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Innovative design solutions of universal motor gearbox with helical gears

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Abstract. Due to very force competition, all producer, not only manufacturers of gearboxes, have been forced to constantly improve their products. This has led to the harmonization of the technical characteristics of their gear units, even though they are not yet defined by the standard. Modern solutions of universal helical gearboxes are characterized, first of all, by large torque capacity and high values of gear ratio in the frame of relatively small overall dimensions of the gear unit. This paper provides an overview of common activities undertaken by manufacturers of universal helical gearbox to improve technical characteristics of their products.

1 Introduction

Due to very strong competition, all manufacturers of universal gearboxes seek the way how to create some competitive advantage of their products to improve their product placement. Many manufacturers try to obtain customers by short-time delivery (mostly within 72 hours), while other manufacturers try to attract customers by the high quality of their products. [1, 2] However, some manufacturers who attract customers by low product price (achieved by either cheaper labour, or a more suitable conceptual solution, either slightly lower quality of their reducers). Of course, some manufacturers try to attract customers by an attractive appearance with the same or even higher quality of their gear units.

Each of these actions has certain effects, and as a result the producer is able to survive on the market.

2 Problem description

The main problem that occurs during improving characteristics of universal gearboxes is how to achieve larger load torque capacity and higher value of the highest gear ratio in the same time within the same axis height. Also, the problem is how to reduce the number of components, how to facilitate the way of installation, how to increase universal using of gearboxes, how to make them cheaper and more attractive [3, 4]. The installation of signal and diagnostic components is also one of the challenges.

3 The ways of solving the problem

Within the same axis height and the same axial distance, increasing of torque capacity can be achieved by gear module increasing (size of the gears) which directly causes the teeth number reduction of both gears. Therefore, this action is possible only for small or medium gear ratio values, but with the strengthening of the other gearbox components. However, for large values of gear ratio, since minimum teeth number is 7 or 8, increasing the module is not possible, so these gear ratios are defined by slightly lower load capacity. Another way is to increase the axial distance (for all gear ratios) which certainly results in higher production costs because larger gear wheels are used.

Transferring older gear manufacturing technology to modern production lines provides production of small pinions with only 7 or 8 teeth which give significant positive effect in this improving (Fig. 1-1,2). Increasing of only axial distance can give similar results (Fig. 1-3), but this requires additional expansion of gear unit housing as well as widening of the openings in the housing (to mount large gear wheel). In the case of applying small pinions and larger axial distance, it is possible to significantly increase the largest gear ratio values (Fig. 1-4), which is most commonly done today [3, 4].

For assembling the large gears into the gearbox housing, it is necessary to open the housing from the top to insert the gear (Fig. 2). In this way, the stiffness of the housing is somewhat reduced, so it is necessary to search a way for the structural solution to increase its stiffness. Increasing the housing stiffness can be done by using a robust cover, by using special shape of housing (Fig. 2-1), or by opening only slow-motion housing chamber (Fig. 2-2).

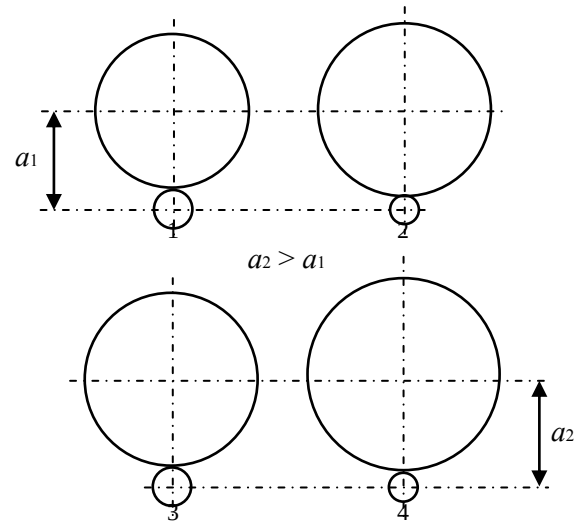


Fig. 1. Characteristic actions for increasing the gear ratio value: reduction of pinion diameter (1-2), increasing of axial distance (3), reduction of pinion diameter and increasing of axial distance (4).

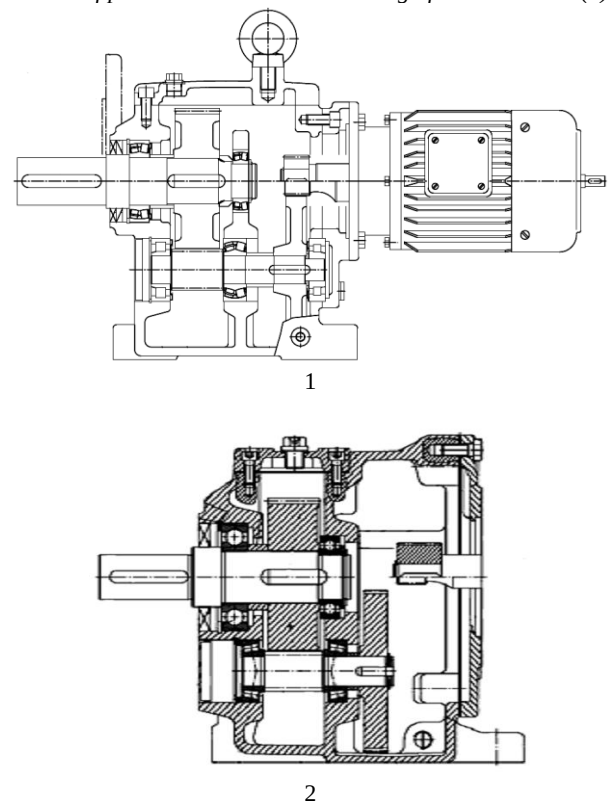


Fig. 2. Structural solution of two-stage motor gearbox with top openings on both chamber (1) and with top opening on slow-motion chamber (2), solution of company Nord [5, 6].

Since within the same axis height and same mounting dimensions, gear units have installed gear wheels with larger axial distance, so they could provide high value of gear ratios. Therefore, the shape of the housing should be modified with additional side extension (Fig. 3) which provides more place for larger gear wheel diameter and increased stiffness of gearbox housing.

In order to reduce the diameter of pinion gear on the shaft of the electric motor, it is necessary to press the pinion into the hollow shaft of the electric motor (Fig. 4). It should not be pressed on the shaft of the electric motor, as it has been the case so far.

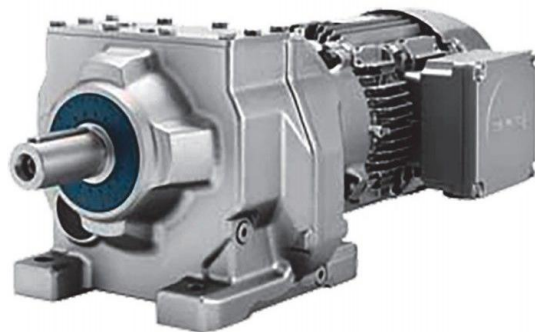


Fig. 3. Structural solution of three-stage motor gearbox with additional side extension, solution of company Siemens [7].

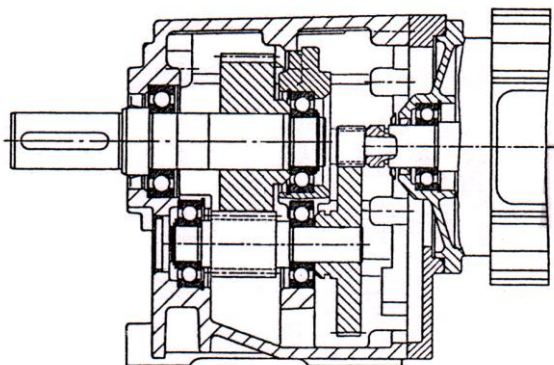


Fig. 4. Structural solution of two-stage motor gearbox with pressed pinion inside the hollow shaft of the electric motor, solution of company Lenze [8, 9].

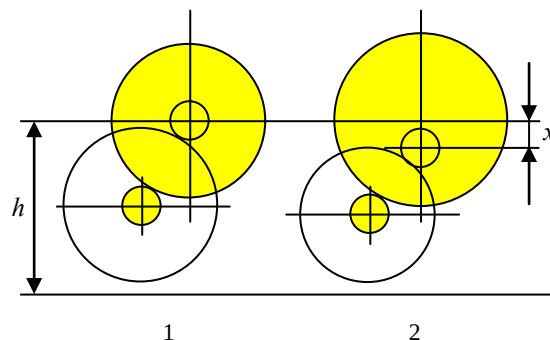


Fig. 5. Schematic display of the influence of gear distance increasing of output gear pair on the reduction of the driven gear diameter of the first pair and obtained axial distance between input and output shaft (x): (1) existing classic solution and (2) the innovative solution.

Increasing the gear distance is required, not only to increase gear ratio but also that torque appeared inside the unit is less as possible. The gear ratio value of output pair must be high to reduce the torque inside the unit which depends on output nominal torque. In that way, significant strenghtness of inside components is not required. Increased gear distance of output gear pair for two-stage and three-stage gearboxes, due to the limited space of axis height, requires a reduction of the driven gear diameter of the first gear pair (Fig. 5-2). If the concept of coaxial input and output shaft has to be maintained, the gear ratio of the first pair will be reduced. However, modern gear units left the concept of the coaxial unit, so the gear distance of the first pair is reduced (Fig. 5-2) and thereby the gear ratio of that pair and the whole unit is increased. [3, 4]

Increasing of the load capacity (T_N) of gear unit certainly requires the reinforcement of all gearbox components, but first of all the output shaft and supported bearings. Further increasing of the load capacity of the gear unit requires finding better materials for gears. While further increasing of the gear ratio per gear pair is limited by the fact that gear units with high gear ratio always use large (expensive) gears. However, in certain segments of gear ratio, it is not justified, since competition can offer a cheaper solution with smaller gear distances. The present values of gear ratio can currently be considered as a limit beyond which it should not go. Technical characteristics of the present solutions of universal gearboxes of leading manufacturers presented in Table 1 confirms this statement.

Table 1. Overview of maximum values of gear ratio and output load capacities (i_{max} / T_N) of gearboxes of leading manufacturers. (Torque value is not related with a given gear ratio)

Axis height- h , mm	Nord [5, 6]		Siemens [7]		SEW [10, 11]	
	Two-stage	Three-stage	Two-stage	Three-stage	Two-stage	Three-stage
90	72.38 / 200	343.92 / 200	41.40 / 140 55.95 / 200	217.89 / 140 235.29 / 200	28.37 / 130 - / 200	135.09 / 130 134.82 / 200
115	54.41 / 430	402.80 / 450	52.14 / 320 59.99 / 450	280.89 / 320 307.02 / 450	33.79 / 300 - / 450	176.88 / 300 186.89 / 450
130	56.65 / 610	362.43 / 640	60.97 / 600	328.97 / 600	- / 600	199.81 / 600
140	26.86 / 820	395.46 / 870	54.47 / 840	330.23 / 840	- / 820	195.24 / 820
180	42.67 / 1600	439.77 / 1700	57.36 / 1680	311.60 / 1680	34.40 / 1550	246.54 / 1550
225	42.76 / 2900	456.77 / 3300	51.17 / 3100	348.88 / 3100	32.05 / 2560	289.74 / 3000

Innovation done by most of the manufacturers of gearboxes is reducing the number of the components or at least reducing the number of expensive components. This is important, especially for smaller manufacturers, since the overall production costs are reduced. This innovation provides such a concept of the gear unit, as well as the

whole row of gearboxes, that will ensure lower costs of manufacturing. Small manufacturers usually produce single-stage or two-stage gearboxes and with their combination get three-stage or four-stage gear units (Fig. 6).

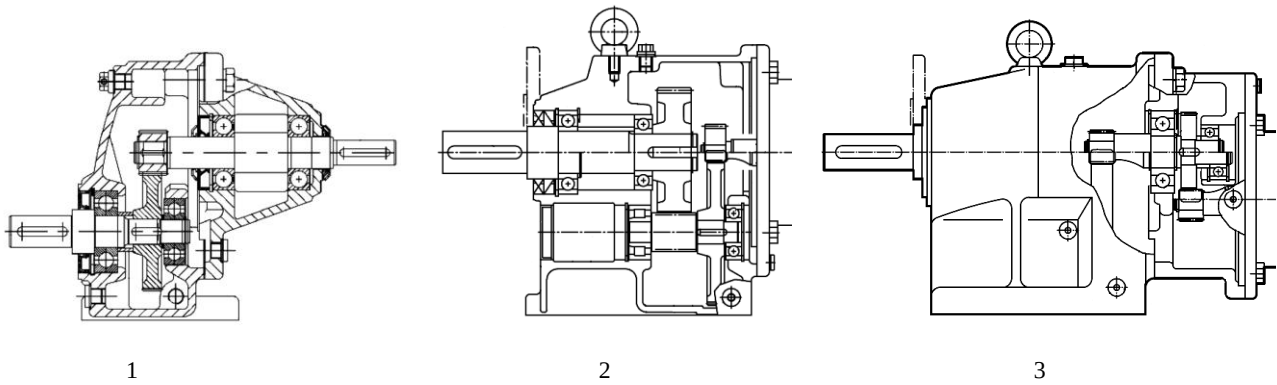


Fig. 6. Innovative solution of single-stage (1), two-stage (2) and three-stage gearbox – built by connecting single-stage and two-stage gear unit (3), solution of company Nord [5, 6].

Large leading manufacturers produce single-stage gearboxes and two-stage and three-stage units assembled in universal housing (Fig. 7). By their combination, they obtain four-stage, five-stage or six-stage gear units. Of course, these manufacturers produce more complex and expensive solutions, although there are manufacturers who produce two sets of the same height of reducer and in such way, they successfully compete with both manufacturers, although this production is most complex and expensive.

Since reducing the production costs can certainly be achieved by reducing the number of expensive components, some smaller gearbox manufacturers produce only one type of universal housing, but with the flange. In that way, using the screw connection and flange, they can connect another housing when they require larger stage variant (Fig. 8).

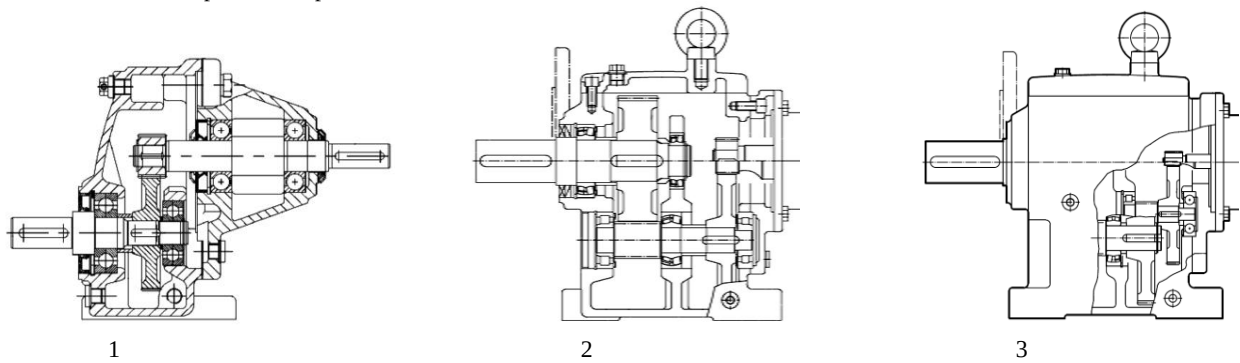


Fig. 7. Innovative solution of single-stage (1), two-stage (2) and three-stage gearbox – built in universal housing for two-stage and three-stage gear unit (3), solution of company Nord [5, 6].



Fig. 8. Innovative solution of gearbox built in universal housing with flange, (1) solution of company Robus [12], (2) solution of company MSF [13].

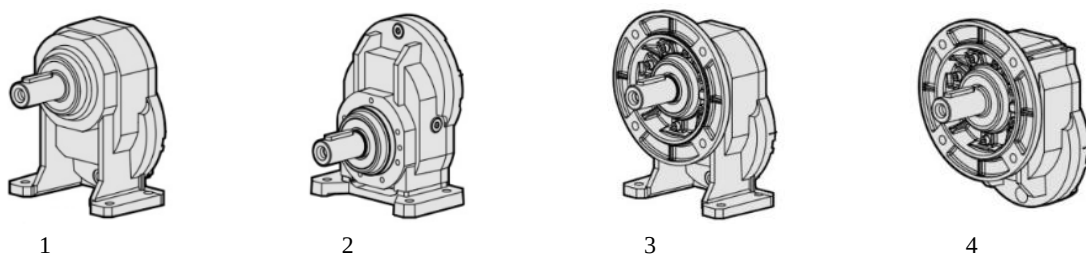


Fig. 9. Innovative solution of single-stage gearbox: (1) foot-mounted and upper output shaft position, (2) foot-mounted and lower output shaft position, (3) foot-mounted with flange and (4) flange gearbox, solution of company Motovario [14].

The greater adaptability of gear units and its components can be achieved first of all by using different mounting shapes and positions (Fig. 9).

Special attention is paid to the possibility of delivering motor gear unit with standard IEC electric motor (Fig. 10-1) which

provides greater adaptability of components, i.e. providing motor is much easier, or in case the customer wishes to purchase and install electric motor separately (Fig. 10-2).

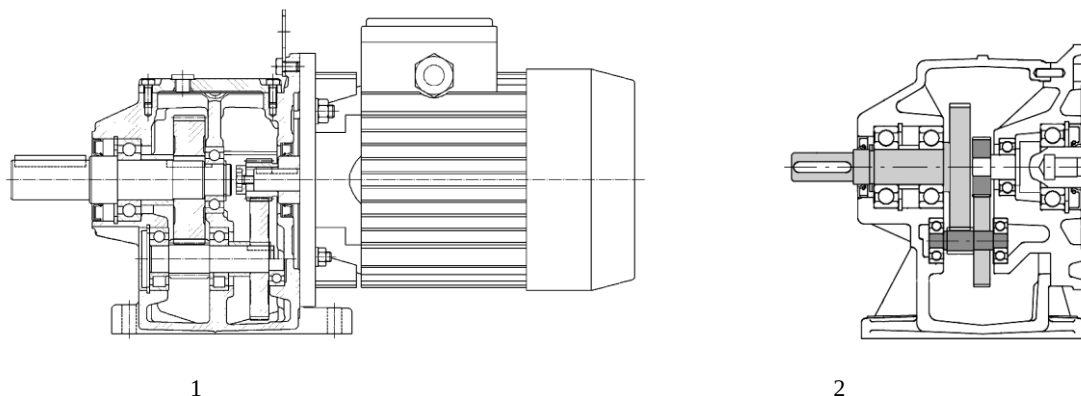


Fig. 10. Innovative solution of motor gearbox: (1) solution of company Rossi [15], (2) solution of company Pujol [16].

Many manufacturers of universal gearboxes want to additionally expand their production program by introducing so-called shaft-mounted gear units (Fig. 11). The installation of these gearboxes is much easier. They do not need large flanges and axial

aligning of an output shaft of reducer with the input shaft of the operating machine. Shaft-mounted gear units offer more compact design, although in some cases they can support input shaft of connected machine.

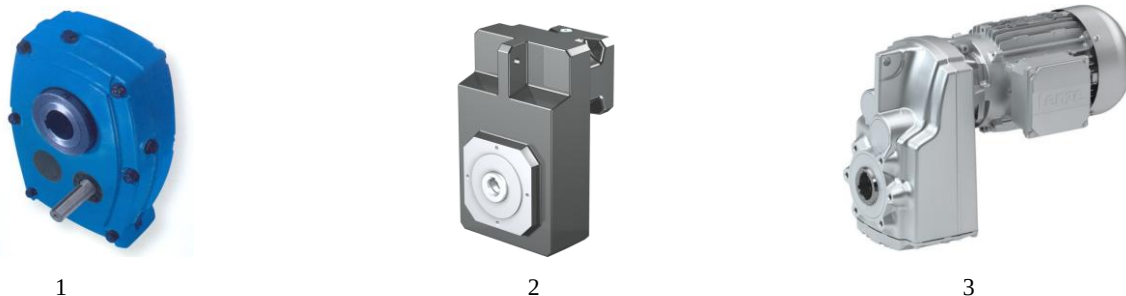


Fig. 11. Innovative solution of shaft-mounted gearbox with axial mounting: (1) solution of company Renold [17], (2) solution of company Stöber [18], (3) solution of company Lenze [8, 9].

Prettier design and more attractive form are always important to improve their placement on the market because the customers like to see the attractive product (Fig. 12 and 13), than the product which is not.

Many manufacturers paint their gearboxes in industrial colours, but also in other colours if there is a special customer's request. These special requirements are additionally paid to customers.

Especially, it should be noted that the largest number of universal gearboxes are today supplied with the so-called regulated drive which adjusts the rotational speed to the specific technological requirements. Usually, these drives have integrated so-called soft starts, which avoids the use of expensive (hydrodynamic) couplings and instead of them they use ordinary couplings.



Fig. 12. Innovative solution of two-stage universal gearbox: (1) solution of company Sesame [19], (2) solution of company Bege [20], (3) solution of company Aokman [21].



Fig. 13. Innovative solution of three-stage universal gearbox: (1) solution of company PGR [22], (2) solution of company ASC [23], (3) solution of company AutomationDirect [24].

Conclusions

Based on the given analysis, it can be concluded that modern solutions of universal gearboxes have fairly uniform technical characteristics. That means these gear units have axis heights produced in a standard row of R20/2, nominal output torques are given in a row R40/6, while the largest gear ratios are given up to 15 for single-stage gearbox, about 50 and more for two-stage gearbox, and about 400 and more for the three-stage gearbox. Additionally, some large manufacturers of universal gear units produce two gear pair sets for the same axis height (large load capacity with lower gear ratio, and smaller load capacity with high gear ratio) in order to successfully compete with small manufacturers of gearboxes. Also, all their gear units are characterized by high quality and attractive form. Such arrangement solutions ensure that almost all customers requirements are fulfilled. However, due to the gearbox market saturation, all the above measures will not produce satisfactory results if it is not required according to the regulations of Industry 4.0. It is necessary to create, so-called, smart product, i.e. gearboxes that will incorporate different sensors, for example, the sensor of temperature, the regulator of oil level, vibration sensor, etc. These additional regulators and sensors will notify the operator or the operating system in the event of reaching limit value. In the case of reaching critical value, they will shut off the driving system and notify the operator accordingly.

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On the kinematics and tooth flank geometry of S_{pr} -gearing

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Abstract. This paper deals with gearing of a novel design. Insensitivity to the gear axes misalignment is the principle feature of gearing of the proposed design. The discussion begins with a brief historical overview on design of geometrically-accurate gearing. Then, fundamentals of gearing that is insensitive to the axes displacements are discussed. Elements of kinematics, an actual configuration of the gear axes of rotation, and so forth are covered in this discussion. Analytical expressions for tooth flanks of a gear and a mating pinion for gearing of the proposed design are derived. Features of interaction of tooth flanks of a gear and a mating pinion are briefly discussed. The paper ends with the analysis of potential applications of " S_{pr} -gearing", and conclusion.

1 A brief historical overview

Transmission of a rotary motion (of a rotation and torque) from an input shaft to an output shaft is the main goal of gears of all designs. For centuries, only low rotations, and low torques were transmitted by means of gears. This is due to the input shaft was driven either by animals, or by water, or by wind, etc. Gearing of these kinds are referred to as "*low-power-density gearing*" (or just as "*LPD-gearing*", for simplicity). Here and below the term "*power-density*" stands for a ratio of a power being transmitting by gears to the gearing mass (sometimes the term "*power-to-weight ratio*" is used instead of the term "*power-density*").

As the "*power-density*" grows up, gearing has to meet additional requirements: a steady rotation of an input shaft has to be smoothly transmitted to a steady rotation of an output shaft (that is, no fluctuation of rotary motion of an output shaft is permissible).

The problem of smooth transmission of a steady rotation by means of parallel-axes gearing (or just by "*P_a-gearing*", for simplicity) has been solved in the 18th century by *Leonhard Euler*. *Euler* proposed [1], [2] involute tooth profile for gears that operate on parallel axes of rotation of a gear and a mating pinion. The proposed by *Euler* gearing is referred to as "*Euler gearing*" (or just as "*Eu-gearing*", for simplicity). Gears according to *Euler* feature equal base pitches. Therefore, "*Eu-gears*" are a kind of "*geometrically-accurate gears*", and, thus, a smooth rotation can be transmitted by means of gears of this kind.

This was *George Grant* who has solved the problem of smooth transmission of a steady rotation by means of intersected-axes gearing (or just by "*I_a-gearing*", for simplicity). In 1887 *Grant* has filed an invention disclosure titled "*Machine for Planing Gear Teeth*" [3]. The proposed by *Grant* gearing is referred to as "*Grant gearing*" (or just as "*Gr-gearing*", for simplicity). Bevel gears according to *Grant* feature equal angular base pitches. Therefore, gears of this design are a kind of "*geometrically-accurate gears*", and, thus, a smooth rotation can be transmitted by means of "*Gr-gearing*".

The problem of smooth transmission of a steady rotation by means of crossed-axes gearing (or just by "*C_a-gearing*", for simplicity) has been solved at around ~2008 by *Radzevich* [4]. Gearing of the proposed design is referred to as "*R-gearing*". "*R-gearing*" features equal angular base pitches. Therefore, "*R-gears*" is a kind of "*geometrically-accurate gears*", and, thus, a smooth rotation can be transmitted by means of gears of these design [5].

Line contact between tooth flanks of a gear, $\mathcal{G}$, and a mating pinion, $\mathcal{P}$, is observed in gearing of all of three kinds, namely, in "*Eu-gearing*", in "*Gr-gearing*", as well, as in "*R-gearing*". This makes gearing of all the designs sensitive to the gear axes displacements, namely, to the linear, and the angular displacements of the axes of rotation of a gear and its mating pinion from their desirable location and orientation.

The main goal of the research reported below is the development of a gear system that is insensitive to the gears' axes misalignment.

2 Fundamentals of gearing insensitive to gear axes misalignment

Compared to gears with point contact, gears with line contact between tooth flanks of a gear, $\mathcal{G}$, and a mating pinion, $\mathcal{P}$, allows for higher power density ("*high-power-density gearing*", or just "*HPD-gearing*", for simplicity). "*HPD-gearing*" are smaller in size, lighter in weight, and are capable of transmitting a required power (rotation and torque). All the industries are concerned with the development of gear transmissions that feature the highest possible "*power-density*".

Line contact between tooth flanks of a gear, $\mathcal{G}$, and a mating pinion, $\mathcal{P}$, is desirable as long as no linear and angular displacements of the gears' axes of rotation are taken into account. Unfortunately, in reality no gearing with zero axes misalignment are possible, as the axes misalignment is inevitable.

In order to accommodate for the gears' axes misalignment in gearing, line contact between a gear, $\mathcal{G}$, and a mating pinion, $\mathcal{P}$, tooth flanks is required to be substituted with point contact of their tooth flanks. When doing that, it is desirable to sacrifice the gear pair bearing capacity as little as possible. In this scenario, a gear pair gets capable of accommodating for the gear and the mating pinion axes displacements, as well as it features the highest possible power density. The proposed novel design of gears is referred to as "*S_{pr}-gearing*".

2.1 Elements of kinematics of gear pair

Vector diagram of a gear pair is a convenient mean for the analysis of kinematics of gearing ("*P_a-gearing*", "*I_a-gearing*", as well as "*C_a-gearing*"). In order to construct the vector diagram for a given gear pair, rotations of a gear, and that of a mating pinion, are required to be represented in vector form.

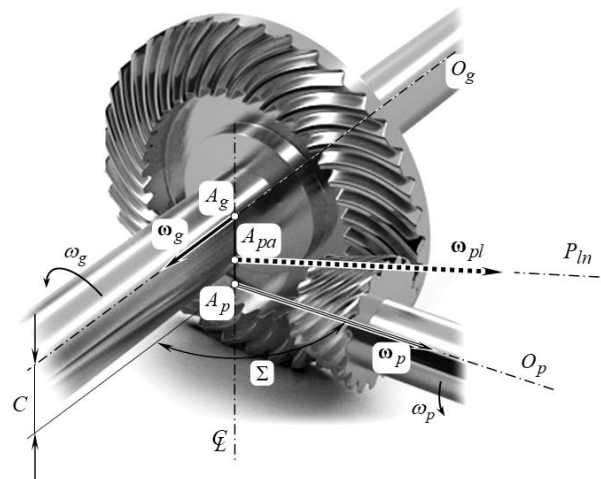


Fig. 1. Elements of kinematics of gear pair.

