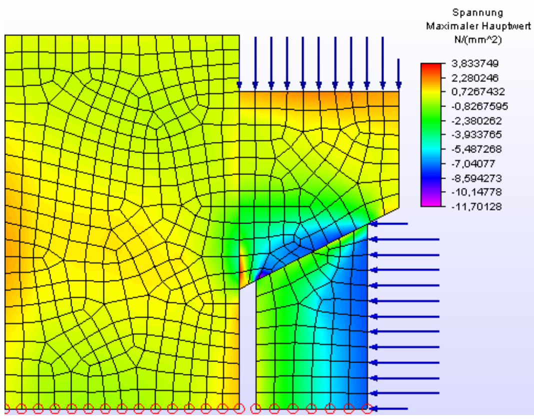

Figure 22: Results of the analysis

The analyses described above all use an axial force on the outer ring to produce the needed normal force F_N. In reality this could only be done by a spring pressing against the ring. This is not the supposed setup. Therefore in the next step the outer ring is connected to the shaft and a stress free temperature for the shaft is defined, in order to create the necessary normal force between the outer ring and the bearing. Through to the conical contact surface the needed temperature is hard to calculate. Because of this reason the temperature has to be determined in an empiric way. Figure 23 shows the results of the analysis without a temperature rise.

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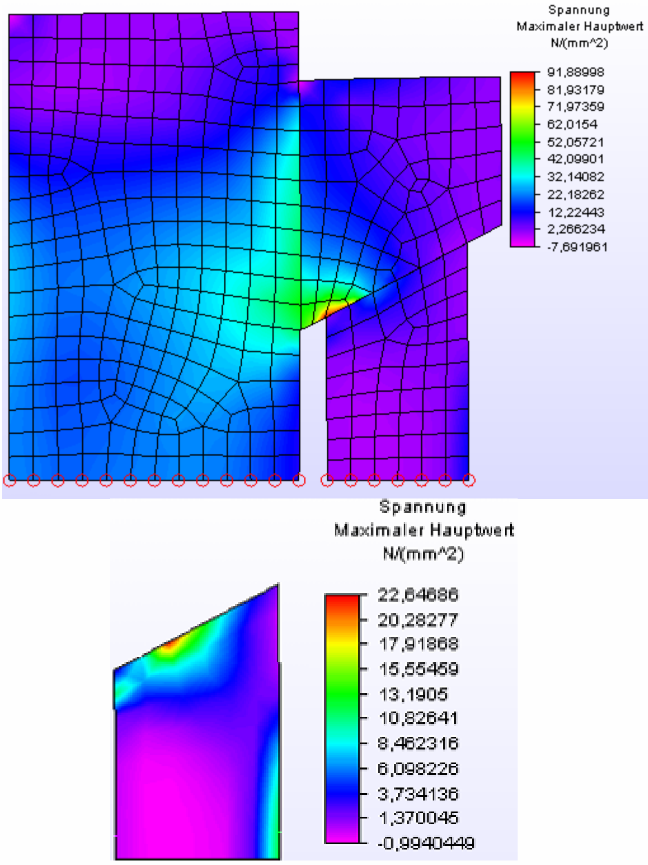

Figure 23: Results of the analyses at room temperature for the whole assembly (left) and just for the bearing ring (right)

The maximum normal stress in the bearing ring is about 23 N/mm^2 and the maximum stress in the assembly according to the distortion energy (von Mises) theory is about 101 N/mm^2. Thus both stresses are within the allowable magnitude for steel respectively silicon carbide. In Figure 24 the results of an analysis with about the same arrangement as above but with a temperature rise of 600°C are shown.

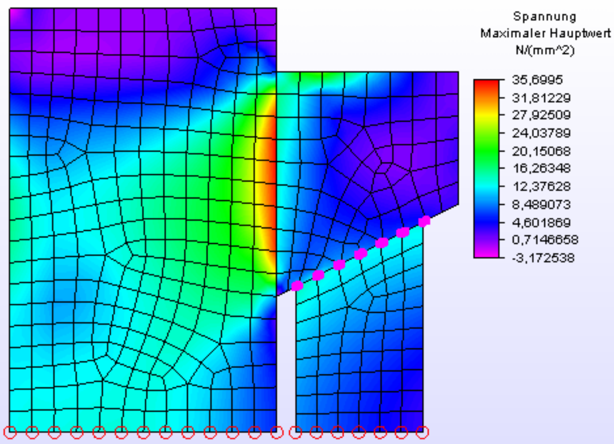

Figure 24: Results of the analysis with a temperature rise of 600°C.