Note: Rotation is not a vector by nature. Therefore, special care is required to be taken when treating rotations as vectors.

An example of the vector diagram for a crossed-axes gear pair is depicted in Fig. 1.

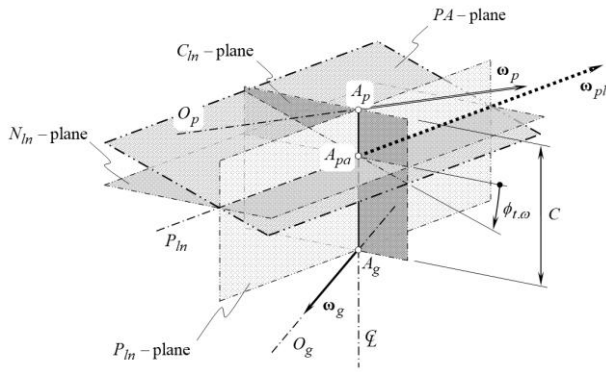


Fig. 2. Principal planes associated with vector diagram of a gear pair (along with the plane of action).

The rotation vectors, ω_g and ω_p , of a gear and its mating pinion, are pointed along the gear, O_g , and the pinion, O_p , axes of rotation. The centre-line, $\mathcal{L}$, is pointed along the closest distance of approach between the crossing axes, O_g and O_p . The centre-distance (the closest distance of approach between the axes O_g and O_p) is designated as C . The shaft angle, Σ , is formed by the rotation vectors, ω_g and ω_p :

$$\Sigma = \angle(\omega_g, \omega_p) \quad (1)$$

It is assumed here that the gear is motionless, and the pinion performs an instant rotation about the axis of instant rotation, P_{ln} .

The vector of instant rotation, ω_{pl} , is specified by the gear ratio, u , and by an expression:

$$\omega_{pl} = \omega_g - \omega_p \quad (2)$$

The following three principal planes are associated with a gear vector diagram (see Fig. 2):

- the pitch-line plane (P_{ln} - plane)
- the center-line plane (C_{ln} - plane)
- the normal plane (N_{ln} - plane)

The plane of action (PA -plane) complements the set of three principal planes to a full set of planes, associated with a gear vector diagram.

A set of the principal planes associated with the corresponding vector diagram is an efficient mean for the analysis and analytical representation of a gear pair when developing gearing of a novel design [4].

2.2 Actual configuration of gears' axes of rotation

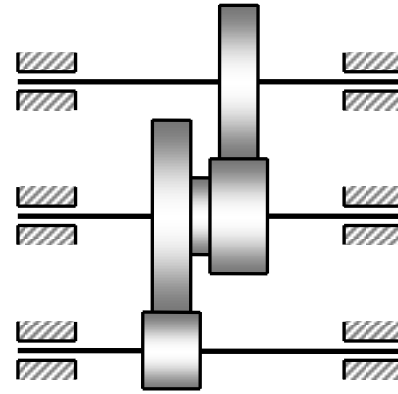
Numerous sources of gears' axes misalignment are observed in real gearing. Inaccuracies of the gear housing, thermal extension of the components of the gearbox, deflections of the gear shafts under operating load, and so forth, are among them.

As an example, the gear axes misalignment due to deflection of the gear shafts under operating load is briefly considered immediately below.

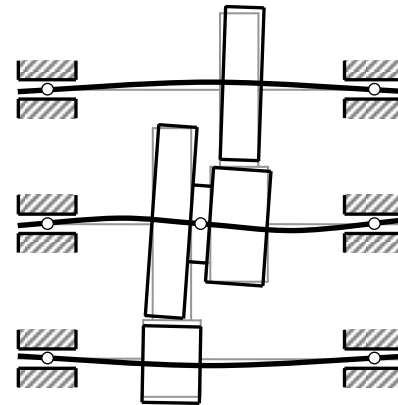
Gear pairs are designed so as to keep axes of rotation of a gear and a mating pinion aligned, as schematically illustrated in Fig. 3.a. When an operated load is applied, the gear shafts got bent and twisted as shown in Fig. 3.b. In the deformed stage gears are rotated about instant axes of rotation. The instant axis of gear rotation (see Fig. 4) can be construed as straight line, tangent to the gear neutral axis at point that corresponds to the middle of the gear face width.

One possible way to determine an actual configuration of the gear instant axis of rotation is to calculate the first derivative of

equation of the neutral axis of the gear shaft. CAD/CAM tools and FEA can be used to determine the anticipated displacements and



(a)



(b)

Fig. 3. The axes alignment in parallel-axes gearing: (a) perfect alignment, and (b) misalignment under operating load.

deformations in a gear drive under an operating load. Generally speaking, the axes of instant rotation of two gears can be viewed as two skewed straight lines.

Ultimately, an actual configuration of instant axes of rotation of two mating gears can be specified just by two design parameters, namely, by an actual value of the closest distance of approach, C^r ,

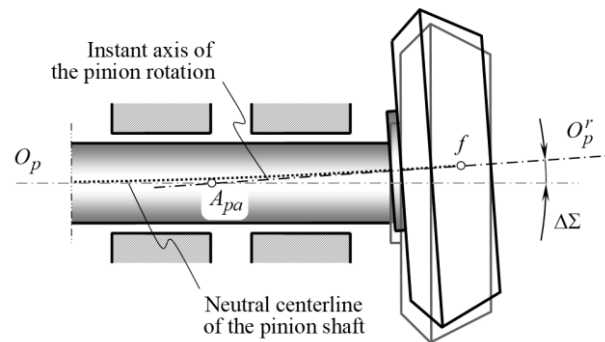


Fig. 4. Shaft deflection of overhung bevel pinion under operating load.

of the instant axes of rotation, and by their crossing angle, Σ^r .

An actual value of the closest distance of approach, C^r , of the instant axes of rotation, is a function of the design parameters, C and Σ , in the gear pair with zero axes misalignments, and of the

tolerances for the total linear, $\{\delta\}$, and the total angular, $\{\varphi\}$, displacements of the axes of rotation: $C^r(C, \Sigma, \{\delta\}, \{\varphi\})$.

In a displaced configuration of a gear and its mating pinion, an expression:

$$C^r = \left| (\mathbf{r}_{og}^0 - \mathbf{r}_{op}^0) + \frac{(be - cd)\mathbf{u}_g - (ae - bd)\mathbf{v}_p}{ac - b^2} \right| \quad (3)$$

can be used for the calculation of an actual value of the closest distance of approach, C^r , of the instant axes of rotation [4], [5].

Here, in Eq. (3), is designated: $a = \mathbf{u}_g \cdot \mathbf{u}_g$, $b = \mathbf{u}_g \cdot \mathbf{v}_p$, $c = \mathbf{v}_p \cdot \mathbf{v}_p$, $d = \mathbf{u}_g \cdot \mathbf{d}_0$, and $e = \mathbf{v}_p \cdot \mathbf{d}_0$. The rest of the design parameters in Eq. (3), are explained in Fig. 5.

The distance, C^r , can be expressed in terms of the distance, C , and of the tolerances for the maximal permissible linear displacements of the gear and the pinion axes of rotation:

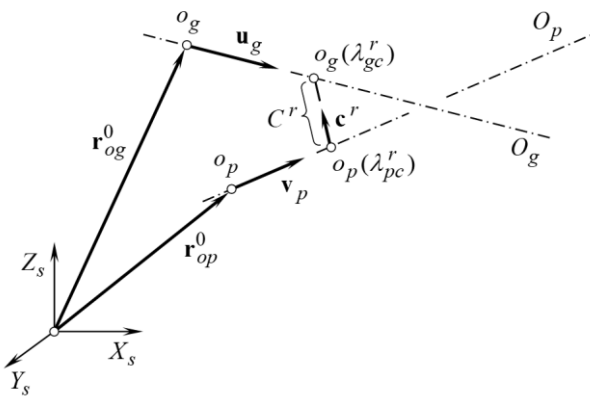


Fig. 5. Calculation of the total displacement along the center-distance in a gear pair.

$$\{\delta_g\} = \begin{bmatrix} \{\delta_{gx}\} \\ \{\delta_{gy}\} \\ \{\delta_{gz}\} \\ 1 \end{bmatrix}, \quad \{\delta_p\} = \begin{bmatrix} \{\delta_{px}\} \\ \{\delta_{py}\} \\ \{\delta_{pz}\} \\ 1 \end{bmatrix} \quad (4)$$

The crossed-axes angle, Σ^r , between the axes of rotation, O_g^r and O_p^r , is a function of that same design parameters, C , Σ , $\{\delta\}$, and $\{\varphi\}$: $\Sigma^r(C, \Sigma, \{\delta\}, \{\varphi\})$. This angle, Σ^r , can be calculated from the following expression:

$$\Sigma^r = \tan^{-1} \left(\frac{|\mathbf{u}_g \times \mathbf{v}_p|}{\mathbf{u}_g \cdot \mathbf{v}_p} \right) \quad (5)$$

The crossed-axes angle, Σ^r , can be expressed in terms of the Euler's angles (the matrix of Euler's coordinate systems transformation is not presented here as it is too bulky) [4], [5].

2.3 Tooth flank geometry in S_{pr} -gearing

As it is already mentioned above, S_{pr} -gearing is a kind of geometrically-accurate gearing. Therefore, the tooth flank geometry has to be derived so as to ensure smooth transmission of a uniform rotation from a driving shaft to a driven shaft. Taken in a whole, tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, of a gear and a mating pinion are neither enveloping, nor conjugate to one another. However, because of point contact between the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, at every instant of time (a) a straight line along the contact perpendicular, $\mathbf{n}_g$, always intersects the axis of instant rotation,

P_{ln} , and (b) instant values of base pitches of a gear and a mating pinion, both, are equal to instant value of an operating base pitch of the gear pair.

For convenience, particular reference systems are preferred to be used when deriving equations for the gear, $\mathcal{G}$, and the mating pinion, $\mathcal{P}$, tooth flanks, correspondingly.

2.3.1 Adopted concept of tooth flank geometry in S_{pr} -gearing

In gears for geometrically-accurate gearing, the tooth flanks are generated by means of a desirable line of contact, LC_{des} .

Note: The tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, are not generated yet, but a desirable line of contact, LC_{des} , can be already specified.

The line of contact, LC_{des} , is specified under an assumption that all the displacements, both, the linear displacements, as well as the angular displacements, are of a zero value. Specified this way, the tooth flank in a gear for " S_{pr} -gear system" is identical to that in " R -gearing" [5]. Then an impact of the displacements onto the geometry of a gear and a mating pinion tooth flanks is incorporated.

For this purpose, when the gears rotate, the line of contact, LC_{des} , is considered as rotated together with the plane of action, PA , that is, the line of contact is motionless in relation to the plane of action. However, configuration of the line of contact in relation to the reference systems, $X_gY_gZ_g$ and $X_pY_pZ_p$, associated with the gear, and the mating pinion, depend on the actual values of axes misalignment in the gear pair. The axes misalignment in the gear pair is kept within the pre-specified tolerances for the accuracy of the axes displacements.

All possible configurations of the plane of action, PA , in relation to the reference systems, $X_gY_gZ_g$ and $X_pY_pZ_p$, are taken into account. The plane of action, PA , is rotated about the axis, O_{pa} , under all possible values of the displacements, both, linear and angular, in terms of which the axes displacements can be expressed. All possible configurations of the line of contact, LC_{des} , in relation to the reference systems, $X_gY_gZ_g$ and $X_pY_pZ_p$, are covered this way. Ultimately, a gear tooth flank, $\mathcal{G}$, is generated in a reference system, $X_gY_gZ_g$ (which the gear will be associated with), as a locus of lines of contact, LC_{des} . Similarly, a mating pinion tooth flank, $\mathcal{P}$, is generated in a reference system, $X_pY_pZ_p$ (which the pinion will be associated with), as a locus of that same line of contact, LC_{des} . Regardless of the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, of a gear and a mating pinion are generated by that same line of contact, LC_{des} , the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, in " S_{pr} -gearing" always make point contact with one another. The degree of conformity at point of contact between the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, in " S_{pr} -gearing" is always the highest possible.

The concept, adopted for the tooth flank generation of gears in " S_{pr} -gear system", can be viewed in alternative manner.

As an example, consider a plane of action, PA , in a geometrically-accurate parallel-axes gear pair as schematically shown in Fig. 6. An effective face width of the gear pair is designated as F_{pa} . Then, a desired line of contact, LC_{des} , in the nominal configuration of the axes of rotation, O_g and O_p , is constructed. After that two extremal configurations of the line of contact, LC_{des} , are constructed as well. One of the extremal configurations corresponds to the maximal positive, and another

one corresponds to the extremal negative deviation of the desired line of contact, LC_{des} , from its nominal configuration. All three

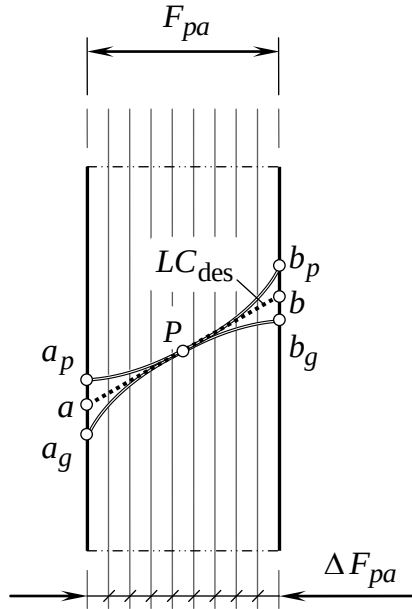


Fig. 6. On the adopted concept of tooth flank generation in gears for “ S_{pr} – gear system”.

the desired lines of contact are located within the plane of action, PA .

The plane of action, PA , can be sliced on an infinite number of slices, each of which is perpendicular to the axis of instant rotation, P_{ln} . Width of each elementary slice is designated as ΔF_{pa} .

The elementary lines of contact, lc_g and lc_p , for the gear and the mating pinion, are constructed so as for a geometrically-accurate crossed-axes gear pair having maximal linear, $\{\delta_g\}$ and $\{\delta_p\}$, and maximal angular, $\{\varphi_g\}$ and $\{\varphi_p\}$, displacements of the gear axis of rotation, O_g , and that, O_p , of the pinion from their nominal configuration in relation to the plane of action, PA . The displacements, $\{\delta_g\}$ and $\{\varphi_g\}$, are in nature the tolerances, $[\delta_g]$ and $[\varphi_g]$, for the maximal elementary displacements correspondingly.

For the slice, ΔF_{pa} , at the middle of the gear pair face, the elementary lines of contact, lc_g and lc_p , are constructed so as for a geometrically-accurate crossed-axes gear pair having zero linear, $\{\delta_g\}$ and $\{\delta_p\}$, and angular, $\{\varphi_g\}$ and $\{\varphi_p\}$, displacements in relation to the plane of action, PA .

For the rest of the slices, ΔF_{pa} , in between the middle slice, and the slices at the both ends of the face, the elementary lines of contact, lc_g and lc_p , are constructed similar to they are constructed for the perfect crossed-axes gear pair having intermediate values of the linear, $\{\delta_g\}$ and $\{\delta_p\}$, and the angular, $\{\varphi_g\}$ and $\{\varphi_p\}$, displacements in relation to the plane of action, PA .

It is right point to stress here that the instant plane-of-action apex, A_{pa} , as well as the instant base-cone-apexes, A_g and A_p , of a gear and a mating pinion occupy different locations for different slices. The central point, P , is remained stationary. There is certain freedom for the gear designer to chose a function of

distribution of the elementary displacements, $[\delta_g]$, $[\varphi_g]$, and $[\delta_p]$, $[\varphi_p]$, within the face-width, F_{pa} . The Gauss' (or normal) distribution is the first one to be considered in this regard.

Two curves, a_gPc_g and a_pPc_p , are generated as envelope curves to corresponding family of the elementary lines of contact, lc_g and lc_p , when the gears rotate.

The tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, are generated by these two curves, a_gPc_g and a_pPc_p , one of which is constructed for the gear, and another one is constructed for the pinion.

2.3.2 Preferred reference system

Prior to begin the derivation of an equation for the gear tooth flank, $\mathcal{G}$, and for its mating pinion tooth flank, $\mathcal{P}$, the following three tolerances are require to be specified :

- the tolerance, $\{C\}$, for the center distance, C
- the tolerance, $\{\Sigma\}$, for the crossed-axes angle, Σ
- the tolerance, $\{A\}$, for the axial location of the gear in the direction of the gear axial vector, $\mathbf{A}_g$.

The actual value of the center distance, C , is within an interval:

$$C - \{C_-\} \leq C \leq C + \{C^+\} \quad (6)$$

where $\{C^+\}$ is the upper, and $\{C_-\}$ is the lower maximum permissible deviations of the center distance, C .

The actual value of the crossed-axes angle, Σ , is within an interval:

$$\Sigma - \{\Sigma_-\} \leq \Sigma \leq \Sigma + \{\Sigma^+\} \quad (7)$$

where $\{\Sigma^+\}$ is the upper, and $\{\Sigma_-\}$ is the lower maximum allowed deviations of the crossed-axes angle, Σ .

The actual value of the axial distance, A , is within an interval:

$$A - \{A_-\} \leq A \leq A + \{A^+\} \quad (8)$$

where $\{A^+\}$ is the upper, and $\{A_-\}$ is the lower maximum allowed deviations in the direction of the gear axial vector, $\mathbf{A}_g$.

To perform the analysis, a convenient reference system can be determined in a few steps illustrated in Fig. 7.

First, for a nominal configuration of the rotation vectors, all the elements of the corresponding vector diagram are predetermined by a configuration of the rotation vectors, ω_g and ω_p . The vector diagram can be constructed in an arbitrary reference system, in a reference system, $X_h Y_h Z_h$, associated with the housing of the gear pair, in particular.

Second, in a geometrically-accurate real gear pair all the elements of the vector diagram are displaced from their nominal location and orientation. None of the angular velocity vectors in a perfect real gear pair, ω_g^r , ω_p^r , and ω_{pl}^r , align with the corresponding angular velocity vectors, ω_g , ω_p , and ω_{pl} , constructed for their nominal configuration. Moreover, the gear base-cone-apex, A_g^r , the pinion base-cone-apex, A_p^r , and the plane-of-action apex, A_{pa}^r , all of them are displaced from their nominal positions, labeled as A_g , A_p , and A_{pa} , correspondingly.

Ultimately, the center distance, C^r , the crossed-axes angle, Σ^r , and the pressure angle, $\phi_{t,\omega}^r$, in a perfect real gear pair differ from that, C , Σ , and $\phi_{t,\omega}$, constructed for their nominal configuration.

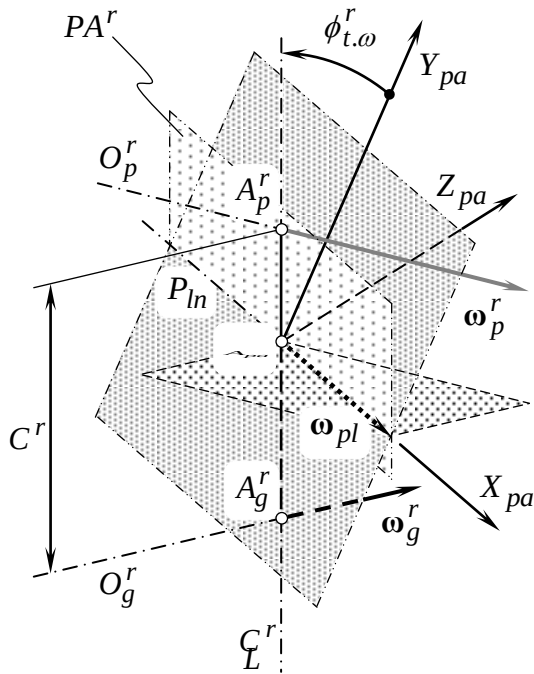


Fig. 7. Vector diagram, and a reference system associated with the plane of action, PA .

Permissible deviations of all the displacements in a perfect real gear pair in relation to their nominal values are limited by:

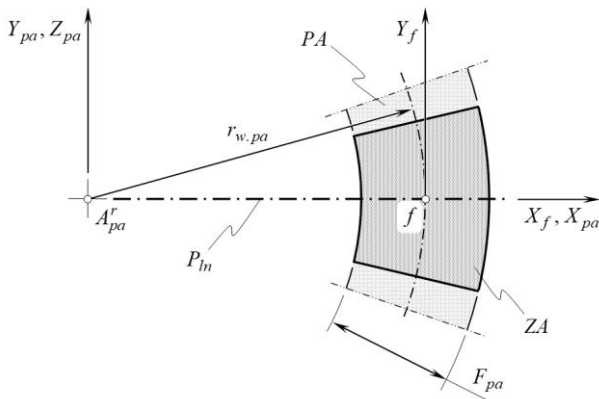


Fig. 8. The Cartesian reference system, $X_f Y_f Z_f$, associated with the plane of action, PA .

- the tolerance, $\{C\}$, for the accuracy of the center distance, C
- the tolerance, $\{\Sigma\}$, for the accuracy of the crossed-axes angle, Σ
- the tolerance, $\{A\}$, for the accuracy of the gear axial distance, A

The displacements could be more visible in a case, when both the vector diagrams are constructed together so as to overlap one another.

Note: All the displacements are of small values (therefore, it is not easy to visualize them).

Third, a reference system, $X_{pa} Y_{pa} Z_{pa}$, can be associated with the plane of action, PA . This reference system is originated at the plane-of-action apex, A_{pa} , and the axis, X_{pa} , is pointed along the vector of instant rotation, ω_{pl} , as illustrated in Fig. 7. In this case an advantage can be taken from the fact that the

configurations of the following elements of the vector diagram, namely:

- of the plane-of-action apex, A_{pa}
- of the vector of instant rotation, ω_{pl}
- of the center-line, L ,

are not altered under the manufacturing errors, as well as under the displacements caused by other reasons.

Ultimately, in the case under consideration, the plane of action, PA , becomes stationary.

Fourth, it is assumed below that when a gear is displaced from its nominal position and orientation, the position of the middle, f , of the effective face width, F_{pa} , do not altered within the axis of instant rotation, P_{ln} , and is remained the same. This point can be chosen as the origin of a Cartesian reference system, $X_f Y_f Z_f$, associated with the plane of action, PA , as illustrated in Fig. 8. The reference system, $X_f Y_f Z_f$, is shifted along the axis, X_{pa} , with respect to the reference system, $X_{pa} Y_{pa} Z_{pa}$, at a distance, $r_{w,pa}$, and turned about this axis through the transverse pressure angle, $\phi_{t,\omega}$.

The motionless reference system, $X_f Y_f Z_f$, is adopted below as a main reference system to perform the analysis. It does not alter its configuration with respect to the gear housing when the gear displaces from its nominal location and orientation because of the manufacturing errors, under an operating load, because of the heat extension of the gear housing, as well as of the rest of components of the gear transmission, and so forth.

2.3.3 Analytical description of tooth flank geometry in S_{pr} -gearing
Tooth flanks of a gear, and a mating pinion in " S_{pr} -gearing" are generated by menas of a desirable line of contact, LC_{des} , between the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$. Regardless of the tooth flanks are not generated yet at this point, a desirable line of contact can be constructed. The line of contact, LC_{des} , is a planar curve that is entirely located within the plane of action, PA .

Note: In gearing of special design, a desirable line of contact, LC_{des} , can be shaped in a form of a spatial curve [5].

When generating the tooth flanks, the desired line of contact travels together with the plane of action, PA .

In the reference system, $X_f Y_f Z_f$, associated with the plane of action, PA , position vector of a point, $\mathbf{r}_{lc}^{pa}$, of the line of contact, LC_{des} , can be analytically described in matrix form as:

$$\mathbf{r}_{lc}^{pa}(r_{pa}) = \begin{bmatrix} X_{lc}^{pa}(r_{pa}) \\ Y_{lc}^{pa}(r_{pa}) \\ 0 \\ 1 \end{bmatrix} \quad (9)$$

Here, in Eq. (9), r_{pa} designates a current value of the distance at which point, f , is remote from a current plane-of-action apex, $A_{pa}^{r,cur}$. For a specified gear pair, the distance, r_{pa} , is within the interval:

$$r_{w,pa} - \{\Delta r_{pa}^{(-)}\} \leq r_{pa} \leq r_{w,pa} + \{\Delta r_{pa}^{(+)}\} \quad (10)$$

Considered in the reference system, $X_f Y_f Z_f$, the line of contact, LC_{des} , remains stationary when the distance, r_{pa} , alters

from its minimum value, $r_{w,pa} - \{\Delta r_{pa}^{(-)}\}$, to its maximum value, $r_{w,pa} + \{\Delta r_{pa}^{(+)}\}$. However, current location of the plane-of-action apex, $A_{pa}^{r,cur}$, travels within the interval specified by Eq. (10), that is, it travels between two extremal positions, $A_{pa}^{r,min}$ and $A_{pa}^{r,max}$. An operator, $\mathbf{Rs}(f \mapsto pa)$, of the resultant coordinate system transformation can be used for the analytical description of transition from the *Cartesian* coordinate system, $X_f Y_f Z_f$, to the reference system, $X_{pa} Y_{pa} Z_{pa}$:

$$\mathbf{Rs}(f \mapsto pa) = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos \phi_{t,\omega} & \sin \phi_{t,\omega} & 0 \\ 0 & -\sin \phi_{t,\omega} & \cos \phi_{t,\omega} & -r_{pa} \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (11)$$

It is important to stress here that the operator, $\mathbf{Rs}(f \rightarrow pa)$, is a function of a current value of the distance, r_{pa} .

Further, the operator, $\mathbf{Rs}(PA \mapsto \mathcal{J})$, of the resultant coordinate system transformation can be used for the analytical description of transition from the *Cartesian* coordinate system, $X_{pa} Y_{pa} Z_{pa}$, associated with the plane of action, PA , to the *Cartesian* coordinate system, $X_g Y_g Z_g$, associated with the gear. Again, in the case of “ S_{pr} -gearing”, the operator, $\mathbf{Rs}(PA \mapsto \mathcal{J})$, is a function of a current value of the distance, r_{pa} .

With that said, for the purpose of analytical description of transition from the *Cartesian* coordinate system, $X_f Y_f Z_f$, to the coordinate system, $X_g Y_g Z_g$, associated with the gear, the operator, $\mathbf{Rs}(f \mapsto \mathcal{J})$, can be used. The operator, $\mathbf{Rs}(f \mapsto \mathcal{J})$, equals to the product:

$$\mathbf{Rs}(f \mapsto \mathcal{J}) = \mathbf{Rs}(PA \mapsto \mathcal{J}) \cdot \mathbf{Rs}(f \rightarrow pa) \quad (12)$$

Once the operators, $\mathbf{Rs}(f \rightarrow pa)$ and $\mathbf{Rs}(PA \mapsto \mathcal{J})$, are functions of a current value of the distance, r_{pa} , the operator, $\mathbf{Rs}(f \mapsto \mathcal{J})$, is also a function of this design parameter of the gear pair. Below this function is expressed in terms of the parameter, $X_f|_{A_{pa}^{inst}}$, which is the X -coordinate of the plane-of-action apex, A_{pa}^{inst} .

For the purpose of analytical description of a gear tooth flank, $\mathcal{J}$, an equation can be used:

$$\mathbf{r}_g^r(r_{pa}, \varphi_p, X_f|_{A_{pa}^{inst}}) = \mathbf{Rs}(f \mapsto \mathcal{J}) \cdot \mathbf{r}_{lc}^{pa}(r_{pa}) \quad (13)$$

The gear tooth flank, $\mathcal{J}$, in “ S_{pr} -gearing” is a surface of three independent parameters. An instant value of the radius, r_{pa} , of point of the line of contact, LC_{des} , is the first of three parameters. An instant value of the angle, φ_p , through which the input shaft is turned about the pinion axis of rotation, O_p , is the second parameter [the operator, $\mathbf{Rs}(f \mapsto \mathcal{J})$, of the resultant coordinate system transformation is a function of the angle, φ_p],

while the coordinate, $X_f|_{A_{pa}^{inst}}$, of the instant plane-of-action apex, A_{pa}^{inst} , is the third of three parameters.

Similarly, for a pinion tooth flank, $\mathcal{P}$, a corresponding equation is derived:

$$\mathbf{r}_p^r(r_{pa}, \varphi_p, X_f|_{A_{pa}^{inst}}) = \mathbf{Rs}(f \mapsto \mathcal{P}) \cdot \mathbf{r}_{lc}^{pa}(r_{pa}) \quad (14)$$

Here, the operator, $\mathbf{Rs}(f \mapsto \mathcal{P})$, of the transition from the reference system, $X_f Y_f Z_f$, associated with the plane of action, PA , to the coordinate system, $X_p Y_p Z_p$, associated with the pinion, equals to the product:

$$\mathbf{Rs}(f \mapsto \mathcal{P}) = \mathbf{Rs}(PA \mapsto \mathcal{P}) \cdot \mathbf{Rs}(f \rightarrow pa) \quad (15)$$

The pinion tooth flank, $\mathcal{P}$, in “ S_{pr} -gearing” also is a surface of three independent parameters. An instant value of the radius, r_{pa} , of point of the line of contact, LC_{des} , is the first of three parameters. An instant value of the angle, φ_p , through which the input shaft is turned about the pinion axis of rotation, O_p , is the second parameter [the operator, $\mathbf{Rs}(f \mapsto \mathcal{P})$, of the resultant coordinate system transformation is a function of the angle, φ_p], while the coordinate, $X_f|_{A_{pa}^{inst}}$, of the instant plane-of-action apex, A_{pa}^{inst} , is the third of three parameters.

Three independent parameters, r_{pa} , φ_p , and $X_f|_{A_{pa}^{inst}}$, are excessive for the determination of a surface, in both cases, that is, for the determination of a gear tooth flank, $\mathcal{J}$, as well as of a mating pinion tooth flank, $\mathcal{P}$, as a surface is a space of two dimensions. An advantage can be taken from the available plurality of independent parameters by setting a favorable functional correlation between two of them, namely, between the parameters, φ_p and $X_f|_{A_{pa}^{inst}}$. The function $X_f|_{A_{pa}^{inst}} = X_f|_{A_{pa}^{inst}}(\varphi_p)$ has to be a kind of function to ensure smoothness of the tooth flanks, $\mathcal{J}$ and $\mathcal{P}$. After two of three independent parameters somehow are correlated with one another, then the tooth flanks, $\mathcal{J}$ and $\mathcal{P}$, turns to be functions of two independent parameters. Below, an example of such a functional correlation, $X_f|_{A_{pa}^{inst}} = X_f|_{A_{pa}^{inst}}(\varphi_p)$, between two of three independent parameters is considered.

The operator of the resultant coordinate system transformation, $\mathbf{Rs}(PA \mapsto \mathcal{P})$, is required to be composed.

The operator of the resultant coordinate system transformation, $\mathbf{Rs}(f \mapsto \mathcal{P})$, as well as the operators, $\mathbf{Rs}(f \rightarrow pa)$ and $\mathbf{Rs}(PA \mapsto \mathcal{J})$, are functions of a current value of the distance, r_{pa} .

The distance, r_{pa} , can be expressed in terms of elementary linear displacements, δ_{gx} , δ_{gy} , and δ_{gz} , and elementary angular displacements, φ_{gx} , φ_{gy} , and φ_{gz} , of a gear, together with elementary linear displacements δ_{px} , δ_{py} , and δ_{pz} , and elementary angular displacements, φ_{px} , φ_{py} , and φ_{pz} , of a mating pinion.

The disclosed approach for generating a gear, $\mathcal{G}$, and a mating pinion, $\mathcal{P}$, tooth flanks can be qualitatively summarized in the following way. Consider that the plane-of-action apex, $A_{pa}^{r.cur}$, at its initial location coincides with the point, $A_{pa}^{r.min}$. For such a configuration of the axes of rotation, O_g and O_p , of a gear and a mating pinion, the tooth flanks, $\mathcal{G}_{min}$ and $\mathcal{P}_{min}$, of an “ R -gearing” can be generated [4]. Then the apex, $A_{pa}^{r.cur}$, is shifted toward the plane-of-action apex, $A_{pa}^{r.max}$, at a reasonably short distance (or at an infinitesimally short distance). As the operators of the resultant coordinate systems transformation are functions of current value of the distance, r_{pa} , the initial configuration of the axes of rotation, O_g and O_p , is altered under such a shift. A novel “ R -gearing” can be generated with a new configuration of the axes of rotation, O_g and O_p . Gradually shifting the apex, $A_{pa}^{r.max}$, toward the plane-of-action apex, $A_{pa}^{r.max}$, a set of tooth flanks, $\mathcal{G}_{cur}$ and $\mathcal{P}_{cur}$, of different “ R -gearings” is generated. Finally, when the plane-of-action apex, $A_{pa}^{r.cur}$, at its current location coincides with the point, $A_{pa}^{r.max}$, the teeth flanks, $\mathcal{G}_{max}$ and $\mathcal{P}_{max}$, are generated.

The tooth flanks of a gear in “ S_{pr} -gearing” are the envelope surfaces to all the intermediate gear tooth flanks, $\mathcal{G}_{min}$, $\mathcal{G}_{cur}$, ..., and $\mathcal{G}_{max}$. Similarly, the tooth flanks of a pinion in “ S_{pr} -gearing” are the envelope surfaces to all the intermediate pinion tooth flanks, $\mathcal{P}_{min}$, $\mathcal{P}_{cur}$, ..., and $\mathcal{P}_{max}$. Under any and all permissible configurations of the axes of rotation, O_g and O_p , the interacting tooth flanks of a gear and a mating pinion in “ S_{pr} -gearing” make point contact.

An in detail analysis of Eq. (13) for the position vector of a point, $\mathbf{r}_g^r$, of a tooth flank, $\mathcal{G}_r$, as well as a similar Eq. (14) for the position vector of a point, $\mathbf{r}_p^r$, of a mating pinion tooth flank, $\mathcal{P}_r$, in “ S_{pr} -gearing” reveal that alterations (in comparison to perfect gearing) should be made to the tooth flanks of both members engaged in mesh. It is not allowed to keep one of the members with the original perfect geometry of the tooth flanks, and to compensate for the required changes the mating member of the gear pair. Making changes to the geometry of the tooth flanks of both members of the gear pair is necessary because in “ S_{pr} -gearing” the geometry of the tooth flanks, $\mathcal{G}_r$ and $\mathcal{P}_r$, is predetermined by a given configuration of the rotation vectors, $\boldsymbol{\omega}_g$ and $\boldsymbol{\omega}_p$, of the gear and the pinion.

The tooth flanks of a gear, and a mating pinion in “ S_{pr} -gearing” are always in point contact. However, a degree of conformity^a of the interacting tooth flanks is always of the highest possible value.

The derived equations [see Eq. (13) and Eq. (14)] for the teeth flanks, $\mathcal{G}$ and $\mathcal{P}$, can be used for both, for design of a gear and

a mating pinion tooth flanks, as well as reference surfaces in inspection of a gear and a mating pinion tooth flanks. Implementation of the proposed geometry of tooth flanks in “ S_{pr} -gearing” as a datum surface for the purposes of gear inspection and for measuring deviations of an approximate gear due to “design” errors, as well as “manufacturing” errors, is an additional advantage of this novel kind of gearing.

The geometry of tooth flanks in “ S_{pr} -gearing” is the target when designing real gears. Later on, the determined the most favorable tooth flank geometry can be approximated by surfaces those convenient for machining/finishing of the gears.

The tooth flank in “ S_{pr} -gearing” is also used as the “reference surface” for any and all desired tooth flank modifications. Once the desired tooth flank of a gear and a mating pinion in “ S_{pr} -gearing” is specified in terms of the linear and angular displacement, the design parameters of the gear set, and so forth, then it can be approximated by a corresponding surface that is convenient for machining. This returns a practical kind of the gear tooth flank modification: not a tooth profile modification, and not a tooth lead modification, but a modification of the entire tooth flank of a gear and a mating pinion.

It is instructive to stress here on the important difference between “conjugate” tooth surfaces, $\mathcal{G}$ and $\mathcal{P}$, in “ S_{pr} -gearing” and that in gearings of other types: parallel-axes gearing, intersected-axes gearing, as well as crossed-axes gearing. In the later cases, a gear and a mating pinion teeth flanks, $\mathcal{G}$ and $\mathcal{P}$, are conjugate to one another, and are in line contact with each other. In “ S_{pr} -gearing”, the condition of conjugacy is also fulfilled, however, the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, make “point contact” (and not “line contact”) with one another. It is proposed to refer to the conjugacy of this sort as to “conjugacy of higher order” of the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$.

2.3.4 Angular base pitch

In “ S_{pr} -gearing”, at every instant of time angular base pitch of a gear, $\phi_{b,g}^{inst}$, and angular base pitch of a mating pinion, $\phi_{b,p}^{inst}$, both, are equal to an instant value of the angular operating base pitch, $\phi_{b,op}^{inst}$, of the gear pair (that is, the equalities, $\phi_{b,g}^{inst} = \phi_{b,op}^{inst}$ and $\phi_{b,p}^{inst} = \phi_{b,op}^{inst}$, are observed). These two equalities have to be fulfilled within the effective face width, F_{pa} , of a gear pair. Only under such conditions a pure rotation can be transmitted from a driving shaft to a driven shaft by means of “ S_{pr} -gearing”.

The actual values of the instant angular base pitches vary depending on the actual configuration of the rotation vectors, $\boldsymbol{\omega}_g$ and $\boldsymbol{\omega}_p$. A current configuration of the rotation vectors, $\boldsymbol{\omega}_g$ and $\boldsymbol{\omega}_p$, varies either in time as a function of the gear pair loading, or it varies for different gear pairs of the same design. A current deviations of the rotation vectors, $\boldsymbol{\omega}_g$ and $\boldsymbol{\omega}_p$, from their desired configuration are not known. However, the corresponding tolerances, $\{\Delta\phi_{b,g}^{inst}\}$, $\{\Delta\phi_{b,p}^{inst}\}$, and $\{\Delta\phi_{b,op}^{inst}\}$, for the accuracy of the configuration of the rotation vectors, $\boldsymbol{\omega}_g$ and $\boldsymbol{\omega}_p$, can be set up. Further, the tolerances, $\{\Delta\phi_{b,g}^{inst}\}$, $\{\Delta\phi_{b,p}^{inst}\}$, and $\{\Delta\phi_{b,op}^{inst}\}$, the actual values of which are known, are handled,

^a The higher the degree of conformity at point of contact of the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, the higher contact strength of the interacting teeth, and vice versa.

instead of the deviations, $\Delta\varphi_{b,g}^{inst}$, $\Delta\varphi_{b,p}^{inst}$, and $\Delta\varphi_{b,op}^{inst}$, the actual values of which are unknown.

Shown in Fig. 9, is an operating angular base pitch, $\varphi_{b,op}^r$, constructed for a nominal configuration of the rotation vectors, ω_g and ω_p . The plane-of-action pitch circle of a radius, $r_{w,pa}$, is intersected by the sides of the angle, $\varphi_{b,op}^r$, at points, a and b , that is, the points, a and b , are at an angular distance, $\varphi_{b,op}^r$, from one another. One of the three design parameters, $\varphi_{b,op}$, $r_{w,pa}$, and ab , can be expressed in terms of the rest of two. Moreover, the linear distance, ab , also can be calculated:

$$ab = 2r_{w,pa} \sin \frac{\varphi_{b,op}^r}{2} \quad (16)$$

For a gear pair with the displaced teeth flanks, $\mathcal{G}$ and $\mathcal{P}$, the actual values of the design parameters, $\varphi_{b,op}$ and r_{pa}^{inst} , alter, while the linear distance, ab , is remained of that same value ($ab = \text{const}$). For a specified value of the distance, Δr_{pa}^{inst} , current value of the angular operating base pitch, $\varphi_{b,op}^{inst}$, is calculated as:

$$\varphi_{b,op}^{inst} = 2\sin^{-1} \frac{ab}{2(r_{w,pa} + \Delta r_{pa}^{inst})} \quad (17)$$

Here, in Eq. (17), the distance, Δr_{pa}^{inst} , is a signed value.

In " S_{pr} -gearing", a gear and a mating pinion feature an "interval" of base pitches, and not a fixed certain value of the base pitch. The "range" of the base pitches makes the gear pair capable of accommodating for a prescribed range of permissible values of the axes misalignment.

The values, $\varphi_{b,op}^{min}$ and $\varphi_{b,op}^{max}$, are the minimum and the

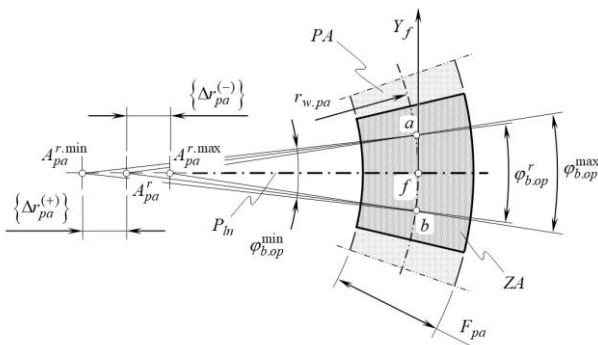


Fig. 9. Operating base pitches, $\varphi_{b,op}^{min}$, $\varphi_{b,op}$, and $\varphi_{b,op}^{max}$.

maximum values of the angular operating base pitch, $\varphi_{b,op}^{inst}$, correspondingly, calculated from Eq. (17). The maximum "negative" deviation, and the maximum "positive" deviation of the angular operating base pitch, $\varphi_{b,op}^{inst}$, from its nominal value, $\varphi_{b,op}$, are designated as $\Delta\varphi_{b,op}^{min}$ and $\Delta\varphi_{b,op}^{max}$, correspondingly. The extremal deviations, $\Delta\varphi_{b,op}^{min}$ and $\Delta\varphi_{b,op}^{max}$, form an interval for permissible variation:

$$\varphi_{b,op}^r - \Delta\varphi_{b,op}^{min} \leq \varphi_{b,op}^{inst} \leq \varphi_{b,op}^r + \Delta\varphi_{b,op}^{max} \quad (18)$$

of the operating base pitch, $\varphi_{b,op}^{inst}$, in a gear pair.

The intervals of permissible variation of the angular base pitches of a gear, $\varphi_{b,g}^{inst}$, and a mating pinion, $\varphi_{b,p}^{inst}$, are identical to the permissible variation of the operating angular base pitch, $\varphi_{b,op}^{inst}$, of the gear pair [see Eq. (18)]. No transmission error is observed if the base pitches of a gear and its mating pinion are equal to one another, and both of them are equal to a current value of the operating base pitch of the gear pair. Ultimately, S_{pr} -gears are capable of transmitting a rotation from a driving shaft to a driven shaft smoothly with no vibration generation and no noise excitation.

In a particular case, when the axes of rotation of a gear and a mating pinion are exactly parallel to one another, the base cones reduce to corresponding base cylinders. Because of this, the angular operating base pitch, $\varphi_{b,op}^{inst}$, approaches zero (that is, $\varphi_{b,op}^{inst} \rightarrow 0^\circ$), and the corresponding radius, r_{pa}^{inst} , approaches an infinity, that is, $r_{pa}^{inst} = (r_{w,pa} + \Delta r_{pa}^{inst}) \rightarrow \infty$ [as well as the axial distance, A , approaches an infinity ($A \rightarrow \infty$)]. Therefore, when determining an actual value of the angular operating base pitch, $\varphi_{b,op}^{inst}$, an indefiniteness of the sort $0 \cdot \infty$ is observed (as an equality, $p_b = \varphi_b \cdot R_{pa}$, is valid). The "L'Hôpital's rule" can be used to resolve this issue. Also, in this particular case (that is, in a case when an actual configuration of the rotation vectors, ω_g and ω_p , corresponds to that in geometrically-accurate parallel-axes gear pair), the instant angular operating base pitch of a gear pair, $\varphi_{b,op}^{inst}$ (a design parameter that is measured in angular units), can be replaced with the instant operating base pitch of a gear pair, $p_{b,op}^{inst}$ (a design parameter that is measured in linear units): $p_{b,op}^{inst} = r_{pa}^{inst} \cdot \varphi_{b,op}^{inst}$. Other techniques for opening the indefiniteness of the sort, $0 \cdot \infty$, can also be used.

The actual value of the operating angular base pitch, $\varphi_{b,op}^{inst}$, depends on the deviation, $\Delta\Sigma$, of the crossed-axes angle, Σ , in a real gear pair, and it does not depend on the actual values of the deviations, ΔC , and ΔA , in the center distance, C , and the length of the axial vector, $\mathbf{A}$. The geometry of the traces of contact point, K , on the gear, $\mathcal{G}$, and the mating pinion, $\mathcal{P}$, tooth flanks, as well as the parameters of the actual portion of the active tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, depend on the actual values of the deviations, ΔC , and ΔA .

2.3.5 Account for normal distribution of gears' axes displacements

In real gearing, tooth flank of a gear, and tooth flank of a mating pinion, both, can be viewed as a kind of a smooth regular surface. A surface is a two-dimensional space, and, thus, it can be analytically expressed in terms of two parameters. Position vector of point, $\mathbf{r}_g^r$, of a gear tooth flank, $\mathcal{G}_r$, as well as position vector of point, $\mathbf{r}_p^r$, of a pinion tooth flank, $\mathcal{P}_r$, are expressed in terms of three parameters, that is, of r_{pa} , φ_p , and X_f [see Eq. (13), and Eq. (14)]. An excessive independent parameter in the equations for position vectors of a point, $\mathbf{r}_g^r$ and $\mathbf{r}_p^r$, can be used with a goal

to develop favorable designs of a gear, $\mathcal{G}_r$, and a mating pinion, $\mathcal{P}_r$, tooth flanks in " S_{pr} -gearing".

There is a certain freedom in varying the design parameters of tooth flanks of the gear, $\mathcal{G}_r$ and the pinion, $\mathcal{P}_r$, within the intervals $-0.5F_{pa} \leq f_{pa} \leq 0$ and $0 \leq f_{pa} \leq +0.5F_{pa}$. It is reasonable to assume that a zero axes misalignment occurs at a midpoint of the effective face width, F_{pa} . Transition from a zero axes misalignment at the midpoint to a maximum axes misalignment at one end of the gear, and to a maximum axes misalignment at the opposite end of the gear, follows a certain function. This makes it possible to synchronize permissible linear and angular displacements in the S_{pr} gear pair. Use of *Gauss'* (of *normal*) distribution of the manufacturing errors sounds attractive from this perspective. The instant values of the design parameters, r_{pa} and X_f , in " S_{pr} -gearing" may correlate to one another so

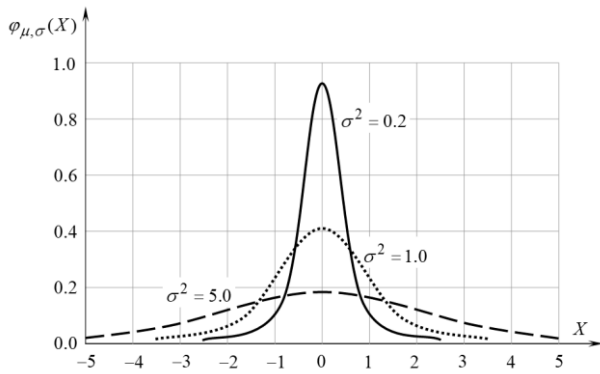


Fig. 10. Probability density function ($\mu = 0$).

as to follow *Gauss'* distribution. When a gear and a mating pinion in " S_{pr} -gearing" are designed this way, for smaller instant values of the axes misalignment (probability of which is higher), contact point is mainly located in the vicinity of the midpoint of the effective face width, F_{pa} . For larger instant values of the axes misalignment (probability of which is lower), contact point is mainly located at the ends of the effective face width, F_{pa} .

As manufacturing errors perfectly follow normal distribution, the implementation of *Gauss'* distribution formula for the derivation of an equation for position vectors of point, $\mathbf{r}_g^r$ and $\mathbf{r}_p^r$, for tooth flanks, $\mathcal{G}_r$ and $\mathcal{P}_r$, becomes reasonable (see Fig. 10):

$$\varphi_{\mu,\sigma}(X) = \frac{1}{\sqrt{2\pi\sigma^2}} e^{-\frac{(X-\mu)^2}{2\sigma^2}} \quad (19)$$

In Eq. (19), the parameter μ is the mean (location of the peak), and σ^2 variance (the measure of the width) of the distribution.

It is reasonable to assume that the actual values of the design parameters, r_{pa} and X_f , correlate to one another following *Gauss'* distribution formula [see Eq. (19)]:

$$\frac{\partial X_f}{\partial r_{pa}}(r_{pa}) = \frac{1}{\sqrt{2\pi\sigma^2}} e^{-\frac{(r_{pa}-r_{w.pa})^2}{2\sigma^2}} \quad (20)$$

The generation of the tooth flank of a gear, $\mathcal{G}_r$, and its mating pinion, $\mathcal{P}_r$, in " S_{pr} -gearing", generated in compliance to Eq. (20), could be of practical importance as the manufacturing errors, as well as the errors and displacements of other nature, follow *Gauss'* distribution formula with high accuracy.

Once one of the design parameters, r_{pa} and X_f , in " S_{pr} -gearing" is expressed in terms another one, that is, a function, $X_f = X_f(r_{pa})$, is established, then position vectors of a point, $\mathbf{r}_g^r$ and $\mathbf{r}_p^r$, of the tooth flank of a gear, $\mathcal{G}_r$, and its mating pinion, $\mathcal{P}_r$, are expressed in terms of two (and not of three) design parameters, as required for any and all smooth regular surfaces.

2.3.6 Features of interaction of gear tooth flanks

A few features of interaction of teeth flanks, $\mathcal{G}$ and $\mathcal{P}$, of a gear and a mating pinion, correspondingly, are briefly discussed immediately below. The readers' attention is focused on the path of contact point over the teeth flanks, $\mathcal{G}$ and $\mathcal{P}$, and on features of geometry of contact between the teeth flanks, $\mathcal{G}$ and $\mathcal{P}$.

2.3.6.1. Trace of contact point on gear tooth flank. As an example, the location and orientation of the trace of contact point, Tr , within tooth flank of a spur S_{pr} -gearing is schematically shown in Fig. 11.

In a case, when a positive axes misalignment is observed, the trace of contact point for the left side, Tr_l^+ , and the right side, Tr_r^+ , of the gear tooth are shifted oppositely toward the ends of the gear, as schematically illustrated in Fig. 11.

In a case of zero axes misalignment, the trace of contact point on the left flank, Tr_l^0 , of the gear tooth, as well as on the right flank, Tr_r^0 , of the gear tooth go through the middle of the face of the gear. Similarly, negative axes misalignment results in the trace of contact point for the left, Tr_l^- , and for the right, Tr_r^- , sides of the gear tooth being shifted oppositely towards the opposite ends of the gear.

The shown in Fig. 11 schematic pertains to " S_{pr} -gearing" that features constant in time values of all the displacements. When the displacements alter in time, then the paths of contact of a more complex geometry are observed.

2.3.6.2. Features of the contact geometry between the tooth flanks. When geometrically-accurate parallel-axes gears rotate the plane of action is unwrapping from one base cylinder and wrapping onto the other base cylinder of mating gears. This schematic is valid as long as the axes of rotations of the gear, O_g , and its pinion, O_p , are parallel to one another. In reality, the axes of rotation, O_g^r and O_p^r , of a gear and a mating pinion are not parallel to one another. At every instant of time, the axes of rotation, O_g^r and O_p^r , cross one another at a certain crossed-axes angle instead. The value of the crossed-axes angle depends on the current parameters of the axes misalignment. Therefore, for desirable parallel-axes real gearing a schematic based on crossing rotation vectors, ω_g^r and ω_p^r , of the gear and the pinion should be applied instead of that for parallel-axes gearing.

Because of the axes misalignment, parallel-axes gearing is actually becomes a kind of spatial gearing that features distance of closest approach of the axes of a gear and a mating pinion, as well

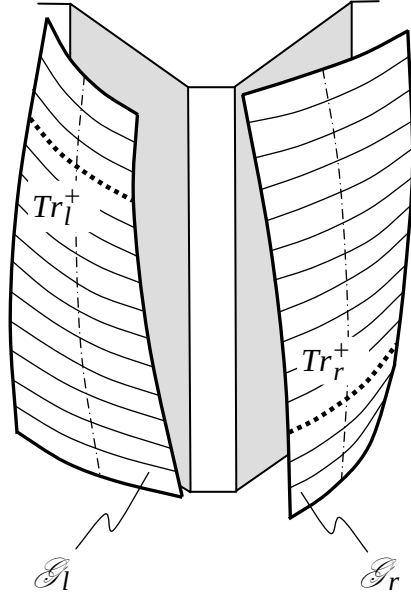


Fig. 11. Traces of contact point, Tr , on tooth flanks in S_{pr} -gearing.

as a crossed-axes angle. In reality, the closest distance of approach of the axes is approximately equal to the distance between parallel axes of the gear and the pinion, and the actual value of the crossed-axes angle is close to either 180° (in external gearing) or 0° (in internal gearing).

In any case, line contact in a geometrically-accurate gear pair is substituted with point contact in its corresponding " S_{pr} -gearing".

The degree of conformity of tooth flanks of the gear, G_r , and the pinion, P_r , at every point of their contact is of the maximum possible degree. The tooth flanks of gears of no other systems of gearing feature degree of conformity as high as gears in " S_{pr} -gearing". The latter is of critical importance from the standpoint of reduction of contact stress, as well as of an increase in the wear resistance of the interacting tooth flanks of the gear and the pinion.

Tooth flanks in " S_{pr} -gearing" always make point contact. The actual location and configuration of the trace of contact point, PC , depends on the actual value of the axes misalignment. As the tooth flanks of a gear, G_r , and a mating pinion, P_r , are always in point contact with one another, they do not envelope each other. However, under any reasonable axes misalignment, instant base pitch of a gear, $\phi_{b,g}^{inst}$, is equal to instant base pitch of a mating pinion, $\phi_{b,p}^{inst}$, and both of them are equal to instant operating base pitch, $\phi_{b,op}^{inst}$, of the gear pair:

$$\phi_{b,g}^{inst} = \phi_{b,op}^{inst} \quad (21)$$

$$\phi_{b,p}^{inst} = \phi_{b,op}^{inst} \quad (22)$$

The proposed geometry of tooth flanks in " S_{pr} -gearing" is derived to accommodate the axes misalignment of all three components of linear displacements, and all three components of angular displacements. This makes " S_{pr} -gearing" insensitive to the axes misalignment within their reasonable range).

The geometry of the tooth flanks of intersected-axes " S_{pr} -gearing" is derived so as to accommodate the axes misalignment of all three components of the both, of the linear displacements as well as of the angular displacements. This makes " S_{pr} -gearing" insensitive to axes misalignment. Gearing of the proposed design is the only kind of self-adjustable gearing possible at all. This significant advantage is achieved due to equality of the angular base pitches of a gear and a mating pinion to the operating base pitch of the gear pair at every instant of time. No transmission error is observed when at every instant of time the base pitches of a gear and a mating pinion are equal to the operating base pitch of the gear pair. Ultimately, by means of S_{pr} -gears a uniform rotation can be smoothly transmitted from a driving shaft to a driven shaft with no vibration generation or noise excitation.

The concept of the "path-of-contact surface, P_{cs} ", can be easily enhanced to " S_{pr} -gearing": the "path-of-contact surface, P_{cs} ", is a continuous set of the circular-arc paths of contact, P_c , at different transverse pressure angle, $\phi_{t,\omega}$. The radius of a current circular-arc path of contact is a "non-linear" function of the distance of point on the pitch line, P_{ln} , from the plane-of-action-apex, A_{pa} ($\neq A_g \neq A_p$). The "path-of-contact surface, P_{cs} ", is a continuous surface of variable transverse pressure angle, $\phi_{t,\omega}$, and variable pitch (along the axis of instant rotation, P_{ln}).

2.4 Potential applications in S_{pr} -gearing, and future work

Two potential areas of application of " S_{pr} -gearing" are discussed immediately below as examples. They are (1) two-degree-of-freedom gearing, and (2) gears having tooth flanks of favorable modification.

Two-degree-of-freedom gearing. Gears that are commonly used in the industry, feature one degree of freedom. The input rotation is the only independent mobility in gear pairs of conventional design. For particular applications, an additional mobility in a gear pair is desired. Gear pairs of this particular kind are referred to as the "two-degrees-of-freedom gearing", or just " $2DOF$ -gearing", for simplicity. The rotation of the input shaft is one of two mobilities, and the motion by means of which the angle between a gear and a mating pinion axes of rotation alters is the other mobility. Other kinds of the "two-degrees-of-freedom gearins" that feature a combination of a translational and a rotational motions are also possible.

"Two-degrees-of-freedom gearing" are investigated since the beginning of 1960s, or even since the earlier time. They can be used in the design of "constant velocity joints", or just "CV-joints", for simplicity. Robot arms is another area of potential application of the " S_{pr} -gearing".