Here also the occurring stresses do not exceed the allowable stresses. Therefore it can be said, that this solution in the described design is adequate to solve the problem researched in this paper. Therefore no changes and no optimisation are needed.

5.3 Tight fit on the outside of the bearing ring

Now the second selected solution with a interference fit on the outside of the bearing ring at lower temperatures and a tight fit on the inside surface at higher temperatures, like shown in Figure 25, will be examined closer.

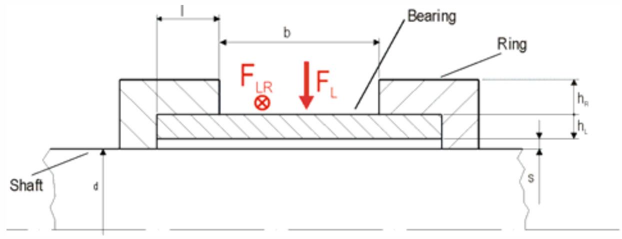

Figure 25: Assembly with a tight fit on the outside of the bearing ring at low temperatures and one on the inside at higher temperatures

A first rough dimensioning of the press fits around the bearing ring is done according to the dimensioning of a press fit between a shaft and a hub as described in [21]. Afterwards the results will be verified with Finite Element Analyses.
During operation of the pump, the fit between the bearing ring and the outer rings always has to be able to transmit the torque caused by the friction between the bearing rings. This torque T can be calculated with Eq. 9 (see Figure 25). The clearance s can be neglected for this calculation, since it will be very small in order to allow a tight fit between the bearing ring and the shaft at higher temperatures.

$$T = \frac{1}{2} \cdot F_{LR} \cdot (d + 2 \cdot h) \qquad \textit{Eq. 9}$$

In the following the equations will be set up to calculate the maximum and the minimal pressure in the fit between the outer rings and the bearing ring. Hereby only one of the fits will be considered, since due to tolerances it will never be the case that both fits equally transmit the torque.
The minimum pressure can be calculated with Eq. 10. In this equation S_R is a safety factor and μ is the coefficient of friction. For the meaning of the other variables see Figure 25.

$$\boldsymbol{p_{\min}} = \frac{\boldsymbol{2 \cdot T \cdot S_R}}{\boldsymbol{\pi \cdot \mu_{SiC-St} \cdot l \cdot (d + 2 \cdot h)^2}} \qquad \textit{Eq. 10}$$

The maximum pressure usually has to be calculated for both the hub and the shaft, but since in this case the ceramic bearing ring is compressed by the pressure and since the allowable compressive stress is much higher than the allowable tensile stress of the outer rings, the maximum pressure is only calculated for the outer rings here. This can be done with Eq. 11

$$p_{\max} = \frac{\sigma_{zul_N}}{2}\left(1-\rho_N^2\right) \qquad \textit{Eq. 11}$$

$$\rho = \frac{r_i}{r_a} \qquad \textit{Eq. 12}$$

In this Formula $\sigma_{zul,N}$ stands for the allowable stress (in the outer rings) and ρ_N for the ratio between the inner and outer radius of the outer rings (r_{Ni} and r_{Na}) (see Eq. 12).

$$\boldsymbol{r_{Ni} = \frac{1}{2}\cdot d + s + h_L} \qquad \textit{Eq. 13}$$

$$\boldsymbol{r_{Ni} = \frac{1}{2}\cdot d + s + h_L + h_R} \qquad \textit{Eq. 14}$$

With the results of the above calculations the minimum and the maximum overmeasure (U_{min} & U_{max}) between the bearing ring and the outer rings can be calculated (Eq. 15 and Eq. 16).

$$U_{\min} = (d + 2\cdot h)\cdot(\zeta_N - \zeta_W)_{\min} \qquad \textit{Eq. 15}$$

$$U_{\max} = (d + 2\cdot h)\cdot(\zeta_N - \zeta_W)_{\max} \qquad \textit{Eq. 16}$$

In these Equations ξ_N and ξ_W stand for the extensions of the outer rings ($_N$) and the bearing ring ($_W$). They can be determined with Eq. 17 and Eq. 18, with E being the modulus of elasticity and *v* the poisons ratio of the used materials.