Schematic of an intersected-axes two-degrees-of-freedom gear pair is illustrated in Fig. 12.

The rotation vectors, ω_g and ω_p , of a gear and a mating pinion form an intersected-axes angle, Σ , that varies in time, that is, $\Sigma = \Sigma(t)$. The gear is rotated about its axis of rotation, O_g , with an angular velocity, ω_g . The mating pinion is rotated about its axis of rotation, O_p , with an angular velocity, ω_p . The axes of rotations, O_g and O_p , intersect one another at the plane-of-action-apex, A_{pa}^{inst} .

In order to perform the complementary instant rotation, ω_{rol} , the gear and the pinion roll over one another. For this purpose, the gear and the pinion rotate with angular velocities, $\omega_{rol,g}$ and

$\omega_{rol,p}$, correspondingly, about the centers, o_g and o_p . The

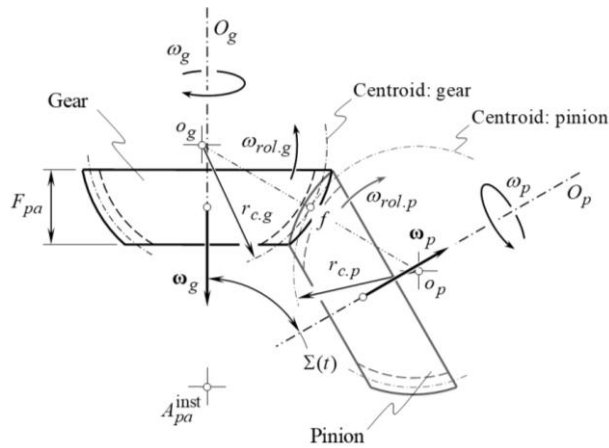


Fig. 12. Schematic of geometrically-accurate intersected-axes two-degrees-of-freedom gear pair.

midpoint, f , of the face width, F_{pa} , in the gear pair is located within a straight line through the o_g and o_p .

When performing these rotations, the circular arc of a radius, $r_{c,g}$, associated with the gear, rolls over the circular arc of a radius, $r_{c,p}$, associated with the pinion. The circular arcs centered at the centers, o_g and o_p . The circular arcs of the radii, $r_{c,g}$ and $r_{c,p}$, are the centrodes in rolling motion of the gear and the pinion I their rolling motion. As the rolling motions, $\omega_{rol,g}$ and $\omega_{rol,p}$, progress, the actual value of the intersected-axes angle, $\Sigma(t)$, varies.

Gears with tooth flanks featuring favorable modification. The determination of the design parameters of the required gear tooth flank modification for a particular application of a gear pair has to be properly engineered, which is a must. This gear problem is a sophisticated one, and it has no reliable solution yet. Nowadays, "trial-and-error" approach is dominated when solving problems of this sort. There is no evidence that the problem has a chance to get solved as long as conventional approaches are used for these purposes.

There is no chance to get the problem solved using for this purpose "trial-and-error" approach, as this approach is endless, and is affordable only to those who has an access to an infinite resource of funds.

A reliable solution to the problem of favorable modification becomes evident due to a novel design of gearing, the so-called " S_{pr} -gearing". The proposed solution to the problem is as follows: for a specified applicaiton of gears, tooth flank geometry of a corresponding " S_{pr} -gearing" has to be determined. After that, gear tooth flanks of the derived geometry are approximated by surface patches that are easy for manufacture. No better modification of gear tooth flanks can be even imagine.

Future work. An in detail investigation of the kinematics and geometry of tooth flanks in " S_{pr} -gearing" is a goal of the future research in the field. Development of method for manufacture of gears for the gearing of the proposed design is of prime importance for the promotion of " S_{pr} -gearing" to application in the industry. The aerospace industry, as well as the automotive industry, are the first two candidates for that.

2.5 Conclusion

The paper deals with gearing for transmission a rotary motion from a driving shaft to a driven shaft that either cross (C_a -gearing), or

intersect (I_a -gearing), or are parallel (P_a -gearing) to one another. If no the gear axes displacements are take into account, the problem of transmission of a steady rotation can be successfully solved of geometrically-accurate gears: by (a) " Eu -gearing" when the gear axes of rotation are parallel to one another, (b) " Gr -gearing" when the gear axes of rotation intersect one each other, or (c) " R -gearing" when the gear axes of rotation cross one another. A proposed novel design of gearing is referred to as " S_{pr} -gearing".

" S_{pr} -gearing" is a kind of geometrically-accurate gearing, as it meets all three fundamental laws of gearing that geometrically-accurate gearing of all designs fulfil. Capability to accomodate for the axes misalignment (both, linear, as well as angular displacements) is the main feature of " S_{pr} -gearing". This is due to point contact between tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, of a gear and its mating pionion. In addition, " S_{pr} -gearing" features a highest possible degree of conformity at every point of contact of the tooth flanks, $\mathcal{G}$ and $\mathcal{P}$, and, thus, it features the highest possible power density being transmitting by the gear pair.

As examples, a few potential applications of " S_{pr} -gearing" are discussed. The first of them pertains to two-degree-of-freedom gearing, and the other one pertains to determination of a most favorable tooth flank modification for a particular gear application.

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Determining the quality of renovation layers by using nano HVOF coatings

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Abstract: The paper presents the results of the research aimed at determining the quality of coatings applied by the HVOF spray coating technology and the advances in its application. **WC-Co-Cr 86/10/4 nano covers were evaluated at two grain sizes.** The effect of nano particle size on coatings quality was evaluated under abrasive wear conditions with bound abrasive in the initial state and after thermal cyclic loading. Coating thickness measurements, microhardness and structural analysis of the coatings were performed. As the number of thermal cycles increased, the hardness of the coatings increased. A thermal load between 600 and 800 °C results in the crystallization of the amorphous phase, which leads to the precipitation of the WC phase from the Co-Cr matrix, and thus to an increase in the microhardness of the coating. The REM analysis of the structure of nano coatings confirmed that the exposure of the samples to the thermal load had an influence on their structural composition. As the number of thermal cycles increased, the wear of the coatings under dry friction conditions increased.

Keywords: HVOF, NANO COATING, RENOVATION

1. Introduction

Thermal spraying is an important surface engineering technique. Thermal spraying of coatings is used in all industrial areas, in primary production, as well as in the renovation of parts. An adequately selected and optimally applied coating significantly increases the service life and reliability of such treated materials.

Trends in surface treatment technology are generally focused on:

- full automation of the coating process and automatic quality control,
- tailored coating systems with unique chemistry and structure,
- nanoscience and nanotechnologies, which will play a crucial role in the next generation of coating technology [1],
- hybridization in surface treatments based on:
 - (a) the formation of multifunctional coatings,
 - (b) multi-process hybrid technologies (e.g., HVOF (High Velocity Oxygen Fuel) and laser remelting [2]),
 - (c) or multi-source hybrid technologies (e.g., PS-PVD (Plasma Spray - Physical Vapor Deposition) of ceramics for protective coatings. PS-PVD method allows thin (less than 10 µm) single layers to be deposited and multilayer coatings of less than 100 µm to be generated with the flexibility to tailor microstructures by changing processing conditions. Coatings of yttria-stabilized zirconia (YSZ) were applied to NiCrAlY bond coated superalloy substrates using the PS-PVD coater at NASA Glenn Research Center [3].

Thanks to the use and development of various techniques and materials for thermal spraying, the nano-thermal spraying process has become one of the important directions in the development of thermal spraying techniques.

Development in the framework of the HVOF thermal spraying technique can be specified especially by:

Development of nano-powders, nano-coatings

As reported in [4] thermal spraying nano-coatings mainly include three types: nano-crystal coating composed of only one material, nano-crystal composite coating composed of two or more nano-materials, and composite coating reinforced by nano-particulate. Adding nano-particulate in traditional coatings can greatly improve coating properties and functions with low cost. Poor mechanical strength as well as mismatches in coefficient of thermal expansion often limits the use of dense ceramic coatings on metals. However, increasing porosity would decrease the protection capability of the coating. It is already recognized that nano crystalline materials have special mechanical properties. Typically the strength of crystalline materials is increased with decreasing grain size and materials with small grain size often exhibit also superplastic behavior at elevated temperature. Nano-crystallinity has a positive influence on toughness of ceramic materials especially if alloyed with nano

phased metals. Furthermore, hardness and wear properties of coatings are usually improved.

Developments in Suspension and Solution Precursor Thermal Spray Processes

As reported in [5] thermal spray coatings from liquid feedstock such as suspensions and solution precursors have received increasing interest due to the unique coating properties obtainable by these processes. Several research groups are working on the basis of plasma as well as on high-velocity oxy-fuel approaches to manufacture advanced nano-structured and nano-phased materials.

Ecologisation

Innovation from the viewpoint of ecologisation in thermal spraying coatings represents green carbides. Recently, just cermet coatings containing hard WC particles in metallic matrix applied using HVOF technology was seen as a less dangerous and more environmentally friendly alternative to hard chrome plating [6]. Because the WC-based powders contain heavy metals such as Co and Ni, there is very strict logistics of powders used for coatings formation in the HVOF process. Currently, effort of materials scientists is focused on developing new powders, in which these elements in metallic matrix is eliminated and are replaced by other alloys. One of them is the powder WC-FeCrAl, called "green carbides"[7].

Development of a new generation of HVOF-Systems

Despite undeniable achievements, there is still potential to improve HVOF technology concerning its versatility and to reduce coating costs. As published in [8] developed new HVOF spraying guns operating at increased combustion chamber pressures show high potential for spraying of coatings consisting of metals that do not feature the outstanding ductility of pure copper or aluminium. High deposition efficiency, i.e. up to 85%, at considerable powder feed rate of 4.5 kg/h is already possible for spraying of corrosion protective iron or nickel based coatings like AISI 446, AISI 316L or MCrAlYs. Also spraying of highly reactive materials like titanium under atmospheric conditions becomes feasible.

High performance thermal-sprayed coatings in narrow and complex areas

A complex geometry is always a challenge for the thermal spray community, because the spray angle affects the droplet's impact on the substrate resulting in changes in coating properties. In the work [9] an attempt has been made to overcome this limitation. WC-Co-Cr coating was deposited on the CA6NM steel using newly developed high-velocity oxy liquid fuel (HVOLF) thermal spraying gun. The angular design of this gun allows spraying in narrow areas, complex parts and inside internal diameter minimum as 140 mm having almost uniform coating properties at both spray angles (45° and 70°); this makes it a promising solution for coating deposition on complex geometries [9].

Post-processing of HVOF coating

Thermally sprayed coating possesses high surface roughness which is beyond the desirable limit for many industrial applications. The coatings are needed to be finished to get desired surface finish

and form accuracy. The post processing techniques are also required to eliminate the defect of the as-sprayed coating as well as to improve the coating properties [10].

The laser remelting technique is considered a promising and effective method for improving the surface of thermally sprayed coatings, eliminating microstructural defects such as pores and cracks, increasing the life of parts and equipment by increasing microhardness and increasing anchoring force between coating and substrate [11].

In the study [12], a multi-step finishing strategy was adopted to improve the surface finish of high velocity oxy-fuel spraying (HVOF) sprayed WC-Co coating. Shape adaptive grinding (SAG) was performed using a zirconia-alumina abrasive polishing pad which resulted in differential finishing of WC and Co binder due to their significant hardness difference. To overcome this situation, chemical assisted SAG is performed using Murakami's reagent. A thin passivation layer was formed on the top of coating due to chemical interaction which has hardness many times lesser than the bare coating and facilitates high and uniform material removal rate.

Thermal spray property-performance maps "TS maps"

The process-structure-property relations can be presented by process maps, which can be used as design tool for coating processing. Process maps are interrelationships among the process variables and output responses [13]. The process map methodology is under development for process control and coating properties optimization. In the process mapping concept, the diagnostic tools are used for understanding the fundamentals of relationships in the thermal spray process, starting from powder to thermal spraying process, to deposit formation, to coating characteristics, and finally to coating performance. The TS maps are based on coating characteristics of major importance, i.e., microhardness, adhesion strength, and the elastic modulus of thermal spray coatings [14]. The process mapping optimization tool has been widely applied for plasma spray process [13,15], but it can be successfully used for HVOF process as well [16,17,18].

2. Materials and methodology of experiments

Materials

The base material AISI 316L (1.4404; 17 349) was chosen for the application of the coatings. It is a chromium-nickel austenitic unstabilized molybdenum low carbon stainless steel. The annealing temperature of this steel is in the range of 1000-1100 °C. This steel is non-magnetic, hardenable and has a tendency to harden during cold forming. As the temperature of this steel increases, the values of Rm and Rp decrease. It is resistant to corrosion in industrial environments, is not prone to intergranular corrosion in heat-affected areas, but is less resistant in nitric acid environments. It is used for welded structures that are located in aggressive industrial environments, in coastal areas and in chemically treated swimming pool environments. It is suitable for the construction of chemical equipment, including pressure vessels. Also suitable is a non-oxidizing medium containing strong organic and inorganic acids at lower concentrations up to medium temperatures. It can also be used in contact with food, but not with drinking water due to its Ni limit value [75]. The chemical composition of the basic material is given in Tab.1.

Table 1: Chemical composition of the base material

element	C	Si	Mn	P	S	Cr	Mo	Ni	Fe
[wt %]	<0.03	1	2	0.4	0.3	16.5-18.5	2-2.5	10.5-13.5	Bal.

WC-Co-Cr 86/10/4 nano coatings were evaluated at two grain sizes.

Nano HVOF technology differs from the classic HVOF coating by the highest quality in every respect. The nano technology uses a very fine powder, which ensures perfect adhesion to the surface of the material. Using cheaper powders that have a larger particle size results in porosity (typically less than 0.5%). The higher porosity of the coatings reduces the spray efficiency. WC-CoCr -9 + / 1µm

(mark E) and -17 / + 3 µm (mark F) coatings were applied with a plasma system designed to generate HVOF and nanoHVOF with an ABB 4600 IRCS robot and a CPF2 powder feeder from Thermico GmbH & Co.KG. The injection parameters of these coatings are in Tab.2.

Table 2: Spraying parameters of nano coatings

WC-CoCr	Powder A	Powder B
Parameter	-9/+1 µm	-17/+3 µm
Nozzle length [mm]	140	140
Nozzle diameter [mm]	11 - 11.8	11 - 11.8
Kerosene [l/hour]	13.5	17
Oxygen [l/hour]	395	570
Nitrogen [l/hour]	580	250
Hydrogen [l/hour]	140	140
Powder [g/min]	50	50
Distance [mm]	220	250

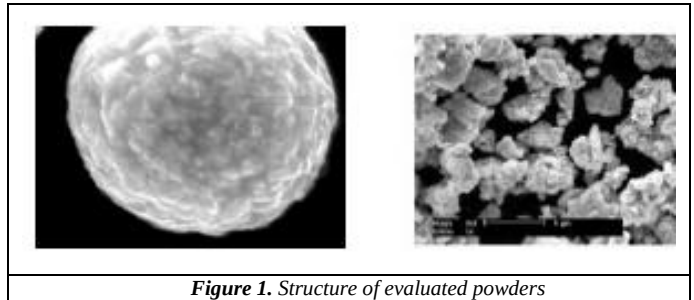


Figure 1. Structure of evaluated powders

Table 3: Chemical composition of nano coatings [wt %]

Powder	Cr	WC	Co
A	4	86	10
B	4	86	10

WC-CoCr (-9 / + 1 µm) / WC-CoCr (-17 / + 3 µm) 86/10/4 these coatings are applied to hydraulic cylinders with high resistance to corrosion and wear, also to compressor shafts, valves and rollers in the paper industry. The hardness of these coatings is greater than 1200 HV 0.3. Determination of properties of selected types of composite coatings:

- Determination of adhesion of coatings
- Determination of microhardness of coatings
- Structural and phase analysis
- Determination of porosity of coatings
- Evaluation of the quality of coatings in tribological conditions depending on thermal cycles
- Determination of fracture toughness of coatings

During the application of the coatings, the samples were subjected to thermal cyclic stress in the electric chamber furnace of SM Lin Elektro Therm GmbH. The cycle itself consisted of heating the samples to 600 °C for 20 minutes and then cooling to ambient temperature. The test specimens were subjected to 5 and 10 thermal cycles.

The adhesion of the coatings was determined according to the standard STN EN 582. The principle of the test is to tear the coating from the base material. Adhesion was evaluated on a ZD10 / 91 tearing machine. Test specimens measuring φ25x50 mm were adhesively bonded to the counterpart using CHS Epoxy 1200 adhesive.

The microgeometry of the coatings was evaluated using a Sufttest SJ-301 instrument.

Prior to coating, the samples were pretreated by blasting technology on a pneumatic blasting machine. The blasting agent was F22 corundum with a grain size of 0.56. It is artificial white corundum A99, which is produced by melting chemically from high-purity aluminum oxide in an electric arc furnace "for casting". Castings weighing 6 tons are then crushed and ground to abrasive and refractory grains after cooling. During processing, they are magnetized several times, sorted on grids and in some cases can

also be chemically treated in hydrochloric acid. White fused alumina is the purest form of alumina. This abrasive is very hard and sharp [76]. The chemical composition of the blasting agent is listed in Table 7.

Table 4: Chemical composition of blasting agents

Element	Al ₂ O ₃	Na ₂ O	Fe ₂ O ₃	SiO ₂
Quantity [wt %]	99.58	0.33	0.03	0.06

The thickness of the coatings was evaluated on cross-sections of metallographic sections with an Olympus Bx FM electron microscope using Quick Photo camera 2.3 software, in accordance with STN EN ISO 9220. The microhardness of the coatings was measured according to STN ISO 4516. The Vickers method was used to determine microhardness. A Shimadzu HMV-2 Micro Hardness tester was used. The loading force was 980.7 mN (100 g) and the holding time was 15 s.

Prior to coating, the test specimens were pretreated on a pneumatic blasting machine. The blasting pressure used at the outlet was 0.4 MPa at a nozzle distance of 300 mm from the base material and with an angle of incidence of the blasting material of 75 °.

Coating adhesion

The average destructive force values on the tungsten carbide nanocoatings ranged from 41.5 to 56.3. Due to thermal cycles, the values of the average destructive force at the failure of the joint did not change significantly. The joint was broken in the area of the adhesive. Due to the fact that the evaluated coatings show a high adhesion greater than 70 MPa, the test was limited by the cohesion of the adhesive.

Microgeometry of coatings

In Tab. 2 shows the values of the mean arithmetic deviation of the coating profile as a function of the number of thermal cycles.

Table 5: Measured values of Ra and Rz

Number of cycles	0		5		10	
	Ra [μm]	Rz [μm]	Ra [μm]	Rz [μm]	Ra [μm]	Rz [μm]
Coating E	1.673	10.75	1.37	8.74	1.4766	8.79
Coating F	2.43	14.306	2.496	14.11	2.653	13.3367

Due to thermal cycles, the Ra values did not change significantly for the studied coatings. The lowest value of the profile unevenness height was recorded for the coating E / 5 8.74. These values did not change significantly due to thermal cycles in the studied coatings. Based on the results of microgeometry of coatings, it can be stated that the height of the coating is affected by the particle size of the coating. The thickness of the coatings ranged from 261 μm to 415 μm. Thermal cyclic loading had no effect on the coating thickness.

Table 6 Measured values of coating thickness

Thickness of coating [μm]/Number of cycles			
Coating	0	5	10
E	261	235	270
F	411	409	415

Microhardness of substrate material and coatings

The microhardness values of the base material, which were measured for comparison of the results with the evaluated coatings, are given in tab. 15. The measured values of microhardness of coatings are given in Tab. 16. The microhardness values of the base material ranged from 278 to 304 HV 0.1. Due to thermal cycles, the microhardness value of the base material did not change significantly.

Table 7: Hardness of substrate material

Number of cycles	0	5	10
AISI 316L HV0.1	304	278	282

Table 8: Microhardness of nano coatings

Coating/ number of cycles	Microhardness HV0.1		
	0	5	10
E	1564	1767	1841
F	1297	1640	1703

The microhardness of the coatings ranged from 1297 to 1841HV0.1. The highest value of microhardness was measured on the coating E10 1841 HV 0.1. Due to temperature cycles, the microhardness values of nanocoatings increased.

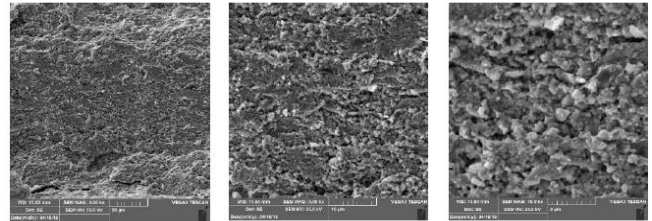


Figure 2. The structure of WC-CoCr (-9/+1 μm) coating in the initial state

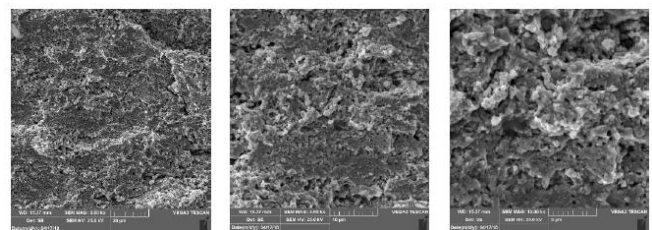


Figure 3. The structure of WC-CoCr (-17/+3 μm) coating in the initial state

The WC, W₂C, WP₂, Co phases were identified on samples E / 10 and F / 0, and a W₂CoB₂-related phase can be observed in the diffraction pattern. Since the composition of the layer is not formed under equilibrium conditions, the W₂CoB₂ phase may occur in a metastable state resp. in metastable state resp. non-stoichiometric. Therefore, its reflections may be shifted in the measured reciprocal space 2 theta. CoW_{0.4} and WO₃ oxide phases were observed on the samples after thermal cycling. The surface was passivated. The oxide phases increase with the number of thermal cycles, which manifests itself as an increase in the intensity of the reflections of the corresponding oxide and a relative decrease in the intensity of the reflections of the WC phase relative to the reflections of the oxides. It can also be observed that with increasing number of thermal fatigue cycles, the oxidic phase of CoWO₄ grows faster. The porosity values for these coatings do not exceed 1%, which was also confirmed for these coatings.

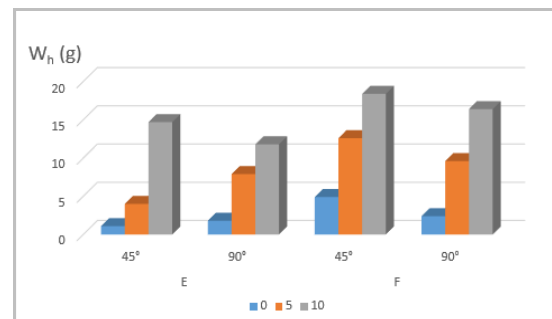


Figure 4. the amount of wear of the coatings depends on the angle of impact of the abrasive and the number of thermal cycles

Evaluation of indentation fracture toughness of coatings

Based on the methodology given in chap. The indentation fracture toughness of the coatings was evaluated. 5 loads, namely

10, 20, 30 40 and 50 N, were used to create the impression in the coating. Certain conditions need to be met to determine some models. Of the given models, it was possible to use three, for other models the conditions were not met. Due to the large number of calculations, a load of 40N was chosen to evaluate the indentation fracture toughness of the coatings and the LF model was used. The calculated values of fracture toughness of the evaluated coatings are given in Tab. 32. FIG. 100 shows these values. Fig. 100 shows indenter indents into coatings.

Table 9: Indentation fracture toughness

Indentation fracture toughness [MPa.m ^{-1/2}]			
Number of cycles/Coating	0	5	10
E	22.2x10 ⁻⁶	14.7x10 ⁻⁶	22.2x10 ⁻⁶
F	23.9x10 ⁻⁶	21.6x10 ⁻⁶	18.8x10 ⁻⁶

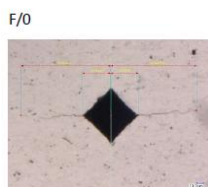


Figure 5. Indentation into coating F

Tungsten carbide-based coatings showed poorer resistance to crack propagation at baseline. This resistance has improved with the number of thermal cycles.

3. Summary

- The evaluated coatings have a very good adhesion to the base material and thus a suitable pre-treatment of the surface of the base material before the application of the coatings was chosen. During the experimental work on the loading of coatings in various environments and tribological tests, there was no peeling or other types of degradation of coatings caused by low adhesion. However, it should be noted that the test result was limited by the cohesion of the adhesive used.
- The lowest values of microgeometry were measured on Ea F coatings, due to the composition of the coatings formed of finer particles compared to other coatings. On these coatings, the Ra values ranged from 1.37 to 2.653 µm and the Rz values from 8.74 to 14.306 µm. Depending on the number of cycles, the values of Ra and Rz did not change significantly. The microgeometry of the coatings depends on the particle size of the additive powder.
- The thickness of the coatings depends on the spray parameters. The nanocoatings reached a coating thickness of 235 to 415 µm.
- Nanocoatings showed microhardness in the range from 1297 to 1841 HV 0.1 with respect to the nanoparticles used and the formation of a fine homogeneous structure. In the case of both nanocoatings, their microhardness increased significantly due to thermal cyclic loading, up to 17 to 31%.
- Due to thermal cycles by phase analysis, new oxide-based phases were identified. Phase analysis showed that new phases were formed after 5 thermal cycles, but no changes occurred after ten cycles.
- The porosity of the evaluated coatings ranged from 0.22 to 0.77%. Due to thermal cyclic loading, the porosity decreased in all types of coatings. Evaluation of the porosity of the coatings confirmed that the coatings applied by HVOF technology did not exceed 1% of the porosity.
- Tungsten carbide-based coatings showed poorer resistance to crack propagation at baseline. By the number of cycles, the crack propagation resistance of this crack coating improved in the nanocoatings.

- These properties can be used in the field of renovation and protection of surfaces of more stressed machine parts. The important contribution of this research work is the knowledge that changing the granularity of powders for thermal spraying of metal-ceramic coatings is a prospective way, how to suitably affect the properties of coatings without changing their chemical composition.

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Adhesive bonding of dissimilar materials

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Abstract: The paper is focused on research results of dissimilar materials joining by adhesive bonding. Galvanized and non-galvanized steel sheets with a thickness of 0.8 mm were joined together. To compare the effect of the presence or absence of the Zn layer on the strength of the adhesive joints, control joints were made, consisting only of galvanized and only of non-galvanized materials. The materials were joined on one hand without any surface preparation, and on the other hand prepared by degreasing and mechanical roughening with sandpaper. Epoxy/PVC-Polymerblend adhesive with glass beads was used for bonding to check the thickness of the adhesive. The tensile lap-shear strength was tested according to DIN EN 1465. The joints made of galvanized materials only proved the highest, the joints made of non-galvanized materials only proved the lowest shear strength. The mixed joints did not reach the joint strength of galvanized materials, but exceeded the joint strength of non-galvanized materials. The adhesion of the adhesive to non-galvanized substrates seems to be the limiting factor of the strength of mixed joints.

Keywords: SURFACE PREPARATION, DEGREASING, ADHESIVE BONDING, DISSIMILAR MATERIALS, SHEAR STRENGTH

1. Introduction

Joining unequal materials is a current topic, especially in the automotive industry. It resulted from the requirement of tailored design of individual parts in order to ensure maximum use of the properties of the individual materials [1]. The most common issue is the joining of different grades of steel, joining steels with non-ferrous metal alloys [2,3], joining metals with GFRP materials [4,5], etc.

Joining unequal materials also brings its problems, which need to be solved and which result from different material characteristics of joined materials [1,6,7]. Different formability affects riveting processes (clinching, riveting), different chemical composition affects the weldability of materials [3], etc. For these reasons, multiple bonding technologies are often combined, leading to heterogeneous bonding, with the term heterogeneous referring to both the combination of bonding technologies used and the bonding of unequal materials.

Great attention is paid to combination of adhesive bonding with other technologies, e.g. laser welding or spot resistance welding, clinching, or thermal drilling [5,8-10]. By combination of these technologies, optimal synergy of strong points of individual technologies can be achieved. Adhesive bonding technology contributes to these combined joints mainly by the load distribution over a larger area, ensures joint sealing, etc. However, if the adhesive bonding technology is to be implemented industrially, sufficient adhesion of the adhesive to the both substrates used must be ensured, as well as the simplest possible surfaces preparation for bonding, that can be applied to both substrates used simultaneously.

It is well known that adhesives cannot adhere sufficiently to surfaces that are contaminated with oil, grease or dust [11,12]. The most commonly used materials in automotive production - thin steel sheets are usually contaminated with preservative oils, which are to protect the materials during storage and transport, or lubricants, which are used to reduce friction during forming. Deep drawing of thin sheets is immediately followed by adhesive bonding, welding, surface preparation of the body for painting and finally curing while passing the EC oven. It would be very expensive to include some wet cleaning process before adhesion bonding. To bond these industrially contaminated materials, progressive adhesives have been developed with the ability to "absorb" oil and concentrate it into discrete islands [13]. Although these islands reduce the contact area with the adhesive, the joints show satisfactory strength.

Other adhesives usually require a clean and adequately roughened surface [14]. Current alkaline degreasing agents are able to remove oil and grease, but also partially remove (etch) the surface oxide layers present e.g. on hot-dip galvanized materials, Fig. 1, and thus contribute to the mechanical anchoring of the adhesive.

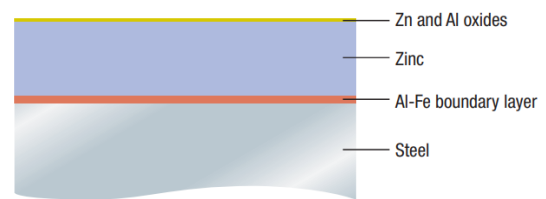


Fig. 1 Surface layers on hot-dip galvanized steel

The aim of the experiments was to determine the strength of adhesive joints of different materials with no surface preparation and with surfaces prepared by degreasing and roughening. The results were compared with the strength of joints made of the same materials.

2. Materials and methods

Substrates

Two materials were used for the production of joints:

- non-galvanized steel sheet made of deep-drawn steel DC04 (1.0338) 0.8 mm thick (next marked DC),
- hot-dip galvanized steel sheet from DP steel HCT600X+Z (1.0941) 0.8 mm thick (next marked ZN).

The mechanical properties of the materials and the condition of the surface on delivery are given in Table 1.

Table 1 Mechanical properties and surface condition of materials

	Re [MPa]	Rm [MPa]	A80 [%]	Zn layer [g/m ²]	Surface conditions	Oil weight [g/m ²]
DC	197	327	39	-	matt	0.5-2.5
ZN	346	654	23.5	105	minimized Zn spangle, improved surface	0.6-2.5

Table 2 shows the chemical composition of the materials used.

Table 2 Chemical composition of materials, wt. %

	C	Mn	Si	P	S	Al	Ti	Mo	Cr
DC	0.04	0.25	-	0.009	0.008	-	-	-	-
ZN	0.09	1.88	0.25	0.012	0.003	0.026	0.002	0.002	0.210

Test specimens with dimensions of 100 × 25 mm were made from metal sheet by cutting.

Adhesive

A one-component adhesive Epoxy/PVC-Polymerblend with glass beads was used to create the adhesive joints. It is primarily intended for the automotive industry. It has good adhesion to bar metal, zinc coated substrates, zinc-magnesium and aluminum alloys. It contains glass beads (diameter 150-200 μm) to create an even adhesive gap. The curing of the adhesive takes place at a temperature of 175°C, 25 min. The tensile strength of the adhesive is 13.5 MPa. Shear strength according to DIN EN 1465 at 23°C is at least 12 MPa. The working range of the adhesive is from -40° to 90°C.

Surface preparation

Two surface preparation procedures were used to determine the effect of surface preparation on joint strength. The first procedure consisted of degreasing the substrates with an alkaline degreasing agent at 60°C, 10 min. After rinsing and drying, the contact surfaces were manually roughened by sandpaper with grit size of #400 and cleaned from abrasive particles. The second, reference procedure was to bond the substrates without any surface preparation, i.e. as supplied by the manufacturer (electrostatically oiled).

Measurement of surface roughness

The surface roughness of both materials was determined with a Mitutoyo SJ-301 contact profilometer using the parameters Ra - arithmetical mean deviation of profile, Rz - maximum height of profile, Rt - total height of profile, RSm - mean width of the profile elements, and RPc - standardized numbers of peaks. Five measurements were performed on each type of material in a degreased state and also after roughening with sandpaper (perpendicularly to roughening direction).

Adhesive joint formation

The joints were made according to DIN EN 1465, with an overlap area of 25 × 12.5 mm, Fig. 2. Curing of the adhesive took place in an oven at 180°C, 30 min.

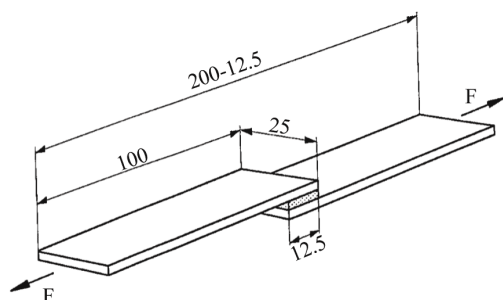


Fig. 2 Geometry of adhesive joint

The number of joints and the combination of materials to be joined was chosen so that the average strength of the joints could be determined from at least five values, Table 3.

Table 3 Material combinations and number of joints

Surface preparation	DC-DC	ZN-ZN	DC-ZN
as supplied (no surface preparation)	5	5	5
degreased/ground	5	5	5

Testing of joints

Determination of tensile lap-shear strength of bonded assemblies was performed on a TiraTest 2300 test device with continuous load-displacement recording. The maximum force at the joint failure was recorded and then the type of failure was visually determined (adhesive, cohesive, mixed, Fig. 3). There are also load - displacement records, from which we can identify some properties of the adhesive (toughness) and the overall course of the test.

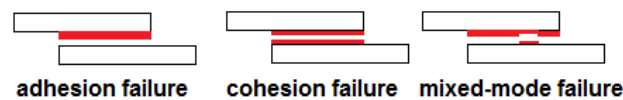


Fig. 3 Modes of adhesive joint failure

3. Results and discussion

Surface roughness

The surface roughness of the materials is given in Table 4 as the average of five measurements.

Table 4 Surface roughness of materials

roughness parameters	Ra [μm]	Rz [μm]	Rt [μm]	RSm [μm]	RPc [number/cm]
DC	1.12	6.19	7.38	297.40	36.06
ZN	1.28	6.23	7.83	145.80	69.44
DC roughened	0.93	5.22	6.17	168.40	60.92
ZN roughened	0.98	6.38	7.91	103.40	96.84

From the measured roughness values it is clear that in terms of the parameter Ra, the surfaces DC and ZN are almost the same. However, the parameter Ra is the mean deviation of the surface profile and does not indicate the nature of the irregularities present on the surface. The parameters Rz and Rt are also very similar for the DC and Zn surfaces, which means that both surfaces have the same range of inequalities from the highest peak to the lowest valley. However, even from these values we cannot say anything about the distribution of individual elements of the profile in the horizontal direction. The parameters RSm or RPc are used to characterize the distribution of inequalities in the horizontal direction, where RPc is the inverse value of RSm. These parameters show that the mean width of the profile elements in DC steel is twice as large as in ZN. The surface of the ZN material thus has twice as many peaks per unit length as the DC material. Based on this, it is possible to assume a larger contact surface between the adhesive and the surface of the ZN material than between the adhesive and the DC material.

Roughening the surface with sandpaper reduced the vertical parameters Ra and Rz (roughness is smaller), which is negative in relation to bonding, but the differences in the horizontal distribution of irregularities between the two materials decreased. The above considerations are also confirmed by surface profilographs, Fig. 4.

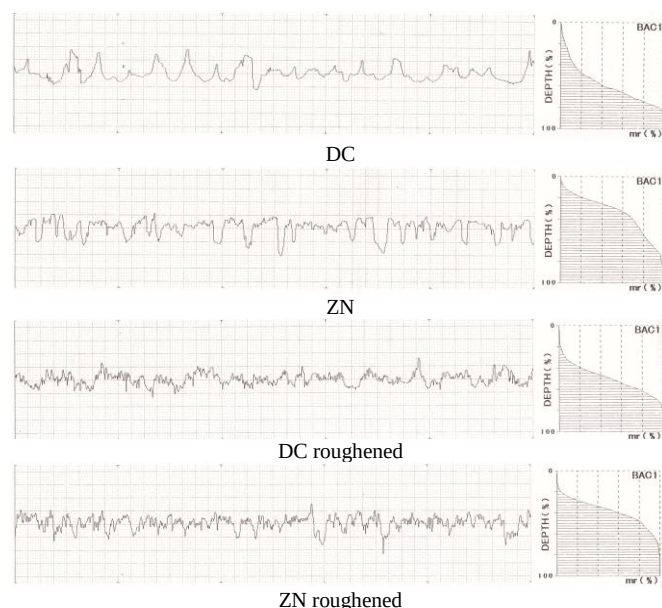


Fig. 4 Profilographs of surfaces for adhesive joining

The Abbot-Firestone curves of DC and ZN materials also indicate differences between the surfaces. Profilographs give reasons to assume a larger straighten surface area in ZN material, which means a larger contact area with the adhesive and thus higher joint strength.

Shear strength of joints

Table 5 shows the maximum forces and achieved bond strengths for particular material combinations and surface preparation, and are shown graphically in Fig. 5.

Table 5 Maximum forces and achieved strength of joints

	Fmax [N]		Shear strength [MPa]	
	as supplied no preparation	degreased, ground	as supplied no preparation	degreased, ground
DC-DC	5180±132	5128±132	16.6±0.4	16.4±0.4
ZN-ZN	6822±145	5479±402	21.8±0.5	17.5±1.3
DC-ZN	5530±97	5058±265	17.7±0.3	16.2±0.8

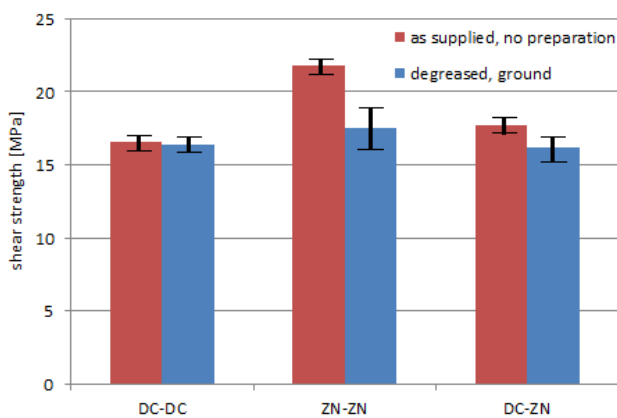


Fig. 5 Shear strength of joints

Fig. 5 shows that, as far as material combinations are concerned, joints of the same DC-DC materials have the lowest shear strength of all combinations, ZN-ZN joints have the highest shear strength, regardless of the surface preparation. The shear strength of mixed joints is logically located in the middle, between the shear strength of DC-DC and ZN-ZN joints.

Regarding the preparation of surfaces before adhesive bonding, from the obtained results it seems that degreasing and grinding does not significantly change the shear strength of DC-DC joints, for ZN-ZN joints degreasing significantly reduces shear joint strength and thus worsens also the strength of mixed DC-ZN joints.

Anyway, determined shear strengths of the joints meet the data of the adhesive manufacturer, which states for the same length of bonding line and at temperature 23°C shear strength > 12 MPa.

The appearance of joints failure is shown in Fig. 6.

Although all failure surfaces show 100% cohesive failure, there are still some differences between fracture surfaces on DC and ZN materials. On ZN substrates, after failure, a relatively thick layer of adhesive remains, while the DC substrate is partially visible below the adhesive layer - only a thin layer of adhesive remains on it. This suggests a better bonding of the adhesive to the ZN substrate, in both types of surface preparation.

Taking into account the course of the shear stress and the peel stress along bondline, Fig. 7, the joints with the ZN substrate showed excellent adhesion even at the edges where the stresses reach a local maximum.

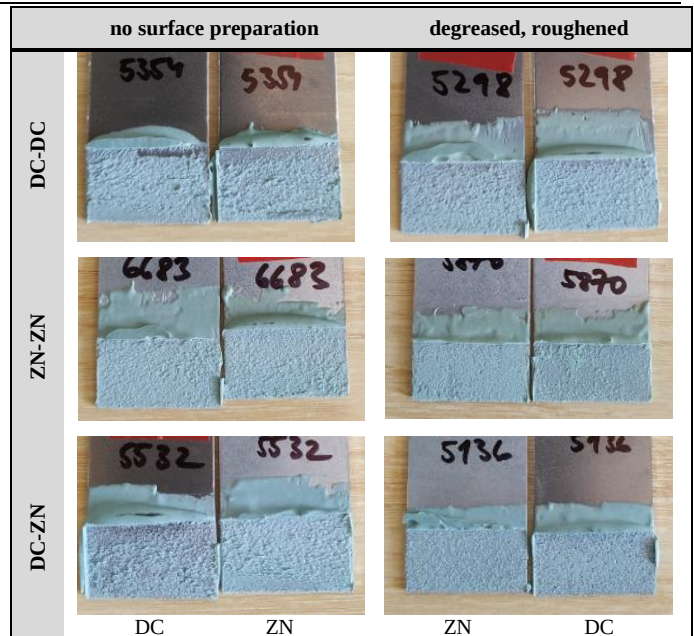


Fig. 6 Appearance of joint failure

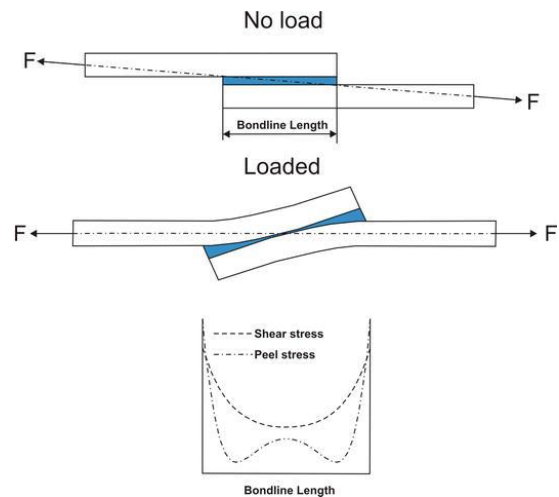
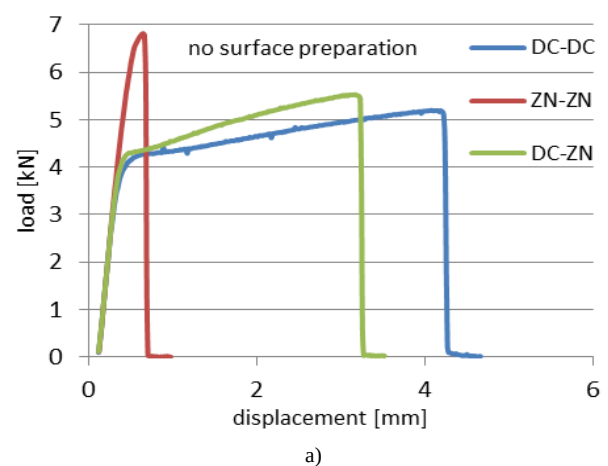


Fig. 7 Course of shear and peel stress along bondline

Fig. 8 shows the time course of the loading of the joints until failure.



a)

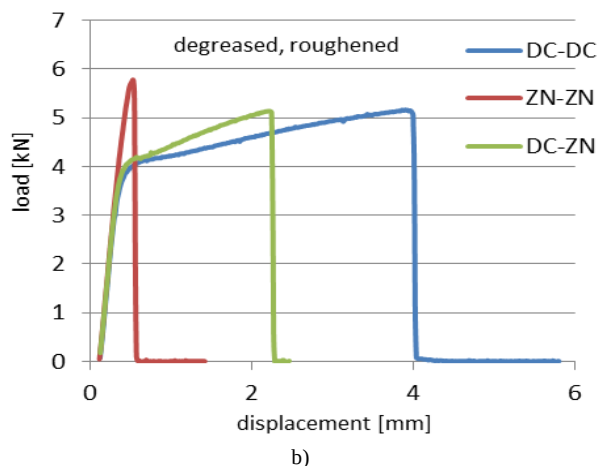


Fig. 8 Load – displacement curves for different material combinations, a) no surface preparation, b) degreasing, roughening

From Fig. 8 can be seen that the first, starting part of all the curves is the same, linear, corresponding to the elastic deformation of the adhesive. At some point, the curves begin to differ.

For ZN-ZN joints (no surface preparation), the load-displacement curve is linear up to the maximum, then suddenly decreases - joint fails. The shape of the curve indicates that the adhesion of the adhesive to the ZN surface is greater than the cohesion of the adhesive. The cohesion of the adhesive in this case determines the strength of the joint.

On the load-displacement curve of DC-DC joints, the linear part is followed by a break, the curve continues to rise, but with less intensity. After reaching maximum strength, it suddenly decreases - joint fails. The initial linear part represents, as already mentioned, the elastic deformation of the adhesive. The breaking point is the state where the cohesive stress in the adhesive reaches the level of adhesion of the adhesive to the substrate and the work required to deform the adhesive begins to be used to gradually break the bonds to the substrate. This continues until F_{max} . The curve ends with a joint failure. Gradual disruption of adhesive bonds to the DC substrate is also evidenced by failure appearance, where DC substrates are partially visible from below the adhesive layer, while ZN substrates remains covered with a continuous layer of adhesive of greater thickness, see Fig. 6.

The load-displacement curve of DC-ZN mixed joints is similar in shape to the curve of DC-DC joints, i.e. the behavior of mixed joints is determined by a weaker element - the DC substrate. The strength of mixed joints is higher than that of DC-DC joints (increased by high adhesion to the ZN substrate), but lower than that of ZN-ZN joints (reduced by lower adhesion to the DC substrate).

The situation was the same for joints prepared by degreasing and roughening, only the maximum joint strengths were lower.

Considering the mechanical properties of the substrates, the DC04 substrate has lower mechanical properties than the galvanized ZN substrate. The experimentally determined yield strength R_e of the material DC04 (not joint) is 197 MPa, which for material thickness of 0.8 mm and specimen width of 25 mm means that the load corresponding to the yield point is 3940 N. At this load, the linear part of the DC04 stress-strain diagram ends. This corresponds exactly to the breaking point on the curves, Fig. 8. This means that after reaching the breaking point, plastic deformation and strengthening of the DC04 material also takes place until the value of the cohesive strength of the adhesive is reached. Then joint fails, at approximately the same load for both DC-DC and DC-ZN joints. This means that during the loading of the joint, a partial reduction of the sample width outside the joint area takes place in the DC material.

4. Conclusion

The paper presents the results of the evaluation of the shear strength of adhesive joints made of the same and different materials. Bonding without surface preparation (bonding of oiled substrates) was performed using the oil-absorbing properties of the adhesive used. For comparison, surface preparation by degreasing with an alkaline degreasing agent and roughening with sandpaper perpendicular to the direction of application of the load was tested.

The results showed that the bond strength exceeded the value declared by the manufacturer, where the joints of galvanized substrates had the highest strength, the joints of non-galvanized substrates the lowest. The mixed joints have reached a strength which lies between the strength of joints made only of galvanized and only of non-galvanized steel sheets.

The connection of the shear strength of the formed joints with the surface microgeometry of the used substrates was proved.

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Fatigue calculation of "SH" drainpipes

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Abstract: During the combustion processes of fuels, fuel gasses are created which flow over the heat exchanger. The flow of fuel gasses brings ash particles which are deposited on heat exchanger surfaces and thus reduce the heat exchange efficiency. The removal of ash deposits from the heat exchanger is performed by a striker. Striker indirectly strikes the floor of the lower chamber of the heat exchanger via the mandrel and transmits kinetic energy to it. The impact results in inertial forces on the layer of deposits that are greater than those of adhesion forces between the surface of the exchanger pipes and the ash deposits. The pipes are loaded with displacement caused by the impact of a mandrel used to clean the heat exchanger, displacement caused by the operating pressure inside the pipe, and thermal elongation. The load of the pipes is cyclic, and calculation was performed according to the standards: EN 12952-3:2012-03 and EN 13445-3:2009. The calculation was performed at critical pipe cross-sections. The Abaqus/CAE2016 software package was used to determine the critical cross-sections of pipes and stress that occur in them. The model was created using beam finite elements. After analyzing the stress and applying the standards, a conclusion is reached on the fatigue strength of the drainage pipes. The heat exchanger pipes are made of austenitic W.Nr.2.5956 and 16Mo3 steels. The pipe with maximum fatigue stress of 208 MPa is made of 16Mo3 steel, which has a tensile strength from 450 to 600 MPa. Young modulus of elasticity at an operating temperature of 383 °C is $E = 173,9$ GPa. The permanent strength diagram is taken from EN 12952-3: 2011 standard shows that this material can withstand 10^7 cycles with such fatigue stress, and the required number of cycles is determined based on the customer's request which is the projected number of years. The plant is projected on operating 5 years. The designed number of cycles is 350400 cycles, so it is concluded that the pipes satisfy the conditions of exploitation.

Keywords: FATIGUE, IMPACT LOAD, STEAM BOILERS, HEAT EXCHANGER, DRAINAGE PIPES

1. Introduction

Experience shows that fracture of structures or machine parts during regular operating conditions are most often due to fatigue.

Fatigue or fatigue damage refers to the modification of the properties of materials due to the application of stress cycles whose repetition can lead to fracture.

Uniaxial loading is defined as the amplitude of the maximum stress during a cycle s_{max} . The stress ratio R is the ratio between the minimum stress s_{min} and the maximum stress s_{max} , $R = s_{min}/s_{max}$.

We sometimes must distinguish the alternating component s_a from the mean stress s_m . Thus, depending on the relative values of these two components, we can differentiate the tests under different stresses (see Figure 1.6), such as:

- fully reversed: $\sigma_m = 0$, $R = -1$;
- asymmetrically reversed: $0 < \sigma_m < \sigma_a$, $-1 < R < 0$;
- repeated: $R = 0$;
- alternating tension: $\sigma_m > \sigma_a$, $0 < R < 1$. [1]

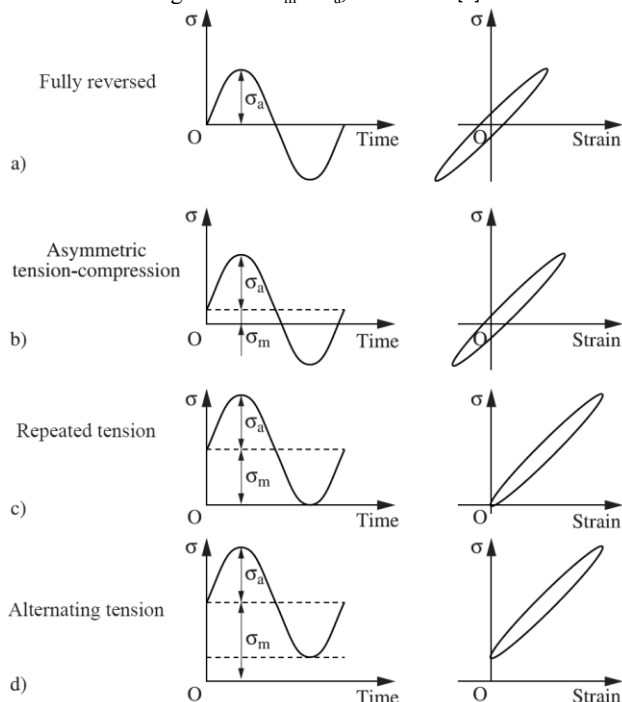


Fig. 1 Different cases of fatigue stresses: load-time; force-strain [1]

Electrical energy has a rising form of use in everyday activity, therefore, producing electrical energy is important in present-day life.

Incineration is used as a treatment for a very wide range of wastes. Incineration itself is commonly only one part of a complex waste treatment system that altogether provides for the overall management of the broad range of wastes that arise in society. The objective of waste incineration is to treat wastes to reduce their volume and hazard, whilst capturing (and thus concentrating) or destroying potentially harmful substances that are, or may be, released during incineration. Incineration processes can also provide a means to enable recovery of the energy. [2]

Waste incineration installations are producers of electricity which transform burning energy from different waste to electrical energy. Burning fire from waste and its smoke that runs through smokestack is heating up the water in pipes that are part of waste incineration installations.

Ashes from the smoke can accumulate on pipes that are filled with water (heat exchangers), therefore, ashes must be cleaned so that heating of water can continue. Cleaning of ashes is done by thorn impact that drains the ashes from the heat exchanger. On those heat exchangers are located drainpipes that enable water drain in time of no use of waste incineration installations.

Thorn impact on heat exchangers and drainpipes is presenting a cyclic load of drainpipes used in this paper.

Drainpipes are moved by thorn impact up to ± 11 mm and those are two types of load, Step A (-11 mm) and Step B (+11 mm). Also, there is pipe displacement due to thermal expansion which occurs because of the different temperatures of water inside of heat exchangers.

Fatigue calculation in this paper is calculated for drainpipes SH3.2, SH3.1, and SH1.2. SH 3.2 drainpipes have the highest calculation temperature which is the most critical case. However, because the different material is used to produce drainpipes, calculations are repeated for SH3.1 and SH 1.2 drainpipes.

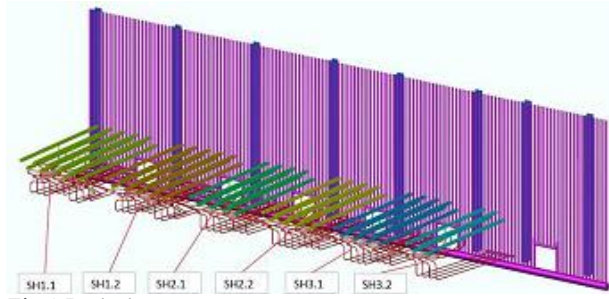


Fig. 2 Drainpipes

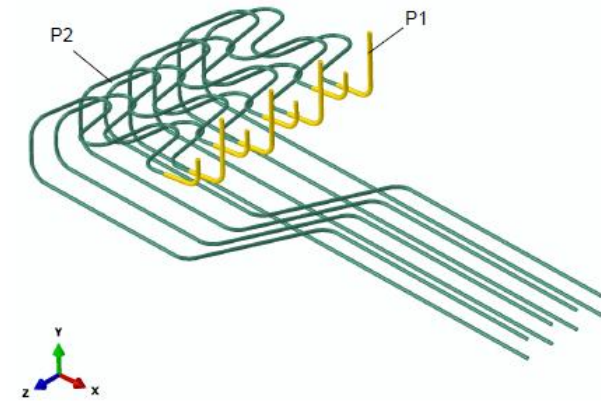


Fig. 3 SH 3.2 drainpipes model

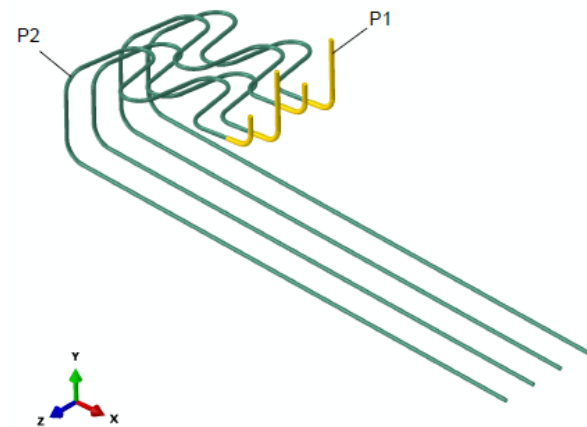


Fig. 4 SH 3.1 drainpipes model

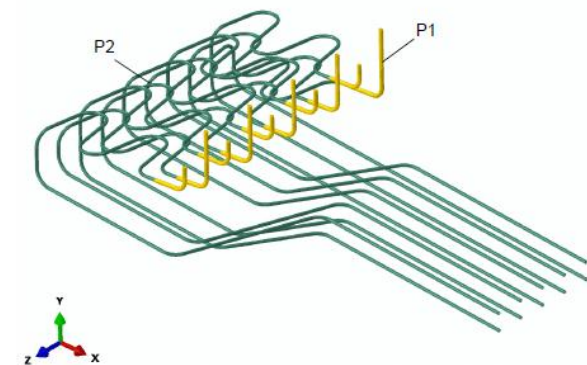


Fig. 5 SH 1.2 drainpipes model

2. Prerequisites and means for solving the problem

Numerical calculation is used for obtaining different sorts of results like axial force, bending moments, and twisting moment.

After placing different displacements on each group of drainpipes due to thorn impact and thermal elongation, the next steps are to put gravity load on pipes and fix lower parts of pipes. Materials properties are included for pipes, and mesh is made by standard beam elements (B31), 5 mm of size. The calculation is performed in Abaqus/CAE 2016 and after obtaining these results, analytical calculation follows.