$$\zeta_N = \frac{p_{\min/\max}}{E_N}\left(\frac{1+\rho_N^2}{1-\rho_N^2}+\nu_N\right) \qquad \textit{Eq. 17}$$

$$\boldsymbol{\zeta_W = -\frac{p_{\min/\max}}{E_W}\left(\frac{1+\rho_W^2}{1-\rho_W^2}-\nu_W\right)} \qquad \textit{Eq. 18}$$

With Eq. 19 the allowable temperature rise can be calculated, when assuming that the press fit has the overmeasure U_{max} before the rise and U_{min} afterwards.

$$\boldsymbol{\Delta T_{\min-1} = \frac{U_{\min}-U_{\max}}{(d+2\cdot h)\cdot(\alpha_W-\alpha_N)} = \frac{U_{\min}-U_{\max}}{(d+2\cdot h)\cdot(\alpha_{SiC}-\alpha_{St})}} \qquad \textit{Eq. 19}$$

In this formula α_W stands for the coefficient of thermal expansion of the bearing ring which is equal to α_{SiC} since the bearing ring is made of ceramics and α_N for the one of the outer rings which is equal to $\alpha_{St.}$

With the lowest occurring temperature T_{min} and Eq. 20 the maximum temperature T_1 at which the press fit on the outside of the bearing ring is tight enough to transmit the torque can be calculated.

$$\boldsymbol{T_1 = \Delta T_{\min-1} + T_{\min}} \qquad \textit{Eq. 20}$$

Since it will never be known how exactly the transmittable torques of the press fit on the inside and on the outside of the bearing ring add up, again only one at a time will be considered. Therefore now a similar calculation will be set up for the press fit on the inside of the bearing ring. Again the calculation starts with the minimum and maximum pressure in the fit. The torque that has to be transmitted is still the same (see Eq. 9).

$$\boldsymbol{p_{\min} = \frac{2\cdot T\cdot S_R}{\pi\cdot\mu_{SiC-St}\cdot(2\cdot l + b)\cdot d^2}} \qquad \textit{Eq. 21}$$

$$p_{\max} = \min\left\{p_{\max_W}\,;\,p_{\max_N}\right\} \quad \text{with}$$

$$p_{\max_W} = \sigma_{zul_W} \quad \text{and}$$

$$p_{\max_N} = \frac{\sigma_{zul_N}}{2}\cdot\left(1-\rho_N^2\right) \qquad \textit{Eq. 22}$$

In Eq. 22 $\sigma_{zul,N}$ stands for the allowable stress in the shaft. With these two results the minimum and maximum overmeasure can be calculated (see Eq. 23 and Eq. 24). The missing values again can be determined with Eq. 12, Eq. 17 and Eq. 18.

$$U_{\min} = d\cdot(\zeta_N - \zeta_W)_{\min} \qquad \textit{Eq. 23}$$

$$U_{\max} = d\cdot(\zeta_N - \zeta_W)_{\max} \qquad \textit{Eq. 24}$$

So the temperature rise between the temperature T_1, where the inner press fit has to transmit the torque and the maximum temperature T_2, where the allowable stress in either the shaft or the bearing ring is reached can be determined with the following equation:

$$\Delta T_{1-2} = \frac{U_{\max}-U_{\min}}{d\cdot(\alpha_W-\alpha_N)} = \frac{U_{\max}-U_{\min}}{d\cdot(\alpha_{St}-\alpha_{SiC})} \qquad \textit{Eq. 25}$$

Adding this to the temperature T_1, leads to the maximum Temperature T_2 (Eq. 26).

$$T_2 = \Delta T_{1-2} + T_1 \qquad \textit{Eq. 26}$$

These Formulae are implemented in a data sheet in order to find a good combination of dimensions, before the verification of the results with a FEA is started. The parameter that can be changed here are the length of the tight fit l, the width of the sliding surface of the bearing b and the heights of the bearing ring h_L and the outer ring h_R. When changing these parameters little effects on the results of the calculation can be seen. Therefore the dimensions are chosen in such a way that a good compromise between the allowable temperature range and the used installation space is found. In Table 9 the chosen dimensions are shown.