Analytical calculation is made by combining two similar standards, EN 12952-3:2012-03 and EN 13445-3:2009. Each drainpipe has 4 critical cross-section which occurs firstly on the place of connections of drainpipes and heat exchangers (P1 pipe), secondly on the connection of different pipes (P1 and P2). The next critical cross-section is on the rounded part of the drainpipe with maximal stress, last critical cross-section is on the lower part of pipes, a connection between pipe and shell. The calculation is performed for every pipe in each group of drainpipes and due to similarities in geometry, some pipes are calculated together.

After creating a numerical model of pipes and solving them, the next step is to read results in critical cross-section. In this part, it is important to read different results on each pipe in the same elements. Also, it is important to read the highest absolute values of different results in each element.

3. Preliminary calculations

After reading the values of forces and moments from the Abaqus 2016 software package, these values are included in the formulas for calculating stresses in previously agreed cross-sections. The stress caused by the bending moment is calculated according to the expression $S_b = \frac{M_b}{W_x}$ where M_b represents the bending moment, and W_x is the axial moment of inertia. The shear stress is calculated according to the expression: $\tau = \frac{M_t}{W_p} = \frac{M_t}{2W_x}$ where M_t represents the torque, W_x is the axial moment of inertia. The axial stress is calculated according to the expression: $S_n = \frac{SF}{A_x}$ where SF represents the axial force, and A_x is the area cross-section. To determine the stress, it is necessary to know the geometric characteristics of the pipe cross-sections.

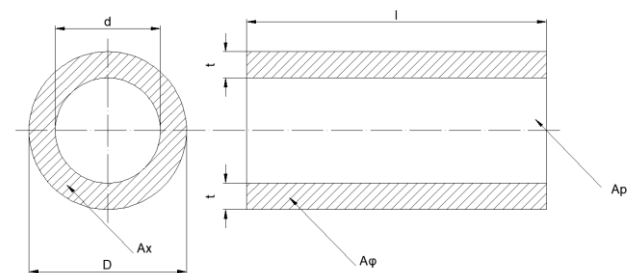


Fig. 6 Dimensions and geometrical characteristics of the pipe cross-section

Internal pressure in each pipe causes circular and longitudinal stress of the pipe. Pipes longitudinal stresses caused by internal pressure is calculated according to the expression $S_x = \frac{pd^2\pi}{4A_x}$, and

circular pressure is calculated according to the expression $S_\phi = \frac{pd}{2t}$.

In those expressions p represents internal pressure, and other dimensions can be seen in fig. 6.

The next step is to determine the total stress in the x-axis direction. The total stress in the direction of x-axis is determined by the expression $S = S_x + S_b + S_n$. The principal stresses can now be

calculated for the individual pipe cross-sections according to the

$$\sigma_{1,2} = \frac{S + S_{\phi}}{2} \pm \sqrt{\left(\frac{S - S_{\phi}}{2}\right)^2 + \tau^2}.$$

After calculating all the necessary stresses for the application of standard, we can proceed to the calculation according to standards *EN 12952-3:2012-03* and *EN 13445-3:2009*. To consider the welded joints, correction factors C_{k1} , C_{k2} , and C_{k3} are used, and for the quality of the treated surface is used C_{k0} . This case uses a weld on only one side so factor C_{k3} will be used. The values of the correction factors are shown in Table 1.

Table 1: Correction factors C_k for taking account of the notch-effect associated with the influence of the weldments [5]

Welded joints			
Tensile strength R_m [N/mm ²]	Configuration group K1 (slight notch effect)	Configuration group K2 (moderate notch effect)	Configuration group K3 (pronounced notch effect)
	C_{k1}	C_{k2}	C_{k3}
400	$1,5 \leq 0,19 \lg N_a + 0,62 \leq 1,8$	$1,6 \leq 0,21 \lg N_a + 0,79 \leq 2,1$	$1,8 \leq 0,34 \lg N_a + 0,66 \leq 2,8$
600	$1,7 \leq 0,40 \lg N_a + 0,20 \leq 2,7$	$1,9 \leq 0,40 \lg N_a + 0,60 \leq 3,1$	$2,1 \leq 0,56 \lg N_a + 0,40 \leq 4,0$
800	$1,8 \leq 0,56 \lg N_a + 0,12 \leq 3,4$	$2,1 \leq 0,56 \lg N_a + 0,44 \leq 4,0$	
1000	$1,9 \leq 0,70 \lg N_a + 0,40 \leq 4,0$	$2,5 \leq 0,75 \lg N_a + 0,25 \leq 5,0$	

Controlling stress range shall be determined depending on elastic range, partly elastic range, or fully plastic range. In case of a load, cycle temperature is higher than 100 °C, the reduction in the fatigue strength caused by the temperature shall be taken into account by means of a correction factor C_{t*} . The last step is to calculate the controlling stress range that takes the high temperature in fatigue calculation. [2,3]

4. Results of fatigue calculations

In chapter 3 is shown how to calculate the fatigue life of drainpipes, for each pipe in a different group of pipes calculation must be derived.

All calculated results, with those of fatigue life of drainpipes, are shown in Table 2.

Table 2: Final results of stress calculation

SH 3.2	$2f_{at}^*$ [MPa]			
Pipe number	Cross-section 1	Cross-section 2	Cross-section 3	Cross-section 4
1-3	108	167	103	11
2-4	157	159	106	8
SH 3.1	$2f_{at}^*$ [MPa]			
Pipe number	Cross-section 1	Cross-section 2	Cross-section 3	Cross-section 4
1-3	110	80	71	14
2-4	123	59	68	9
5-7	116	175	120	15
6-8	179	163	123	10
SH 1.2	$2f_{at}^*$ [MPa]			
Pipe number	Cross-section 1	Cross-section 2	Cross-section 3	Cross-section 4
1	92	68	53	11
2	103	51	53	8
3	95	104	60	13
4	105	87	55	10
5-7	118	101	66	14
6	202	78	67	9
8	170	157	111	10
9	120	175	106	14
10	208	154	119	8

From table 1 can be seen that the maximum stress appears in pipe 10, SH 1.2. Comparing this value of stress to the diagram shown in the Figure 5, the pipe can withstand 10^7 cycles.

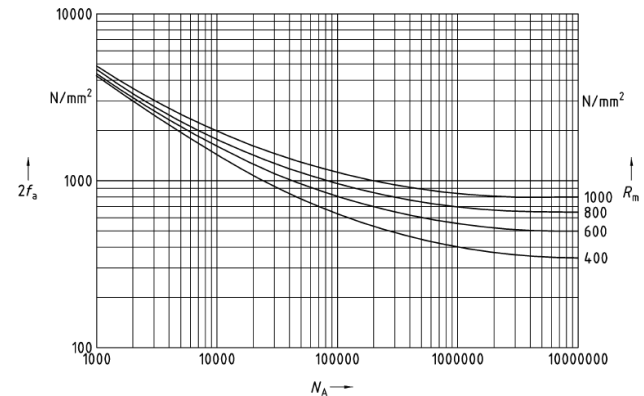


Fig. 5 Number of load cycles N_A for crack initiation (mean value of the scatter band) as a function of the stress range for unnotched bar specimens of high-temperature ferritic rolled or forged steels at room temperature and $f_v = 0$ [5]

5. Conclusion

This paper describes the generation of electricity through waste incineration as a fuel source. The problem of fatigue that occurs in cyclically loaded structures is explained. Fatigue calculation was performed for each of the drainage pipes groups for four critical cross-sections.

The calculation was performed according to the norm for each of four agreed critical cross-sections from each group. It has been found that all pipes can withstand 10^7 cycles. It would be good to examine the credibility of the selected cross-section in such a way that every final element on one tube is subjected to the calculation.

Because of the great effort of performing such calculation, it will be of a big time-saver to use software like Fe-Safe to calculate fatigues life.

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Technique of tooth relieving of worm hobs for cutting Novikov gears with double lines of action

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Abstract. Worm hobs for cutting Novikov gears have the profile of a generating worm, which varies not only in a radius, but also in a sign of curvature. A technique for relieving tooth flanks of these hobs by disk grinding wheel is developed. It allows for bringing together grinding and ground profiles in an axial section of the generating worm as close as possible. A method for calculating setting-ups and profile coordinates of the grinding wheel is given for real production conditions.

KEY WORDS: WORM HOBS, GRINDING, RELIEVING, TOOTH PROFILE, NOVIKOV GEARS.

Introduction

Formation of screw and relieved surfaces by disk grinding wheels includes inevitable deviation of ground surface and wheel profiles. Such a deviation was called an organic error. For a straight basic rack the organic error has to be minimized as much as possible or the obtained profile is taken as the basic rack at the attained degree of minimization [1-3]. For the screw and relieved surfaces when a profile has a substantial and variable curvature, this organic error must be eliminated to the maximum degree.

This problem was solved in a well-known example for the unrulid worm of ZT2 type, proposed by F.L. Litvin in 1961 (later, in the monograph [4] such worms are considered as worms ZF-II type). An axial profile of the grinding wheel was set in the form of a circle arc, and then an angle of installation of the grinding wheel axis and an axial profile of a worm thread were sought based on the condition that this arc was a contact line of the wheel and the ground surface. Obviously, a solution of an inverse problem is required to obtain a worm hob with relieved surfaces of teeth being close to the found worm thread profile.

In [2, 5], authors proposed the solution of a similar problem for gears with liquid friction with respect to the convex thread profile of a worm and a hob for cutting teeth. It was specified for the substantially curvilinear axial profile of the working worm (or the hob generating worm) and also taking into account a profile of a grinding wheel in the axial section of the worm. The setting angle of the grinding wheel axis was determined from the condition that this axis intersects two normal lines to the thread axial profile and it is located in a plane parallel to the axis of the worm. The proposed method ensures minimization of the organic error in the profiling of the worm thread. Similar organic error in tool profiling was accounted, because of the necessary identity of the profile of the hob generating worm.

In [6] authors made the first attempt to apply general principles of this method to the study of the relieving process of worm hobs for cutting Wildhaber-Novikov gears. In this case, two significant factors were taken into account. At first, a profile curvature of a generating rack of these worm hobs is a variable not only in a radius of curvature, but also in its sign. At second, the possibilities of a relieving machine allow for setting the axis of a grinding wheel in a plane not parallel to the axis of the ground product; it improves grinding conditions for a number of parameters.

In the present work, this research is developed and continued. It solved the main issues of relieving technique that are functionally oriented for solving the following problems: providing rear angles of the teeth necessary for the wear resistance of cutting edges of a hob, minimizing the organic error of a profile of a generating rack, determining the main parameters for setting up a relieving machine and profile of a grinding circle.

1 Wildhaber-Novikov gears. Parameters of the basic rack

For the first time, helical gears with an initial point tangency of the circular-helical surfaces of the teeth were proposed by the eminent

American Inventor E. Wildhaber in 1926 in US Patent No 1,601,750. However, at that time this invention was unnoticed, and such gears received rapid development only after M. L. Novikov had formulated a general principle of their formation and showed that such gears will have the increased load capacity, primarily in a contact endurance [7].

Initially, M. L. Novikov proposed a version of helical overcentre gears, in which tooth profiles, convex on a pinion and concave on a wheel, were delineated with circle arcs in the face section of gears. Later, V. N. Kudryavtsev showed that the same effect could be achieved in the gears synthesized based on two incongruent basic racks with the circle arc profiles and cut by two worm hobs: one for the pinion with convex teeth, the other for the wheel with concave ones [8].

However, at present, basic racks are standardized and widely used in production with a full tooth profile, which provide in gears two lines of action and allow for cutting the pinion and the wheel by one worm hob [9]. In particular, according to the Russian Standard GOST 15023-76, the basic rack consists of a convex addendum, a concave dedendum and a short straight section between them. Geometry of Novikov gears with two lines of action is calculated acc. Russian Standard GOST 17744-72.

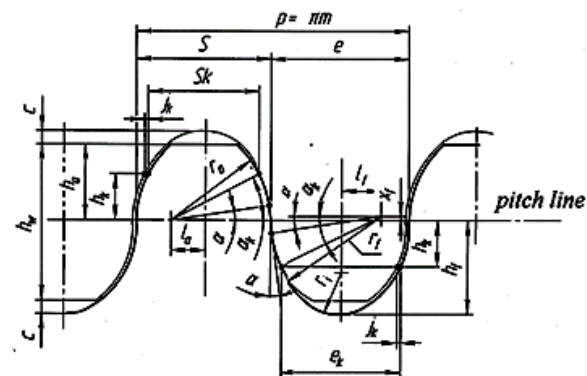


Fig. 1. Basic rack for Novikov gears acc. GOST 15023-76.

Fig.1 shows the main parameters of the basic rack, the values of which depend on the range of modules. The pressure angle α_k at the predetermined contact points on the tooth addendum and dedendum is equal 27° for any case. These points are also characterized by parameters: distance h_k from the pitch line, hob tooth width e_k at its addendum and space width S_k at its dedendum, the difference $j_k = e_k - S_k$ provides a backlash between the teeth of the wheels to be cut. The pressure angle α in the straight section of the tooth depends on the module and is equal to $\approx 8^\circ$.

The hob tooth in the normal section is formed according to the dimensions of the space of the basic rack. Similarly, the space between the hob teeth forms the tooth of the wheel to be cut.

The convex profile of hob tooth is outlined by a radius r_f , the center of curvature of which is located at a distance l_f from the axis of symmetry of the head and is shifted relative to the pitch line to the hob axis by x_f . The center of the concave arc radius r_a of the hob

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line is $\alpha_0 \approx 8^\circ$, and therefore the rear angles in this section are minimal, which during cutting significantly reduces the wear resistance of the hobs, since it is precisely in this section that they are intensively formed wear sites [10]. In addition to arc sections with large pressure angles, a large shift of the relief surface profile relative to the pitch line will occur, which will appear at re-sharpening of the hob. So such hobs cannot be relieved with a purely radial method and radial-axial relieving must be used.

In general case of radial-axial relieving of the teeth of the worm hob [1, 2] the direction of relieving motion makes up the angle φ_c with the perpendicular to the hob axis, which ensures the obtaining of a necessary rear angle λ_r near the cutting edges of the hob teeth.

The grinding wheel axis O_w-O_w is turned (in the projection to the horizontal plane) at an angle φ_0 to the hob axis O_1-O_1 (see below Fig.5) and is simultaneously inclined to this plane by the angle β_w . Such a method of setting the axis of the grinding wheel eliminates large differences in curvature of the grinding surface of the wheel, improves the grinding conditions and increases the life of the grinding wheel.

During the relieving, the hob rotates around its axis O_1-O_1 with the angular velocity ω_1 , the grinding wheel moves along the hob axis at a speed $p\omega_1$ and performs reciprocating motion with a speed $k\omega_1$, where $k = Kz_t/2\pi$ is the relieving parameter; K is the recession of the Archimedean spiral of the cam of the relieving mechanism on the angular pitch of the hob teeth. The relieving carriage slides are turned relative to the perpendicular to the hob axis by an angle φ_c which allows increasing the rear angle λ_r on the lateral surface of the tooth.

The current radius r_w of the grinding wheel is determined at the point of its contact with the tooth of the hob on its current cylinder. For the initial position of the wheel, it is advisable to take the tangency of its maximum radius r_{wm} with the minimum radius of the work piece surface $r_{1\min}$.

With radial-axial relieving, the lead angle γ_0 of the relieved surface at each of two selected points ($i_{1,2}$) is determined from the expression [1, 2]

$$\tan \gamma_{0i} = [p \pm k \sin (\alpha_i + \varphi_c) / \cos \alpha_i] / x_{0i} \quad (6)$$

where the radius x_{0i} of the location of the selected point on the ground surface is determined from the drawing of the product.

The sign "+" in square brackets refers to the teeth side where the lead angle of the relieved surface is greater than the lead angle of the generating surface of the hob, sign "minus", respectively, to the opposite side of the teeth.

From formula (6) it follows that the values of the angles γ_{0i} depend on the angle φ_c of the installation of the relieving support, which is determined by the required rear angle near the cutting edge of the tooth. For hobs with variable profile curvature, this angle is determined at the location of smallest value of angle α_{0i} and equal to $\alpha_{0\min}=7.7523^\circ$ in the considered example for a straight line segment near the pitch line.

4 Calculation of the angle φ_c installation of the relieving support of the machine

For worm hobs intended for finish cutting of involute gears with module 12 mm of the thermally hardened steels, the Russian State Standard 9324-80 recommends radial relieving parameters providing the normal relief angle on a pitch cylinder $\approx 4^\circ$. Based on this recommendation, for the considered type of hobs, we establish the following dependence, which ensures the determination of φ_c with the sufficient accuracy for practical calculations:

$$(\tan \gamma_{0F} - \tan \gamma_{xF}) \cos \alpha_{0\min} = \tan 4^\circ \quad (7)$$

Expanding the meanings of $\tan \gamma_{0F}$ from (6) and $\tan \gamma_{xF} = p/r_F$, we obtain the following expression for determining the value of φ_c

$$\sin (\alpha_{0\min} + \varphi_{cR,L}) = 0,44 r_F / (Kz_t) \quad (8)$$

Indexes R, L relate to the right and left sides of the hob tooth. For the example considered above with $\alpha_{0\min}=7.7523^\circ$ we obtain $\varphi_{cR,L}=12.229^\circ$. Taking into account the price of rotation angle scale we set according to the rounding rules $\varphi_{cR,L}=12^\circ$.

Table 1 shows the calculated parameters of the profile of the hob tooth and the lead angles of the relieved surface of the teeth at seven key points of the profile $i=1...7$:

- 1 - the boundary of the active section of the profile on the tooth leg,
- 2 - the contact point on the dedendum of the hob tooth,
- 3 - the conjugation of the arc concave section with a rectilinear section,
- 4 - the point on the hob pitch cylinder,
- 5 - the conjugation of the rectilinear and arc sections on the tooth head,
- 6 - the contact point on the addendum of the hob tooth,
- 7 - the boundary of the active section of the profile on the tooth head.

Table 1. Parameters of hob tooth profile in key points (for the given example).

№ p/p	Radius of point, mm $x_i = r_i$	Pressure angle in axial section, α_{0i}°	Lead angle of screw surface of cutting edges, γ_i°	Lead angles of relieved surface on the tooth sides:	
				right, γ_{0iR}°	left, γ_{0iL}°
1	67.575	47.338	5.09	21.207	-11.853
2	71.467	27.079	4.814	13.598	-4.200
3	75.906	7.752	4.587	8.576	0.446
4	77.775	7.752	4.425	8.373	0.435
5	79.25	7.752	4.343	8.219	0.427
6	84.083	27.35	4.094	11.679	-3.635
7	88.095	46.951	3.909	16.439	-9.001

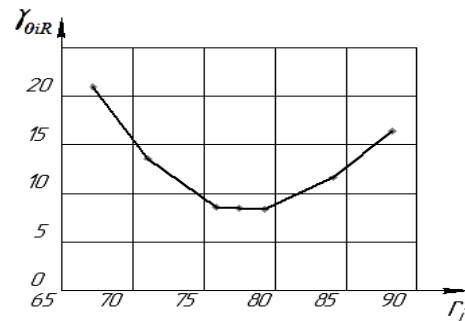


Fig. 3. Diagram of the function $\gamma_{0iR} = f(r_i)$ – right flank.

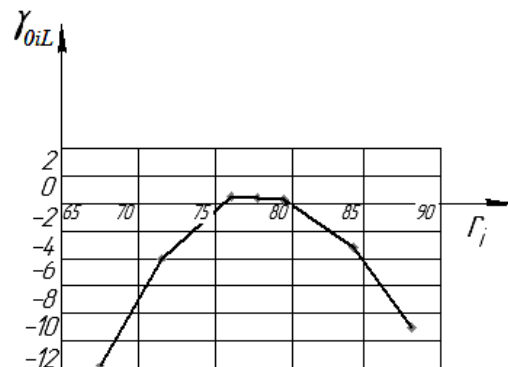


Fig. 4. Diagram of the function $\gamma_{0iL} = f(r_i)$ – left flank.

In Fig. 3 and 4 the graphs of the dependence of the lead angles of the relieved surface on the radius $x_{0i} = r_i$ of the profile point, taking into account the pressure angle α_{0i} at $\varphi_c = 12^\circ$ are shown for both sides of tooth.

The analysis of the calculated data shows that the lateral surfaces of teeth of a hob are organically characterized by the presence of paired points with equal lead angles of the relieved surface, but the values of these angles differ significantly from the lead angle of the helical generating surface.

5 Choice of the designed points to construct designed normals

The choice of two points of the axial profile of the relieved surface with the parameters: $A_1(x_{01}, z_{01}, \alpha_{01})$ and $A_2(x_{02}, z_{02}, \alpha_{02})$, must satisfy two conditions. Firstly, the part between the selected points should cover the active part of the tooth profile as much as possible, including contact points ($\alpha_0 \approx 27^\circ$). Secondly, undercutting of the tooth cutting edges should be excluded.

To do this, on the right side of the teeth of the right-hand hob the angle β_w of setting the axis of the grinding wheel in a plane parallel to the hob axis should not exceed significantly the maximum lead angle of the generating surface. When relieving the opposite side of the tooth, the angle β_w should not be substantially less than the lead angle of the helical surface of the cutting edges on the tooth head.

Point A_1 should be selected on the tooth leg of the hob. A study conducted with specific examples (see the calculation example below) showed that these conditions are most fully satisfied if, as point A_1 , the design contact point in the tooth space on a concave arc at $\alpha_{01} \approx 27^\circ$ (point 2 in the table 1). As point A_2 , we should choose the conjugation of the convex arc section with the transition curve on the tooth head at $\alpha_{02} \approx 47^\circ$ (point 7 in table 1). In this case, the angle β_w of the inclination of the wheel axis is less than the smaller of the values of the angle γ_{0i} at the selected design points.

6 Determination of the angle β_w of inclination of the grinding wheel axis

The design scheme for determining the angle β_w of the installation of the axis of the grinding wheel for radial-axial relieving of surfaces with a variable curvature profile is shown on Fig. 5.

In the coordinate system XYZ (Fig. 5), the Z axis is directed along the axis of the hob, the X axis is aligned with the axis of symmetry of the thread, XZ is the plane of the axial section of the hob, and the Y axis is perpendicular to this section. The plane of the axis of the grinding wheel is parallel to the Y axis, intersects the X axis at a distance a from the Z axis under an angle $\varphi_0 \neq 0$.

As shown in the lower projection of Fig. 5, the value of distance a is determined as:

$$a = r_{i \min} + 0.5b \tan \varphi_0 + r_{\text{wm}} / \cos \varphi_0 \quad (9)$$

where b is the thickness of the worm thread being ground with a minimum radius $r_{i \min}$.

The equation of the plane containing the axis of the grinding wheel is written in the form:

$$x(z) = a - z \tan \varphi_0 \quad (10)$$

Coordinates z_i of the intersection of normals recovered from points A_1 and A_2 with the plane of the axis of the grinding wheel:

$$z_{i(1,2)} = (a - x_{0i} + z_{0i} \tan \alpha_i) / (\tan \alpha_i + \tan \varphi_0) \quad (11)$$

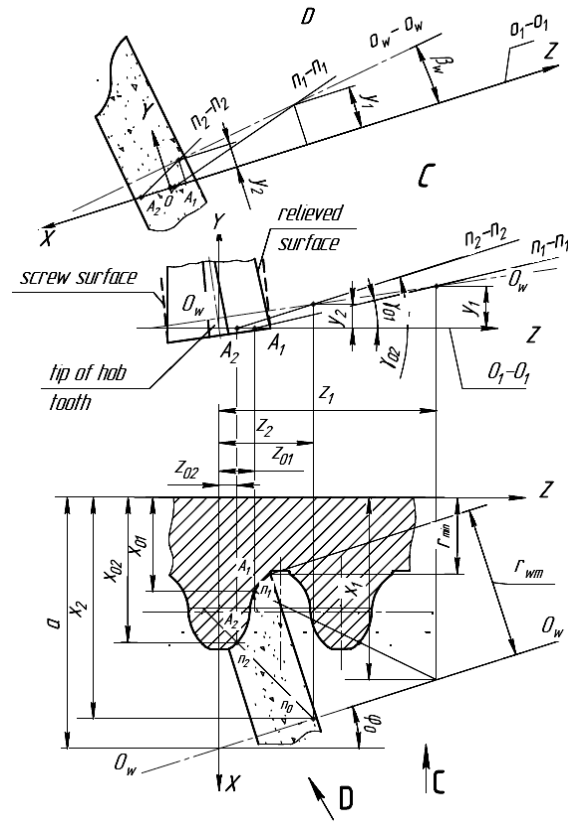


Fig. 5. The design scheme of the installation of the axis of the grinding wheel.

The two other coordinates $x_{i(1,2)}$ and $y_{i(1,2)}$ of each of the intersection points of the normals with this plane are determined through z_i . When determining the coordinate $y_{i(1,2)}$ we should consider the frontal projection (view C):

$$x_{i(1,2)} = a - z_i \tan \varphi_0; y_{i(1,2)} = (z_i - z_{0i}) \tan \gamma_{0i} \quad (12)$$

The angle β_w of the inclination of the axis of the grinding wheel is determined in the tangent function (from top view on the plane of the axis of the grinding wheel) by the formula:

$$\tan \beta_w = \frac{y_1 - y_2}{\sqrt{(x_1 - x_2)^2 + (z_1 - z_2)^2}} \quad (13)$$

7 Choice of angle φ_0 for relieving the tooth flanks

As mentioned above, turning the hob axis by the angle φ_0 , improves grinding conditions by increasing the curvature of the surface of the wheel at the points of contact with a processed surface, and reduces the difference between the angle β_w of the inclination of the wheel axis and the lead angles of the helical line of the cutting edges of the hob teeth. The choice of the angle φ_0 is determined by two restrictions. On the one hand, it is necessary to minimize the mentioned difference of angles β_w and γ_{0i} . On the other hand, the value φ_0 should not lead to pruning of the cutting edges of the teeth on the opposite side of the thread of the generating surface, i.e., the grinding wheel should enter the space without damaging the adjacent tooth.

Consider the issue of pruning prevention in more detail. The minimum distance between the back (flat) side of the grinding wheel and the opposite side of the thread of the generating surface takes place opposite to the most protruding point on the head of the hob tooth. Such a point is a contact point on the tooth head with a pressure angle $\alpha_{0i} \approx 27^\circ$.

Due to the difference between the angle β_w of the wheel inclination and the lead angle γ_i of the generating surface, the dangerous approach of the surfaces is shifted from the horizontal

plane (downwards when processing the right side of the tooth). Its value reaches maximum at the extreme point of intersection of the face sections of the outer wheel cylinder (radius r_{wm}) and the cylinder of the hob radius (r_F+h_k) at the contact point on the tooth head. The distance from this point to the horizontal plane for simple geometric reasons is determined by the formula:

$$l_{\max} = r_{wm} \sin \psi, \quad (14)$$

$$\text{where} \quad \psi = \arccos \frac{r_{wm}^2 + (r_{wm} + r_{\min})^2 - (r_F + h_k)^2}{2r_{wm}(r_{wm} + r_{\min})} \quad (15)$$

The linear displacement Δ of the back surface of the grinding wheel to the middle of the space at a length $l_{\max}$ in a screw projection onto the axial plane of the cutter to both tooth sides will be:

$$\Delta_{R,L} \approx \pm l_{\max} [\tan \beta_{wR,L} \cos \varphi_0 - p/(r_F + h_k)] \quad (16)$$

where the sign "+" relates to right side of the right-hand hob, the sign "-" to its left side.

In the angular dimension, this displacement is determined from the expression:

$$\tan \varphi_\beta = \Delta/(r_F + h_k) \quad (17)$$

The pruning of the cutting edges of the opposite side of the thread is guaranteed to be excluded if condition $\varphi_0 + \varphi_\beta \leq \alpha_{0i}$ is met, where the value α_{0i} is accepted for the key point 6 from Table 1. For preliminary calculation of the relieving parameters, the angle φ_0 should be taken equal to $\varphi_0 = 18^\circ$. Then, having determined the angle φ_β by formula (19), it is possible to correct the angle φ_0 by increasing it to the maximum permissible value

$$\varphi_{0\max} = \alpha_{0i} - \varphi_\beta \quad (18)$$

Given that increasing φ_0 reduces the needed value of the angle β_w , such an adjustment is advisable.

As an example, we will calculate the relieving parameters β_w and φ_0 as applied to the hob considered above, and study the features of the resulting side surfaces.

As the initial points for constructing the designed normals, we take the points: on the leg A_1 , at the contact point with pressure angle $\alpha_{01} \approx 27^\circ$ (key point 2 from table1), on head A_2 at the conjugation between the active profile and the transition curve with the angle $\alpha_{02} \approx 47^\circ$ (key point 7). The coordinates of these points and the exact values of the pressure angles are in table 1.

The maximum radius of the grinding wheel is assumed to be equal to $r_{wm} = 60$ mm, the minimum radius of the relieved surface on the space $r_{\min} = 66.215$ mm, the height of the grinding profile of the hob tooth is $H_1 = 23.785$ mm for both sides of the teeth of the manufacturing surface. The value of the turning angle of the relieving support to provide a rear angle in the middle section of the profile $\varphi_c = 12^\circ$ for both sides of the teeth.

The lead angles γ_{0i} of the relieved surface at the points A_1 and A_2 were shown in table 1 for each side of a tooth of the right-hand hob. On the right side of the tooth, at which the lead angle of the relieved surface is greater than the lead angle of the generating worm they are equal $\gamma_{01} = 13.598^\circ$, $\gamma_{02} = 6.439^\circ$. On the opposite, for the left tooth side we have $\gamma_{01} = -4.200^\circ$, $\gamma_{02} = -9.001^\circ$.

Negative values of lead angles γ_{0i} of the relieved surface at the points of arc sections of the profile mean its transformation in these sections into an analogue of the left helical surface, which is characteristic for single-thread hobs with a standard lateral rear angle of 4° or more, with the number of modules in the pitch diameter of 12 or more.

Table 2 summarizes the main stages of the calculations by means of the corresponding formulas.

Table 2. Calculation of the parameters β_w and φ_0 of the hob teeth.

Data	Formula number	Estimated Values		
		preliminary	final	
		Right	Left tooth side	
φ_0°		18.0	18.0	19.0
a ,	(9)	134.47	134.47	135.15
x_1	(12)	107.76	107.76	107.21
z_1	(11)	82.21	82.21	81.14
y_1	(12)	17.17	- 5.17	- 5.13
x_2	(12)	122.47	122.47	122.44
z_2	(11)	36.94	36.94	36.91
y_2	(12)	9.47	- 5.09	- 5.08
β_w°	(13)	9.19	-0.095	-0.064
ψ°	(15)	34.64	34.64	34.64
$l_{\max}$	(14)	34.11	34.11	34.11
Δ	(16)	2.81	2.49	2.47
φ_β°	(17)	8.93	7.95	7.89
$\varphi_{0\max}^\circ$	(18)	18.42	19.40	19.46

For the right side, the difference between the values of the initial φ_0 and the adjusted $\varphi_{0\max}$ angles in the preliminary calculation is less than 1° , therefore, a change in φ_0 is not required. For the left side, the final calculation is carried out by increasing the angle φ_0 to 19° , and recalculation of β_w , which however has changed very slightly and which in practice in both cases is rounded to the value -0.1° .

In [1, 2] it was shown that the discrepancy between the profiles of the grinding wheel surface and the hob tooth flank to be ground will be minimal if in the vicinity of the calculated contact point, the line of their contact will not be crossing the axial section of the hob, but touching it. This condition is satisfied in the case when the axis of the grinding wheel is inclined to the axial plane at an angle β_w , which is determined depending on the pressure angle α_0 , the lead angle γ_{0i} and the radius r_i of the relieved surface, taking into account the installation angle φ_0 of the wheel axis in the projection onto the horizontal plane, by the expression:

$$\tan \beta_w = \tan \gamma_{0i} \cos \alpha_0 / \cos (\alpha_0 + \varphi_0) \quad (19)$$

Table 3 shows the values of the angle β_w calculated according to this dependence for seven key points of the axial section of the right and left sides of the hob teeth.

Table 3. Values of the angle β_w calculated for the different sides of the hob tooth.

Point numbers	1	2	3	4
x_{0b} , mm	67.575	71.467	75.906	77.775
α_{0i}°	47.34	27.08	7.75	7.75
γ_{0i}° (right side)	21.207	13.598	8.576	8.373
β_w° (right side)	32.22	16.96	9.42	9.198
γ_{0i}° (left side)	-11.853	-4.20	0.446	0.435
β_w° (left side)	-18.882	-5.29	0.491	0.479
Point numbers	5	6	7	
x_{0b} , mm	79.25	84.083	88.095	
α_{0i}°	7.75	27.35	46.95	
γ_{0i}° (right side)	8.219	11.679	16.439	
β_w° (right side)	9.029	16.641	25.441	
γ_{0i}° (left side)	0.427	-3.625	-9.001	
β_w° (left side)	0.470	-4.590	-14.325	

In Fig. 6 (upper part) a graphical interpretation of the dependence (19) of the angle β_w on the hob radius is given for the right side of the teeth. Let us draw two boundary straight lines: one through the starting points A_1 and A_2 of the designed normals, and

the second horizontal line at minimum value $\beta_w = 9.19^\circ$ (from Table 3) until they mutually intersect in some point C. The sections of the profile that fell into the sector between the boundary lines have practically no organic error, since any straight line drawn through the intersection point C inside the sector also pass through two points of the axial profile of the relieved surface. Outside the obtained sector, the relieved profile has a small organic deviation from the given profile of the grinding wheel.

At the lower part of Fig. 6 a similar dependence of the angle β_w on the hob radius is shown for the left side of the teeth. One boundary straight line passes through the points of the line corresponding to the values of the radius at points A_1 and A_2 . The second line runs horizontally, touching the line at its upper point with a value of $\beta_w = -0.1^\circ$. The intersection of these lines forms a sector in which the organic deviation of relieved profile is practically absent.

8 Profiling of grinding wheels

Significant difficulties in the manufacture of worms and worm hobs with a profile of substantial and variable curvature are due to the fact that the circle dressing copiers included in the delivery of grinding machines are not suitable for reproducing such a profile [1, 2]. At present, with the development of CNC, ruling devices with diamond rollers and programmed control of their movement relative to the working surface of the grinding wheel have gained a certain distribution.

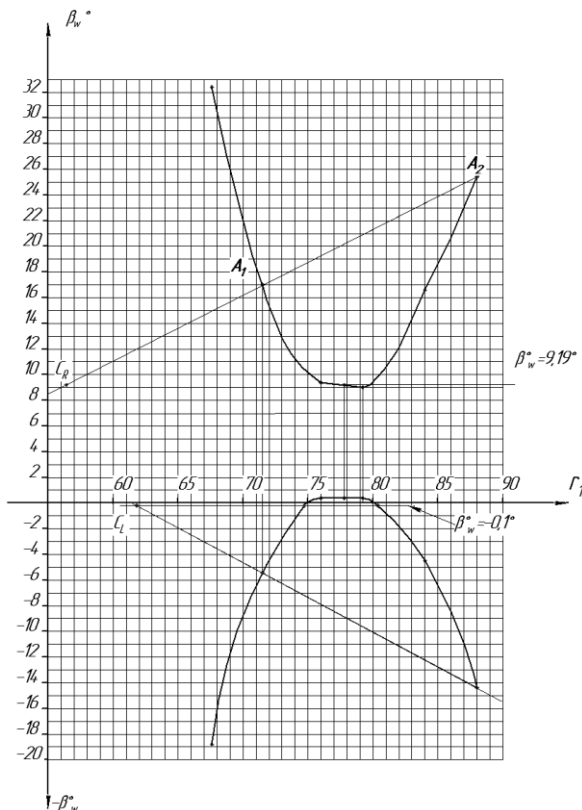


Fig. 6. Angles β_w vs. the current radius of the hob.

The characteristic of the surface of the grinding wheel, i.e., the line of its contact with the surface to be ground, is oriented along the axial section of the latter. Therefore, profiling of the wheel in its axial section with a sufficient degree of approximation reproduces the section of the surface being ground that is normal to a helical line with the lead angle equal to β_w . That is, the axial profile of the grinding wheel for single-thread hobs practically repeats the profile of the basic rack. In this case, the value of the angle α_w of the wheel profile in a rectilinear section is determined from the expression:

$$\tan \alpha_w = \tan(\alpha_{0\min} + \varphi_0) \cos \beta_w \quad (20)$$

In general, the coordinates of the axial profile of the grinding wheel, based on the coordinates of the axial section of the relieved surface of the hob teeth, the lead angles of this surface and the found angle of inclination of the wheel axis, are determined by the following expressions:

$$\begin{aligned} z_w &= z_{0i} / (\cos \beta_w + \tan \gamma_{0i} \sin \beta_w); \\ x_w &= x_{0i} + z_w^2 \sin 2\beta_w \tan \gamma_{0i} / (4x_{0i}). \end{aligned} \quad (21)$$

9 Sensitivity to the hob regrinding

If the relieving of surfaces with a variable curvature profile is performed by the radial-axial method, then the axial component $k_x = k \sin(\alpha_{0i} + \varphi_c) / \cos \alpha_{0i}$ of the relieving parameter included in formula (6) is variable along the profile. In this case, strictly speaking, the profile distortion during regrinding of the hob teeth on the front surface is inevitable. However, the considered profile has the points with identical pressure angles α_{0i} (for example, at the conjugate points of large and small radii on the tooth head and leg) and the lead angles γ_{0i} of the relieved surface, and therefore, at these points and in the areas between them, the profile deviations as regrind are minimal.

Conclusion

The relieving process for tooth flanks of worm hobs for cutting Novikov gears with two lines of action has been studied. The results of this research are shown below.

1. Change of the relieved surface lead angle of hob teeth has a parabolic dependence on the radius and pressure angle of the tooth profile. The relieved tooth surfaces of such hobs are organically characterized by the presence of paired profile points with the equal lead angles.

2. The minimum rear angle of the hob teeth takes place near the hob pitch cylinder, and the installation angle φ_c of the relieving support of the grinding-relieving machine is necessary to be calculated based on the minimum value of the rear angle 4° .

3. For both right and left sides of the teeth it is preferable to assign the installation angle φ_0 for the grinding wheel axis at 18° . Also, pruning on the opposite side of the generating surface of the cutting edges has to be prevented.

4. At the relieving of the left side of the right-hand hob teeth the directions of the lead angles of the relieving surface and the helical surface of the cutting edges have different signs. Accordingly, the angle β_w of inclination of the grinding wheel axis to the horizon can also have a sign opposite to the lead angle γ_1 of the hob generating surface.

The technique has been developed for calculating the parameters of installation, movement and profile of the disk grinding wheel for the relieving process of the tooth flanks. In the axial section of the hob it allows to obtain at least 4 points of contact between the wheel and the surface to be ground. Thus, the organic grinding error is minimized during the profiling of the grinding wheel, and generated profile is as close as possible to the profile of the hob basic rack.

The above technique is also proposed to be used in the development of control programs to set up the grinding-relieving machine and the dressing mechanism of the grinding wheel for worm hobs manufacturing on relieving machines with the numerical program control.

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Comparison of corrosion resistance of traditional and modern alternative pipeline steel

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Abstract: The paper presents results of research aimed at comparing selected properties of two types of pipe steels, conventional 17G1S-U steel and alternative S960QL steel. The steels were compared in terms of their chemical composition, microstructure, hardness and corrosion resistance. Static electrochemical corrosion tests were performed on the materials, namely electrochemical impedance spectroscopy (PEIS) and linear polarization (LP) in two electrolytes. The results show different chemical composition and microstructure of materials. 17G1S-U steel has a row-like ferritic-pearlitic structure with a hardness of 200 HV 0.1. S960QL steel has a martensitic structure with a hardness of 353 HV 0.1. Steels showed very close corrosion rate in both electrolytes.

Keywords: PIPELINE STEEL, MICROSTRUCTURE, HARDNESS, CORROSION BEHAVIOR, ELECTROCHEMICAL IMPEDANCE SPECTROSCOPY, LINEAR POLARISATION

1. Introduction

Worldwide oil and gas consumption is increasing, resulting in the construction of new piping systems. High-strength steels are used to build new oil and gas pipelines. In order to reduce the cost of pipeline construction, great emphasis is placed on construction and material selection. Therefore, the wall thickness of the pipeline is reduced, resulting in a reduction of the economic cost during pipeline building. Over time, however, installed pipes may be attacked by external or internal corrosion processes. These processes can take place in above-ground parts, where degradation identification is easier. The problem arises in parts that are placed in soil, where acts also biological corrosion by soil microorganisms [1,2]. It is therefore important to monitor residual lifespan of pipe steels by both static and dynamic corrosion tests. The corrosion properties of materials under simulated laboratory conditions depend on the working procedures and the composition of the test solutions. Therefore, stable conditions (temperature, solution composition, electrode condition) and repeatability of the tests must be ensured.

The paper deals with testing of two materials - traditional pipeline steel of 17G1S-U and its potential replacement of S960QL in terms of their chemical composition, microstructure, hardness and corrosion properties. Static electrochemical corrosion tests such as electrochemical impedance spectroscopy (PEIS) and linear polarization (LP) in two test solutions were performed on the materials [3].

2. Materials and methods

2.1 Materials

The first type of material: 17G1S-U steel (YS 365-490 MPa) is used in industry for the production of welded structures, pipes, gas pipeline components operating under pressure at temperatures from -40°C to +478°C, at operating pressure up to 2.2 MPa. Pipes carrying non-corrosive gas are most commonly spirally welded. Such pipes are designed to operate under a pressure of up to 7.4 MPa with an external anti-corrosion coating.

The second type of material: S960QL steel (YS_{min} 960MPa) is high strength structural steel in hardened and tempered state, which ensures high strength and good abrasion resistance. Because of these properties, this steel is used in transport and mining industries where excessive load and excessive abrasion damage are common.

2.2 Chemical composition of materials

To determine the exact chemical composition of both steels, the Belec Compact Port - portable hybrid spectrometer was used. The measurement was performed after appropriate surface cleaning three times on each sample.

2.3 Metallographic analysis of materials

Metallographic sections with dimensions of 25×25×15 mm were made to determine the microstructure as well as hardness of materials. The samples were first degreased in acetone and then purified ultrasonically in distilled water for 10 min. Sample grinding was performed on wet sand papers with grit size of #200, 400, 600, 800 and 1200. The samples were etched in Nital etching agent. Optical microscope Keyence VHX-5000 was used to observe the microstructure.

2.4 Measurement of hardness

The microhardness of the materials was determined by the Vickers hardness test using microhardness tester Shimadzu HMV2. The test load was 0.9807 N (HV 0.1), the dwell time was 15s. The spacing between individual indentations was 1mm. Ten measurements were performed on each material to determine the average hardness.

2.5 Measurement of corrosion characteristics of materials

The tests were performed on test samples with dimensions of 25×25×15 mm. The samples were prepared by grinding on a set of sandpapers with a grit size of #200, 400, 600, 800 and 1200, followed by rinsing in ethanol and drying with warm air.

When choosing a suitable solution we used the knowledge from the paper [4] and we chose the solution NS4, which is suitable for measurements on pipelines located in the soil. The composition of the solution per litre was as follow: 483 mg NaHCO₃, 122 mg KCl, 137 mg CaCl and 131 mg MgSO₄. The reference solution in the experiment was chosen 3.5% NaCl solution.

The experiment was performed using a 3-electrode connection in a corrosion cell, Fig. 1. As the working electrode a sample was connected, the auxiliary electrode was a platinum electrode, the reference electrode was a saturated calomel electrode (SCE). The measurement was performed using BioLogic SP-150 instrument at 22°C and processed with EC-Lab v11.27software.

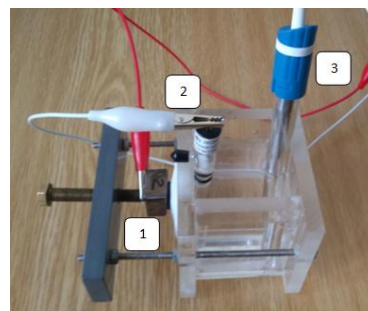


Fig. 1 Three-electrode connection (1-sample, 2-SCE, 3-platinum electrode)

The measurement was performed on a surface area of 0.636 cm². Each measurement started with measurement of potential over time (EVT) for 30 min followed by potentiostatic

electrochemical impedance spectroscopy (PEIS). The range of measured frequencies was from 10 mHz to 20 kHz and the frequency distribution was 10 points per decade. The last part of the experiment was the linear polarization (LP) method. The potential range was set from -0.25 V to +0.25 V [1].

For processing the PEIS measurement, Bode imaging of the impedance Z versus frequency and phase angle versus frequency was used, Fig. 2. The measurement evaluation was based on the relationship at very low frequencies (1),

$$Z = R_s + R_p \quad (1)$$

and also at very high frequencies (2),

$$Z = R_s \quad (2)$$

where Z means impedance, R_s is resistance of solution (electrolyt) and R_p is polarisation resistance of material measured.

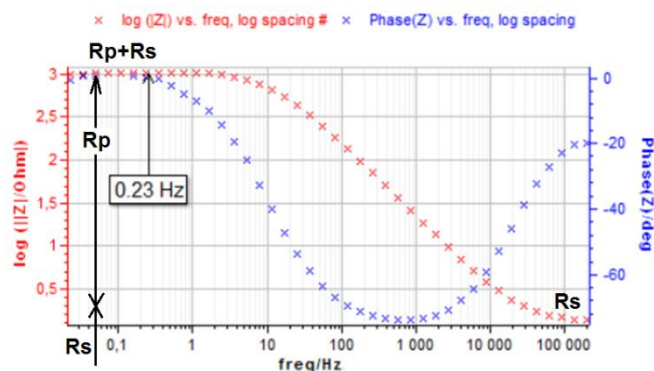


Fig. 2 Bode impedance diagram

The aim of this measurement is to determine the polarization resistance R_p , which corresponds to the resistance of the measured material. High polarization resistance means high corrosion resistance of the material.

Imaging in Tafel coordinates was used to process the LP measurement, Fig. 3. The tangent lines β_c and β_a were put over strongly linear parts of curves. The shift from E_{corr} to the anodic and cathodic part of the curve was approximately $\pm 100\text{mV}$.

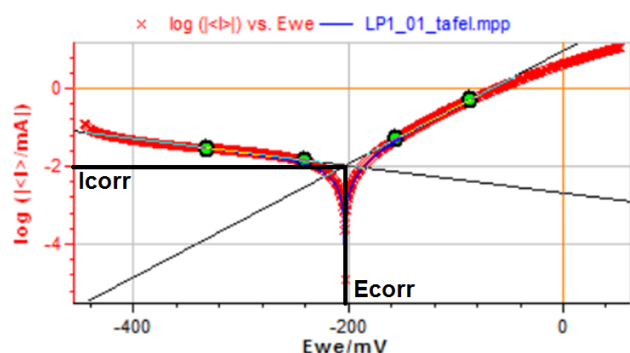


Fig. 3 Tafel plot

The aim of this measurement is to determine the corrosion rate r_{corr} of the material based on the corrosion potential E_{corr} and the corrosion current density I_{corr} .

3. Results and discussion

3.1 Chemical composition of materials

Two types of materials were chosen for the study of electrochemical properties of pipeline steels, their chemical composition declared by producer and determined by Belec spectrometer is given in Table 1.

Table 1 Chemical composition of materials, wt. %

elements [%]	material			
	measured		declared	
	17G1S-U	S960QL	17G1S-U	S960QL
Fe	98.04	98.24	bal.	bal.
C	0.163	0.114	0.15-0.20	max. 0.20
Si	0.355	0.366	0.40-0.60	max. 0.50
Mn	1.026	0.945	1.15-1.55	max. 1.60
P	0.007	0.015	max. 0.03	max. 0.02
S	0.011	0.011	max. 0.035	max. 0.01
Cu	0.099	0.045	max. 0.30	max. 0.50
Al	0.068	0.051	---	---
Cr	0.043	0.032	max. 0.30	max. 0.80
Mo	0.044	0.041	---	max. 0.70
Ni	0.052	0.051	max. 0.30	max. 2.00
V	0.009	0.009	max. 0.12	max. 0.10
Ti	<0.002	<0.002	---	0.05
Nb	0.015	0.015	---	0.06
Co	0.039	0.037	---	---

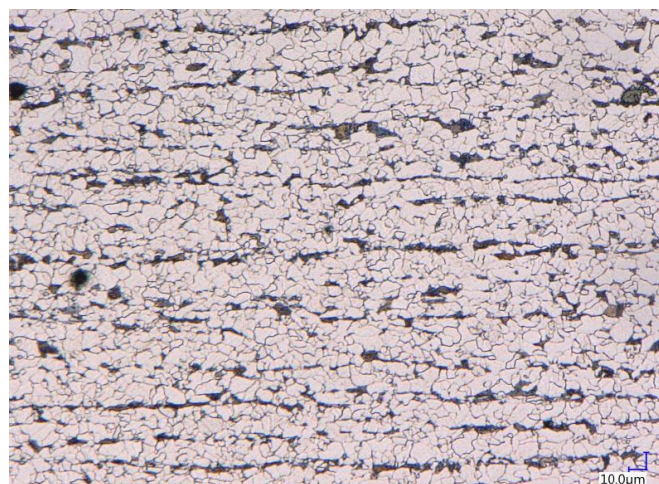
The measured chemical element contents of both steels are within the limits specified by the manufacturer. The content of elements, determined analytically, was used in the calculation of the carbon equivalent, according to the formula (3):

$$C_{ekv} = C + \frac{Mn}{6} + \frac{Cr+Mo+V}{5} + \frac{Cu+Ni}{15} \quad (3)$$

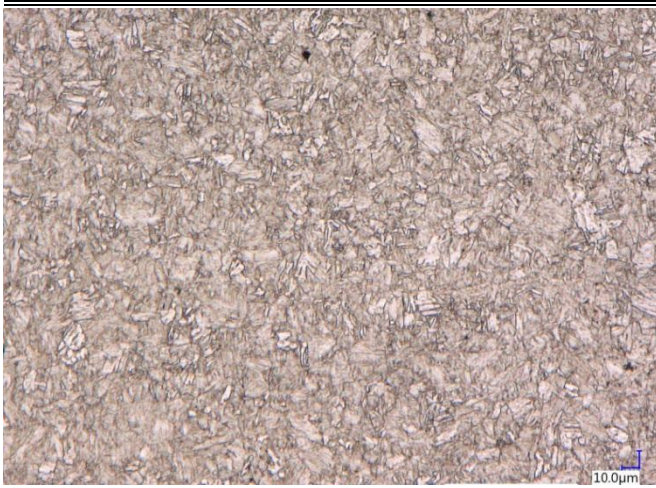
Carbon equivalent of 17G1S-U steel determined by calculation according formula (3) equals to 0.363, and for S960QL steel equals to 0.294. In terms of weldability, they are therefore materials with good weldability.

3.2 Metallographic analysis of materials

The microstructure of the materials is shown in Fig. 4. Metallographic study proved row-like ferritic-pearlitic structure in 17G1S-U steel, and martensite structure in S960QL steel. The weak point of ferritic-pearlitic microstructure of 17G1S-U steel is the arrangement of pearlitic grains in rows, which can cause delamination damage during welding and subsequent stresses, especially in the direction perpendicular to the lines. The problem of 17G1S-U steel delamination was addressed by Shapoval [5] in 1984. In contrast, the martensitic structure of the S960QL steel is homogeneous, with no apparent directionality.



a) microstructure of 17G1S-U steel



b) microstructure of S960QL steel

Fig. 4 The microstructure of analysed materials

3.3 Hardness of materials

The hardness of 17G1S-U steel was 202 ± 11 HV 0.1 and for S960QL steel hardness was 353 ± 8 HV 0.1. The higher hardness of S960QL steel is a natural consequence of its martensitic structure. For martensitic structures, however, it is important to ensure the minimum required impact toughness of the material, especially under low operating conditions of the gas pipeline, which corresponds to the winter season in the given climate zone [6].

3.4 Corrosion characteristics of materials

The open circuit potential (OCP) of steels became stabilized after approximately 15 minutes of immersion in the measuring solution. The resulting OCPs can be found in Table 2. In 3.5% NaCl solution, S960QL steel had about 0.03V more negative OCP compared to 17G1S-U steel. In NS4 solution situation changed, 17G1S-U steel had about 0.08V more negative potential compared to S960QL steel. A more negative potential indicates less material nobility. Corrosion potential can show corrosion tendency but not corrosion rate. Corrosion current is a kinetic value and potential is thermodynamic value. Corrosion rate is proportional to corrosion current (corrosion current density). Corrosion current will be measured by LP method.

Table 1 Open circuit potential of materials

solution	material	
	17G1S-U	S960QL
3.5% NaCl	-0.59 ± 0.01 V	-0.62 ± 0.01 V
NS4	-0.69 ± 0.04 V	-0.61 ± 0.03 V

The results of the PEIS measurement were depicted by Bode Impedance diagram, Fig. 5 and Fig. 6.

Results of polarization resistance R_p in NS4 solution for 17G1S-U steel varied between 1253 and 1314 Ohm and for S960QL steel between 1213 and 1849 Ohm.

The polarization resistance R_p of 17G1S-U steel measured in 3.5% NaCl solution varied between 1387 and 1555 Ohm. For S960QL steel, the polarization resistance varied between 1030 and 1474 Ohm.

The Tafel curves measured in the NS4 solution are shown in Fig. 7 and in 3.5% NaCl solution in Fig. 8. The resulting processing of all measurements using Tafel curves is shown in Table 3.

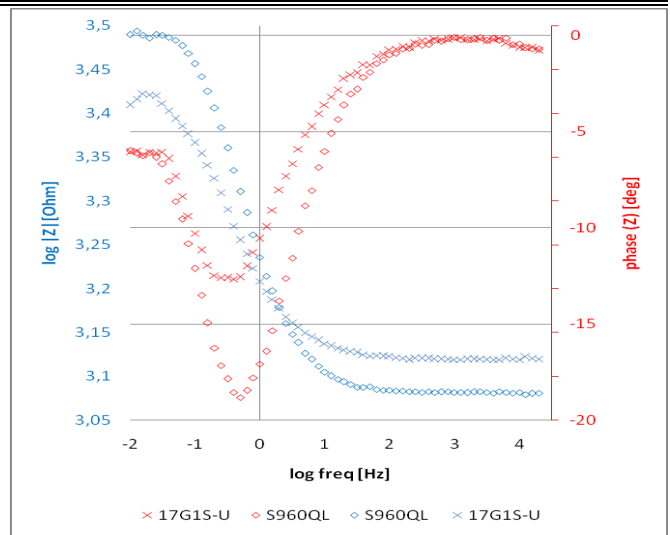


Fig. 5 Average measured curves obtained by PEIS in NS4 solution

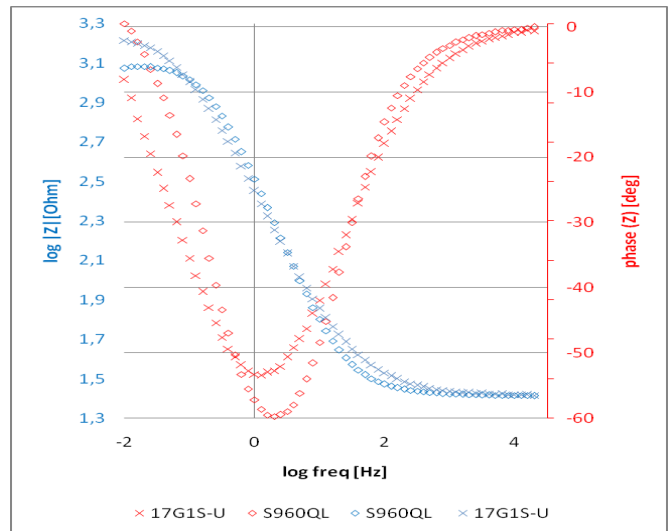


Fig. 6 Average measured curves obtained by PEIS in 3.5% NaCl solution

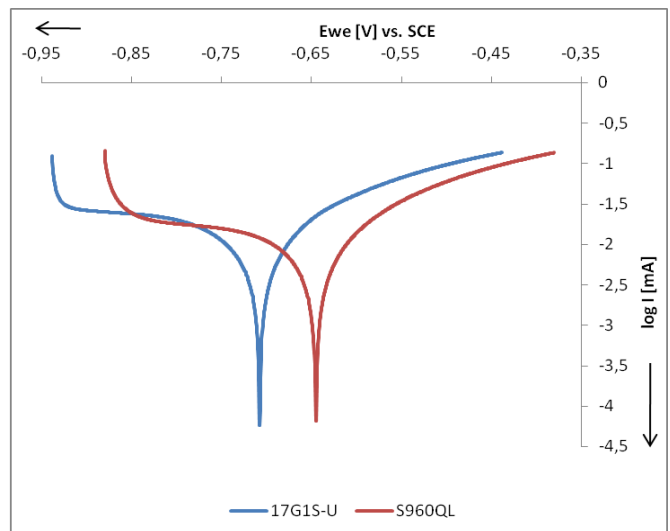


Fig. 7 Average measured curves obtained by LP in NS4 solution

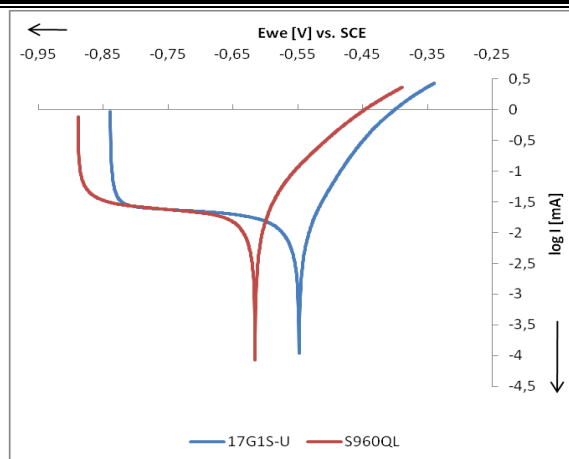


Fig. 8 Average measured curves obtained by LP in 3.5% NaCl solution

In NS4 solution, material 17G1S-U had a more negative corrosion potential E_{corr} compared to S960QL material. However, the corrosion current density I_{corr} , describing the kinetics of the corrosion process at steady conditions, differs minimally for both materials. In 3.5% NaCl solution, material S960QL had a more negative corrosion potential E_{corr} than material 17G1S-U. But the corrosion current density I_{corr} was again almost the same for both materials. The corrosion rate r_{corr} calculated from Tafel curves in NS4 solution reached values 0.406-0.442 mmpy for 17G1S-U material and for S960QL material corrosion rate r_{corr} varied between 0.417-0.460 mmpy. Corrosion rate r_{corr} identified in 3.5% NaCl solution reached values 0.502-0.526 mmpy for 17G1S-U and 0.509-0.573 mmpy for S960QL. In terms of the achieved corrosion current densities I_{corr} and corrosion rates r_{corr} , the 3.5% NaCl solution appears to be more aggressive than the NS4 solution, simulating more closely soil conditions.