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Dimensions		
d	25	mm
h_L	6	mm
h_R	6	mm
b	20	mm
l	10	mm

Table 9: The dimensions of the assembly

With these dimensions the allowable temperature rise for the press fit on the outside of the bearing ring is calculated to ΔT_{min-1} = 73.615 K and the allowable rise for the inner press fit is calculated to ΔT_{1-2} = 17.204 K. So the possible overall temperature rise with this solution is 90.819 K. This does not fulfil the stated requirements at all, especially when considering that the calculations do not consider any safety factors and no machining tolerances. In order to verify these results in the next step a simplified arrangement consisting only of a shaft and two concentric rings on the outside is modelled for some Finite Element Analyses. Afterwards, when the calculations have been proven correct, the whole assembly will be analyzed.
The clearance between the shaft and the first concentric ring as well as the press fit between the two rings, is modelled by adding stress free temperatures to the two concentric rings.
Since in this case not necessarily the bearing ring is the weakest part not only the maximum normal stress is determined but also the stress according to the Von Mises Theory. The results of the analysis at the minimum temperature T_{min} are shown in *Figure 26*.
In the diagrams it can be seen, that nowhere the allowable stresses are exceeded, but there is also not much safety to increase the press fit left. All other additionally determined pressures and surface forces are comparable to the calculated ones. Similar notice can be made when comparing the results of the analyses at the transitional temperature T_1 and at the maximum temperature T_2 with the corresponding results of the calculation (see *Figure 27*).

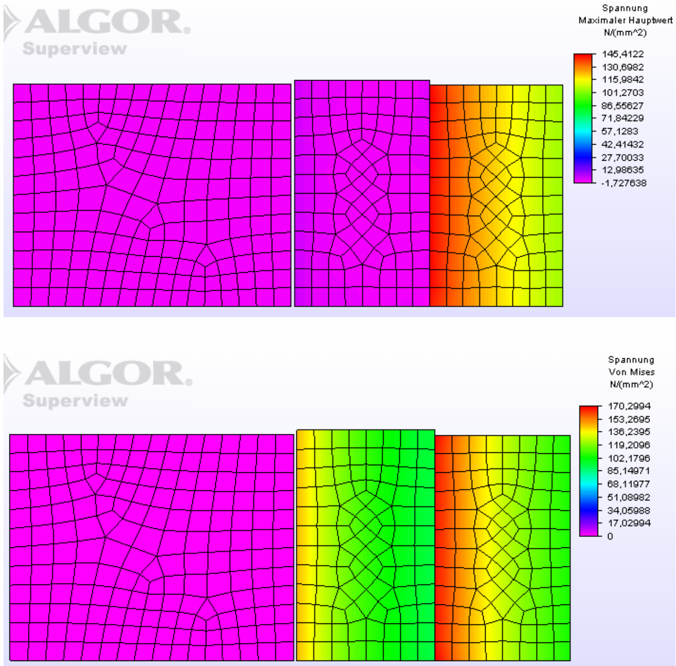

Figure 26: Maximum normal stress (top) and stress according to the Von Mises Theory (bottom) in the simplified arrangement at the minimum temperature T_{min}

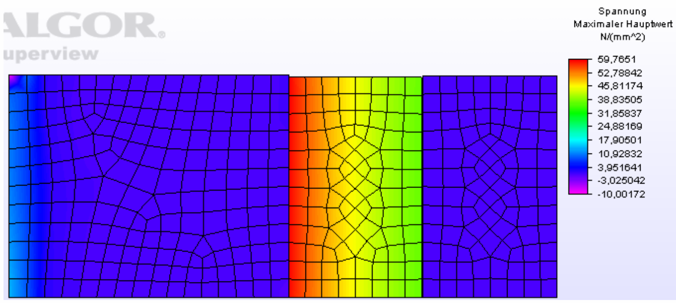

Figure 27: Maximum normal stress in the simplified arrangement at the maximum temperature T2

As described above the calculations have been proven right and therefore the whole assembly will be analysed now. In *Figure 28* and *Figure 29* some results of the analyses.

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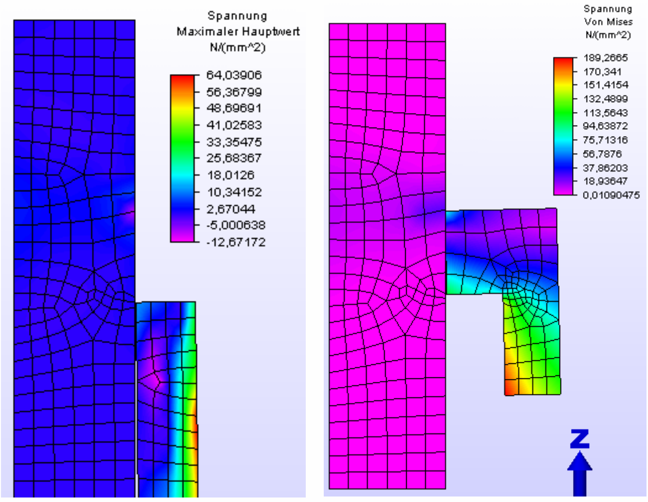

Figure 28: Maximum normal stress (left) and stress according to the Von Mises Theory (right) in the bearing assembly according to this solution at the minimum temperature T1

All four graphs show stresses similar to the calculated ones. Again hardly anywhere much safety is left. Only on the left side of Figure 29, it can be seen that the maximum normal stress with 67 N/mm^2 is lower, than the allowable and calculated stress of 83.4 N/mm^2. This is probably caused by the additional surface on the outside of the bearing ring through to the press fit on the outside that was not considered at this point in the calculations. So the temperature rise could probably be increased a bit but not significantly.
Considering the bearing force in order to ease the stress in the bearing ring does not make sense, since it only acts in a small area normal to the bearing surface.

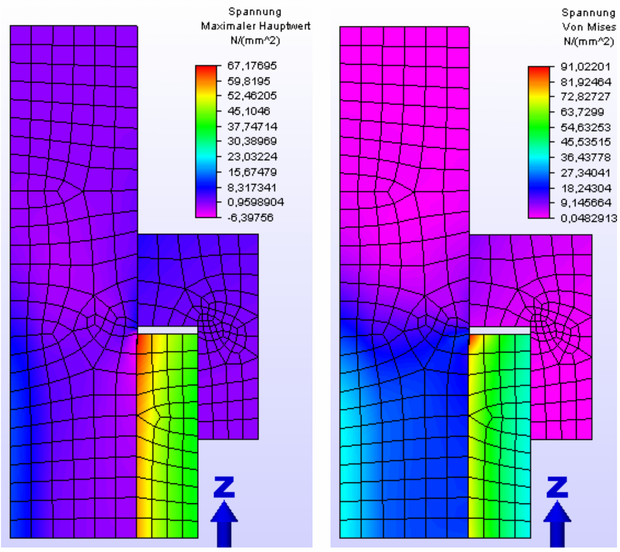

Figure 29: Maximum normal stress (left) and stress according to the Von Mises Theory (right) in the bearing assembly according to this solution at the maximum temperature T2

In contrast to possible increase of the temperature rise for the inner press fit, the temperature rise ΔT_{1-2} has to be reduced since the allowable stress for steel of 185 N/mm^2 is exceeded in the tip of the ring. This can be seen in the right part of Figure 28. Alternatively the shape of the parts can be optimised, for example similar to the solution proposed in [7]. But due to the large difference between the desired temperature range of more than 600 K and the so far calculated temperature range of 90 K, it does not seem to appropriate to add any effort. Therefore it can be said that this solution is not able to fulfil the requirements.

6. Summary

The aim of the paper was to find solutions to the problem of combining physically different materials such as ceramic and metal, particularly due to the different expansion coefficient (17*10^{-6} 1/K for stainless steal und 4*10^{-6} 1/K for SiC) and due to the brittleness of the SiC, which only allows low tensile stresses. In a patent search some solutions have been found. These solutions were described in chapter 2. Legal aspects of the patents were not considered in this paper. The internet and literature search however remained unsuccessful, since the companies for obvious reasons do not publish information as detailed as it was needed here.
In the next step the solutions described were evaluated and two general assemblies of these solutions were selected for further examination. A third solution seemed to be able to fulfil the requirements immediately and therefore was not selected for further examination. During the examination first the developing forces and stresses were calculated with simple calculations. Afterwards these calculations were verified with finite element analyses. During the examining it turned out that the general arrangement with V-groove (see chapters 2.3 to2.5) can fulfil the stated requirements, while the solution with a tight fit on the outside of the bearing ring only allows a very small temperature range. Here changes in the design can help to increase this temperature range slightly, but due to the large difference between the calculated and the desired temperature range it does not seem appropriate to add any effort into this solution.
So it is suggested to use either one of the two solutions that fulfil the requirements or to start a systematic search based on the design process according to Pahl/Beitz to find new solutions. Before using the described solutions that fulfil the stated requirements, the legal aspects of the related patents have to be clarified.

7. Bibliography

[1] Adams, V.; Askenazi, A.: Building better models with Finite Element Analysis. Sante Fe, NM: OnWord Press 1998
[2] CH000000341362A (30.09.1959), **CARO WERK GES MBH**. - Gleitlager
[3] **Deutsches Patent- und Markenamt**. N.N.: Einsteigerrecherche URL: http://depatisnet.dpma.de/DepatisNet/depatisnet?window=1&space=main&content=recherche&action=einsteiger [21.02.2006]
[4] **Deutsches Patent- und Markenamt**. N.N.: Internationale Patentklassifikation URL: http://depatisnet.dpma.de/ipc [21.02.2006]
[5] DE000001068516B (28.04.1960), **Caro-Werk Gesellschaft m.b.H.** - Gleitlager
[6] DE000002000601A (07.01.1971), **LEYLAND GAS TURBINES LTD**. – Wellenlager
[7] DE000004032803C2 (16.10.1990), D.O.R.N.E.Y. Technologie GmbH. – Formteil aus einem metallischen und einem keramischen Körper
[8] DE000009102933U1 (11.07.1991), **Rheinhütte GmbH & Co**. - Wellenlagerung, insbesondere für eine Laufradwelle einer Magnetkreiselpumpe
[9] DE000009419709U1 (15.05.1996), **Sihi GmbH & Co KG**. - Anordnung zum Zentrieren einer Buchse aus zugspannungsempfindlichem Werkstoff auf einer Welle
[10] DE000010045301A1 (04.04.2002), **Allweiler AG**. - GLEITLAGER FUER EINE WELLE
[11] DE000069412616T2 (15.04.1999), **BOFORS AB**. - ANORDNUNG FUER TEMPERATUR-UNABHAENGIGE LAGERUNG IN EINEM LAGER
[12] EP000000264661B1 (25.09.1991), **KRUPP GMBH**. - PLANE BEARING WITH EXPANSION-COMPENSATING ELEMENTS
[13] EP000000280030B1 (22.11.1990), **GKSS-FORSCHUNGSZENTRUM GESTHACHT GMBH**. - Adapterteil
[14] EP000000345214B1 (11.08.1993), **SULZER AG**. - BEARING COMPRISING A SHAFT, PUMP COMPRISING SUCH A BEARING AND USE OF THE PUMP
[15] EP000000563437B1 (06.03.1996), **FEODOR BURGMANN DICHTUNGSWERKE GMBH & CO.** - Anordnung zur konzentrischen Positionierung eines ersten Teiles relativ zu einem zweiten Teil, insbesondere einer Büchse gegenüber einer Welle
[16] EP000000771956B1 (21.06.2000), **KSB AKTIENGESELLSCHAFT**. - Gleitlagerung für eine Maschinenwelle
[17] EP000000771957B1 (31.05.2000), **KSB AKTIENGESELLSCHAFT**. - Gleitlagerung für eine Maschinenwelle
[18] **ESK Ceramics GmbH & Co. KG**. N.N.: Engineered Ceramics. Materials Data. Kempten: 2006
[19] **Fachinformationszentrum Technik e.V.** N.N.: DOMA ® Maschinenbau und Anlagenbau. URL: http://193.97.200.140/cgi-bin/websuche [09.01.2006]
[20] FR000001343721A (22.11.1963), **COMMISSARIAT ENERGIE ATOMIQUE**. - Palier à autocompensation en température
[21] **Gold, P.**: Maschinenelemente. Band I. Aachen: 04/2001
[22] **Google Inc.** N.N.: Google. URL: http://www.google.de [21.02.2006]
[23] **Grote, K.-H.; Feldhusen, J. (Hrsg.)**: Dubbel. Taschenbuch für den Maschinenbau. 21. Aufl. Berlin: Springer 2005
[24] **Institut für Keramische Komponenten im Maschinenbau. N.N.:** Phone call
[25] **Mersch, A.**: Phone Call (25.01.2006)
[26] **Morgan Advanced Materials & Technology, Inc.** N.N.: Typical Properties Grade PS-5000. St. Marys, PA: 2001
[27] **Morgan Advanced Materials & Technology, Inc.** Mersch, A.: Pump Designs. St. Marys, PA: 2005
[28] NL000001027464C (03.01.2005),**CERATEC TECHNICAL CERAMICS B V**. - Ceramic slide bearing for e.g. pumps, comprises ceramic lining mounted in stainless steel cylinder with integral resilient parts
[29] **OTTO JUNKER GMBH**. N.N.: Austenitische Stahlgusswerkstoffe URL: http://www.otto-junker.de/ger/geschaeftsbereiche/tab_austenit.php [21.04.2006]
[30] US000002590761A (25.03.1952), **GEN ELECTRIC**. – Bearing
[31] **Willmann, G.**: Basis für die konstruktive Gestaltung keramischer Bauteile. In: Konstruktion 38 (1986), S. 341-347

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