Table 2 Results obtained by linear polarization of materials

solution	NS4	
material	17G1S-U	S960QL
E_{corr} [mV]	-698 ± 12	-651 ± 22
I_{corr} [uA]	12 ± 0.6	12 ± 0.6
r_{corr} [mmpy]	0.42 ± 0.02	0.44 ± 0.02
β_c [mV]	654 ± 70	497 ± 139
β_a [mV]	81 ± 9	74 ± 4
solution	3.5% NaCl	
material	17G1S-U	S960QL
E_{corr} [mV]	-544 ± 23	-588 ± 51
I_{corr} [uA]	14 ± 0.8	15 ± 0.9
r_{corr} [mmpy]	0.51 ± 0.01	0.54 ± 0.03
β_c [mV]	472 ± 42	727 ± 65
β_a [mV]	204 ± 3	204 ± 7

4. Conclusion

The problem of corrosion of gas pipelines is much more complicated, the environments in which they operate have a great influence on their service life. Therefore, it is necessary to know the properties of the material in particular environment. The experiment was focused on pipes located in a soil, where selected materials (17G1S-U and S960QL) were tested in two solutions (NS4 and 3.5% NaCl).

Measurement of two types of pipe steels revealed following:

- The materials have different microstructures. The type of microstructure is reflected in the basic properties of steels, hardness and abrasion resistance (although later has not been analyzed in the paper).
- Measurement of the polarization resistance showed minimal differences in R_p values for both solutions.
- I_{corr} and r_{corr} values, which describe the kinetics of the corrosion process, have very close values, taking into account the standard deviation of repeated measurements.

Based on the experiments carried out, it can be concluded that the S960QL steel can serve as a potential equivalent to traditional 17G1S-U pipeline steel in terms of its corrosion characteristics. For a comprehensive assessment of its resistance, it is necessary to take into account other important properties - mechanical properties and resistance to brittle fracture.

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Corrosion of high purity magnesia refractories from iron-rich slags of ferronickel enrichment in OBM converters

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Abstract: The corrosion of high purity MgO refractories from molten iron-rich OBM converter slag has been investigated by performing experiments in an electric laboratory furnace. Corrosion tests were conducted at 1600 - 1650°C for 1, 2 or 4 hours. Following lab experiments, the mineralogical phases of MgO refractories, slags and the intermediate infiltration zone were identified by SEM/EDS. It has been shown that Fe infiltrates and replaces Mg in MgO refractories, thus forming magnesio-wustite ((Mg,Fe)O). The maximum depth of the slag infiltration zone was estimated to 1.31 mm in the laboratory test where slag was in contact with the refractory for 4 h, at 1650°C. Thermodynamic modeling of the slag and refractories equilibrium conducted with FactSage 7.0, confirmed the formation of magnesio-wustite in the intermediate zone, with progressively higher substitution rate of Mg by Fe. This increasing substitution results in a decrease of the melting point of magnesio-wustite, thus causing the collapse of the refractories even at temperatures lower than the operational temperature.

Keywords: REFRACTORIES, MAGNESIA, FERRONICKEL, FENI, CORROSION, SLAGS, CONVERTER, OBM

1. Introduction

The conventional RKEF (Rotary Kiln-Electric arc Furnace) method is applied by LARCO SA at the metallurgical plant in Larymna, Greece, to treat Greek laterites for ferronickel production. The nickel content in the ferronickel alloy produced from the five submerged electric arc furnaces is around 10%. In order to refine the alloy and meet the market specifications, this is further enriched by prolonged blowing of oxygen in OBM converters. The molten alloy is fed in the converters at temperatures around 1350°C and due to the highly exothermic reactions of impurities oxidation and preferential iron oxidation to iron oxides, thus forming an iron rich slag, the working temperature inside the converter is usually around 1600 – 1650°C, but often reaches 1700°C. Due to high temperatures in OBM converters and the corrosiveness of the iron-rich converter slags, the refractory lining receives both thermal and chemical stresses, resulting in refractories damage and furnace lining failure thus posing significant risks to both the operations' and personnel's safety. In addition, the total cost of refractories replacement including the cost of magnesia refractories, the labor cost for lining replacement and the inevitable waste of operational time is high.

Chemical corrosion of magnesium refractories is a major problem in pyrometallurgy. There have been several research studies on the mechanisms of magnesia refractories corrosion by various slags including ferronickel ones, however, almost all of them refer to the corrosion of refractories of electric arc furnaces of both steel and iron alloys production. However, in these systems the working temperature as well as the iron content are significantly lower compared to ferronickel converter slag and, consequently, the refractory corrosiveness is lower.

Porosity plays an important role in the corrosion of refractories. If the surface of the refractories has no pores, the corrosion phenomenon occurs only on the front surface of the lining. Otherwise, molten slag penetrates inside the refractory mass and chemical reactions take place both on the external surface as well as in the refractories, resulting in the reduction of the thickness of the refractory lining at a faster rate. On the other hand, the temperature is not constant throughout the mass of the refractory but, instead, a temperature gradient is developed with the maximum temperature occurring at the refractory and slag interface. It is therefore quite possible that molten material penetrating the lining will progressively solidify forming new phases when the prevailing temperature is lower than the melting point.

In almost all studies of refractory-slag corrosion systems presented in literature [1-4], the slag is characterized by high concentrations of Si and Ca oxides, in contrast to the slag of Fe-Ni converters of this study, where slag consists almost exclusively of iron oxides and Si and Ca oxides are below 6%. This creates different corrosion conditions, due to the more aggressive nature of FeO that has not been studied in detail to date.

2. Materials and Methods

Experiments were conducted using a sample of OBM slag from LARCO metallurgical plant, Greece, and high purity (type A) commercial magnesia refractories (CIMM Group Co, China) used in the same plant for OBM converter lining. Chemical analyses of both samples have been performed following digestion and AAS analysis. The results are given in Table 1.

Table 1: Chemical analysis of slag and refractory sample

Elements	Slag (%)	Refractory (%)
FeO	64.35	
Fe ₂ O ₃	17.64	0.90
NiO	0.42	
CoO	0.04	
SiO ₂	6.28	1.30
CaO	5.93	1.40
MgO	2.85	96.00
Al ₂ O ₃	1.59	0.40
Cr ₂ O ₃	0.82	
MnO	0.08	

X-Ray diffraction of magnesia refractories sample indicated the presence of magnesia (periclase, MgO) and quartz (SiO₂) (Fig. 1)

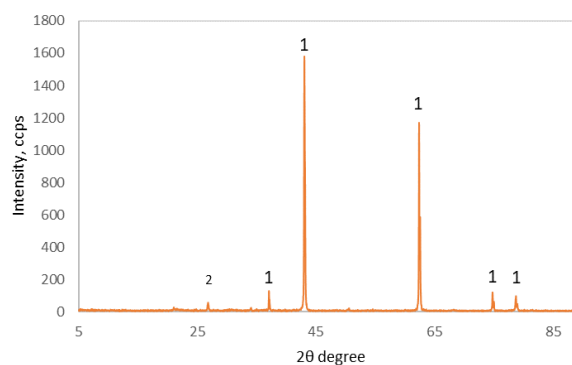


Fig. 1 XRD pattern of magnesia refractories (1: MgO, 2: Quartz)

In order to investigate the corrosion of magnesia refractories in contact with the OBM converter slag, laboratory experiments were conducted using magnesia crucibles constructed from refractories which were formulated into cubes with 7cm side and a cylindrical hole 4.5 cm in diameter and 5 cm deep (Fig. 2). Experimental tests were conducted using a laboratory electric furnace. A slag sample of 100 grams was placed in each crucible and the furnace operated with a heating program of 4.5°C/min at ambient atmosphere. Tests were carried out at 1600°C and 1650°C for 1, 2 and 4 hours retention time (a total of 6 samples). Then, cross sections at the slag-crucible interface were made and thin polished sections were prepared for conducting Scanning Electron Microscopy (SEM) analyses and measurements. Thin polished sections were analysed using a JEOL 6380LV electron microscope, with 10 to 300,000

times magnification capability for morphological observation taking high quality Secondary Electron Images and Backscattered Electron Images. The microscope was also equipped with a system for elementary micro-analysis with scattered X-ray spectrometry (EDS, Energy Dispersive System) for qualitative and quantitative micro-analysis and element mapping.



Fig. 2 Crucible constructed from magnesia refractories

A theoretical simulation of the slag-refractory system has also been performed using the FactSage 7.0 thermodynamic software to understand the processes of their interaction. The databases FactPS and FToxide of FactSage were used.

3. Results and Discussion

By the use of the SEM microscope, images of the corrosion area were received for all six thin polished sections at the slag – refractory interface. Fig. 3 presents the cross section of the sample after contact of the slag with the refractory at 1650° C for 1 hour.

All samples are characterized by clear contact borders between refractory and slag. The slag on the right side is whiter due to the presence of high iron content, unlike the refractory on the left, which is almost pure magnesium oxide and dark in color. SEM analysis indicated formations of cellular patterns consisting of iron rich magnesiowustite in the periphery and successive magnesium or iron rich either thick or thin layers, in the center.

In all samples, an infiltration zone (intermediate or penetration zone) was present, where iron oxides from slag have penetrated the

refractories area. The depth of penetration depended on contact time and temperature and there were significant differences in the form of the penetration zone, with higher iron content near the slag which gradually decreases when moving towards the refractory. Significant differences were also observed at the same depth of penetration.

Using the SEM images, the infiltration depth of slag in the refractory mass was measured, as shown in Fig. 3. Then, the average values of slag infiltration zone depth and the infiltration rate were determined as shown in Fig. 4 and 5.

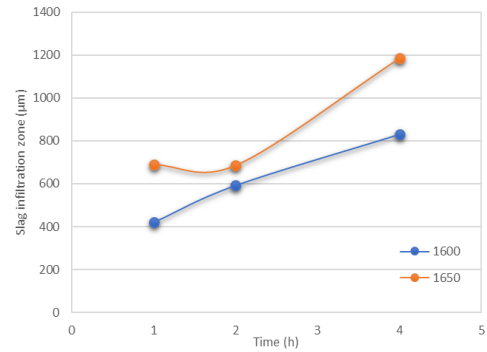


Fig. 4 Depth of infiltration zone (μm) vs. contact time at temperature 1600 and 1650°C

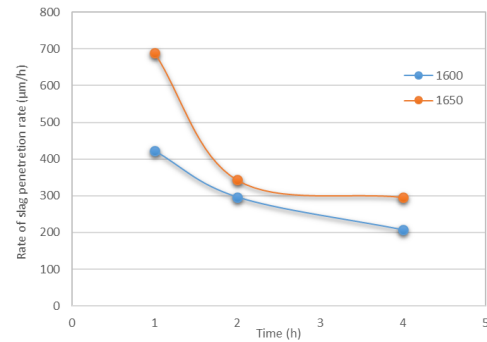


Fig. 5 Rate of slag infiltration (μm/h) vs. contact time at temperature 1600 and 1650°C

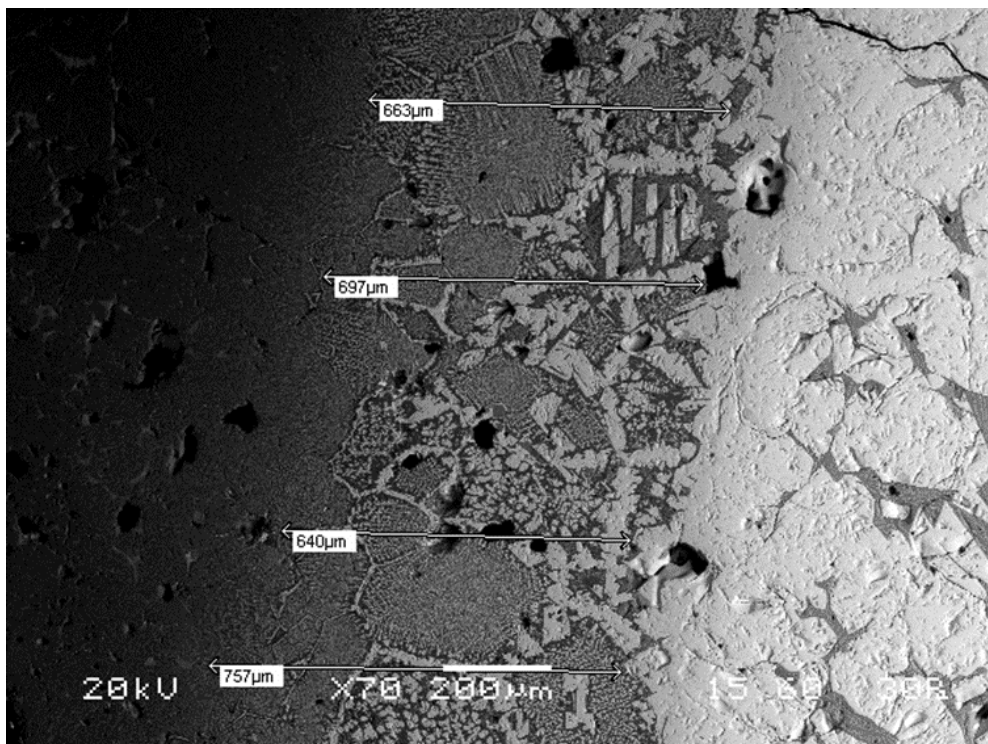


Fig. 3 Refractory-slag interface after treating at 1650° C for 1 hour -

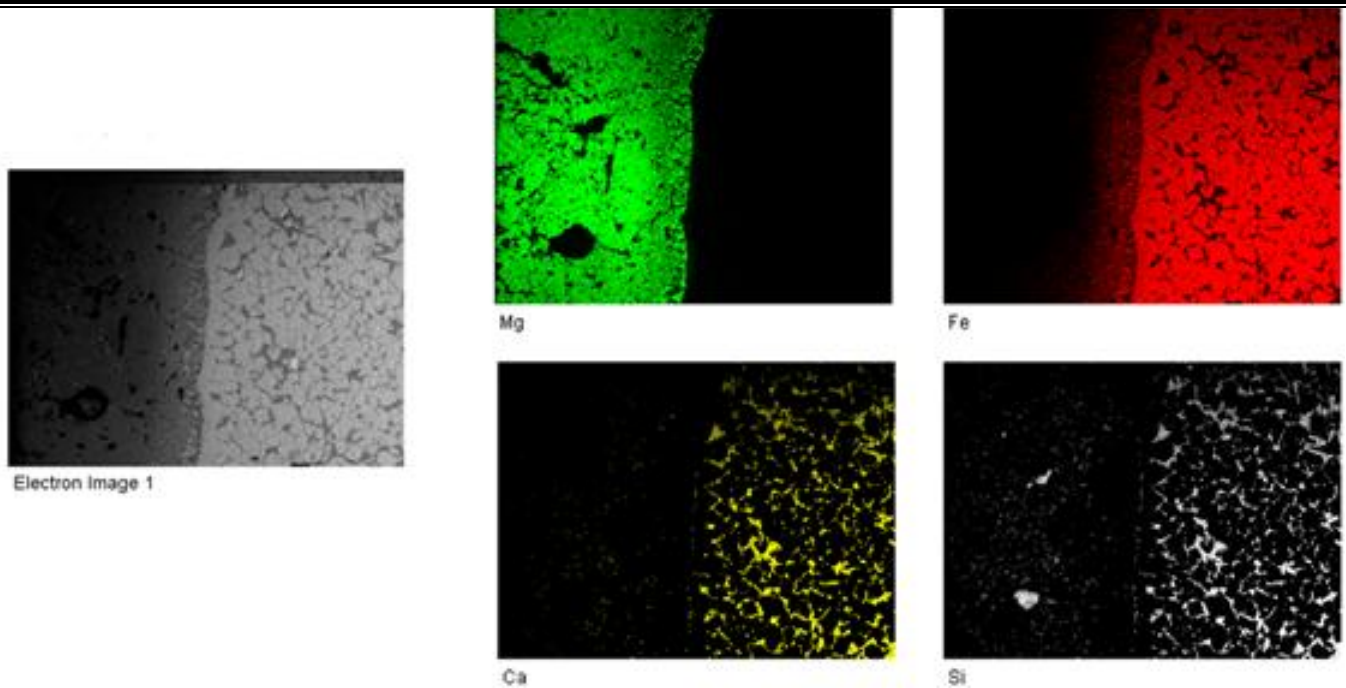


Fig. 6 Mapping images of a sample after treating at 1650° C for 1 hour (refractory on the left, slag on the right)-

As seen in Fig. 4, the increase of the retention time or the temperature results in the increase of the slag penetration, as it was expected. However, it was observed that there was no significant difference of the depth of infiltration zone when increasing the retention time from 1 to 2 hours at 1650°C, while the rate of slag penetration decreased with increasing retention time (Fig. 5).

In order to illustrate the spatial distribution of the basic elements in all the zones, mapping has been performed. The four chemical elements examined were magnesium, iron, silicon and calcium. Results are presented in Fig. 6, where the refractory mass is seen on the left, the metallurgical slag on the right, and the intermediate zone in the middle. In this zone, the iron of the slag reacted with the magnesium of the refractory. More precisely, liquid iron penetrated the refractory initially through the open pores on the surface. As penetration evolves, diffusion of iron takes place, gradually substituting magnesium in the periclase, thus forming magnesiowustite ((Mg,Fe)O) with magnesium following the opposite path to slag. The characteristics of the mixed magnesium iron oxide formed do not favor its dissolution in slag. The constant diffusion of iron in the penetration zone resulted in the creation of a thin tree structure, which was expanded to give large formations of iron-magnesium oxides of a characteristic white color as shown in the Fig. 7. In addition to the penetration of iron, there was also a small penetration of silicon and calcium.

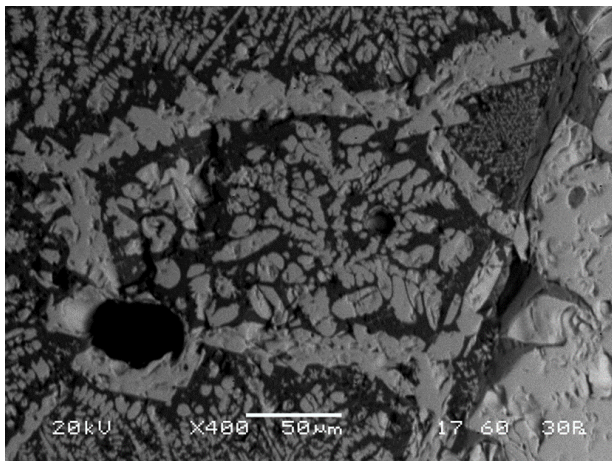


Fig. 7 Formations of iron-magnesium oxides with a characteristic white color

As it can be seen from Figures 3 and 6, the gradual decrease of Mg, the increase of Fe content and the formation of the intermediate zone was evident in all cases. The formation of calcium-silicate compounds in slags upon their cooling is also observed.

In addition to the experimental process, a theoretical simulation of the slag-refractory system was also performed using the FactSage 7.0 thermodynamic software. The software analyses and calculates the thermodynamic equilibrium of the metal oxides contained in the slag and the refractory at various mixing ratios declared by variable A varying from 0 (refractory only) to 100% (slag only). For A = 0 the composition of the oxide mixture corresponds to that of the refractory while for A = 1 the composition is the same as that of the slag. The total mass of the system theoretically examined is 1 g. The graphical representation of the mass of each constituent in the different phases produced in relation to the relative participation (variable A) of the refractory or the slag in the mixture, for temperature 1600°C is given in Fig. 8. As can be seen from this figure, only three phases appear in the system:

a) Pure solid magnesium oxide (MgO(s)), which is the main phase of refractories (A = 0); its percentage decreases linearly with increasing percentage of slag in the mixture and becomes zero for a value of A = 0.36 at 1600°C.

b) The mixture of magnesium and iron oxides (magnesiowustite, (Mg,Fe)O) symbolized by FactSage as MeO_A#1. The amount of this phase increases as iron gradually substitutes magnesium forming (Mg,Fe)O. Thus, the amount of solid MgO decreased as the amount of (Mg,Fe)O increases. In Fig. 8, the amount of FeO and MgO as constituents of phase MeO_A#1 are given. Upon increase of iron content in magnesiowustite the melting point of magnesiowustite decreased; when the latter is equal or below 1600°C, magnesium oxide reached its maximum value since magnesiowustite melted thus being incorporated into the liquid slag phase.

c) The liquid slag, which is symbolized as SLAGA # 1, was formed even at very low slag/refractory ratios. At 1600°C it started being formed when refractory impurities were liquified, but its quantity decreased exponentially when slag ratios in the mixture became very high.

4. Conclusions

Refractories interact with iron-rich converter slags both at their interface and through their pores. A zone of FeO infiltration in the

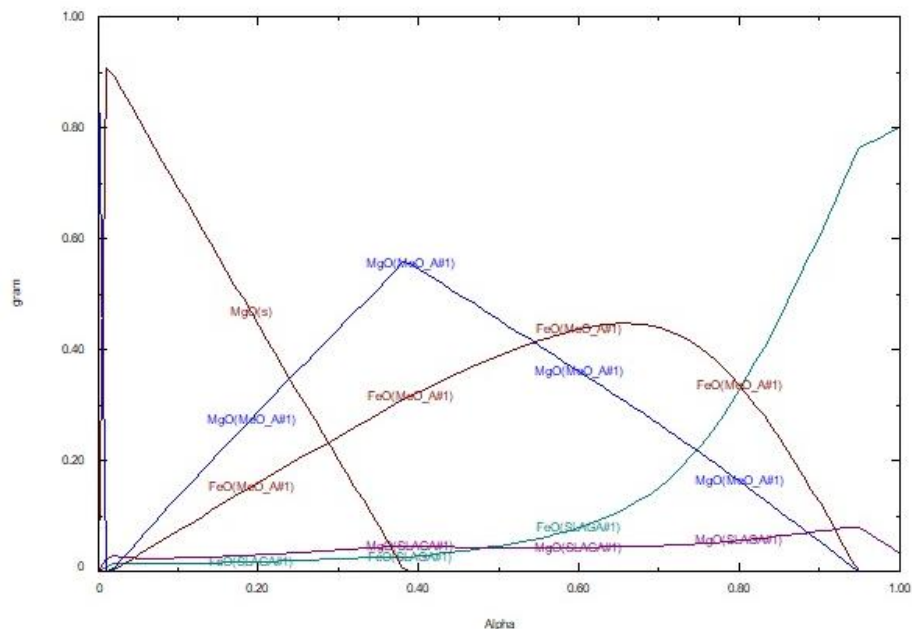


Fig. 8 Phases formed and their composition in the magnesia refractory – slag system at 1600 °C determined by FactSage vs. their mixing ratio (A=0 refractory no slag, A=1 pure slag)

mass of refractories is formed, which is characterized by the substitution of MgO by FeO thus forming magnesiowustite ((Mg,Fe)O). SEM analysis indicated the formation of a cellular pattern consisting of dark and light grey formations associated to the increased magnesium and iron content in magnesiowustite respectively, with iron rich magnesiowustite in the periphery and successive magnesium or iron rich thick or thin layers in the center.

The thickness of the penetration zone increased with contact time and temperature and reached up to 1.3 mm in 4 hours of processing. The rate of increase of the penetration zone decreased when retention time was increased.

The FactSage thermodynamic simulations indicated that pure solid magnesium oxide (periclase) in contact with the iron rich slag was progressively converted to magnesiowustite by substituting Mg by Fe. When the content of iron in magnesiowustite substantially increased, the consequent decrease of the melting point of

magnesiowustite, resulted in melting of magnesiowustite and its dissolution in the liquid slag, thus contributing to the refractory collapse and failure of refractory lining.

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Characterization of Sri Lankan brick clays for more advanced industrial uses

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Abstract: Brick clay is a well known type of clay which is available at various locations in world and the name was realized because of the constantly usages in the industry of brick manufacturing that based on the condign characteristic with the necessitate features for the bricks. The investigations of the fundamental and advanced physic-chemical characteristics of a type of Sri Lankan brick clay and bespeaking of the more advanced industrial uses of such clays are the ultimate realities of the existing study. The collected clay samples from a particular location were fundamentally characterized by testing the natural moisture content, grain size distribution, acidity of raw clay samples and the water absorption, bulk density, porosity and mechanical strengths of prepared bricks under 800°C using such clay. In addition that those clay samples were characterized using X-ray fluorescence (XRF) spectrophotometer, Fourier transform infrared (FT-IR) spectrophotometer and X-ray diffraction (XRD) spectrophotometer. As the retributions of those analyses, mainly observed 6.57 of P^H , uniformly graded grain size distribution, 37.36% clay and silt portion according to the weight and 21.45% natural moisture content from raw clay samples and 19.88% water absorption, 3.15 gcm^{-3} of bulk density, 62.70% of porosity, 17.25MPa compressive strength, 0.31MPa splitting tensile strength from prepared bricks. The compositional analysis results showed 84% of Fe content with trace amounts of Ti, Ba and Ca, presence of calcite, quartz and more Fe minerals while lacking of any hazardous elements or compounds such as the heavy metals. By considering the characters of brick clay, it is possible to recommend this clay for the applications of water treatments, ceramic manufacturing, refractory material and sorption (adsorber) material for removal of metals.

Keywords: BRICK CLAY, PHYSICAL ANALYSIS, CHEMICAL AND COMPOSITIONAL ANALYSIS, MECHANICAL STRENGTH, ADVANCED USES

1. Introduction

Clays are a group of minerals which are available in large number of locations around the world while having a series of various characteristics including physical, chemical and mechanical characteristics. In addition, the mineralogy of clay plays a dominant role in the industrial uses. According to the foremost characteristics of clays, less permeability and high porosity are really useful as a raw material. Among the traditional applications of clays the followings are mainly highlighted [1, 2, 3, 4, 5, 6, 7].

- Pottery industry
- Building material industry
- Ceramic and porcelain industry

In addition that there were disclosed the suitability of different clay types for different advanced industrial applications through the outcomes of the valuable researches as given in the below [2-9].

- Water treatment applications
- Manufacturing of advanced composite materials
- Heat resistant materials (refractory)

In the characterizations of clays for advanced technological uses, mineralogy and the chemical composition of such clay would be considered as a foremost property because the clay minerals may have some specific characterizations which are useful in the specific advanced industrial tasks such as the adsorption, ion exchange and heat resistant. The common minerals that commonly found in clays are kaolinite, montmorillonite and some other ferrous minerals. When concerning their characteristics, they were identified as strong adsorbers for some metallic elements [1, 2, 4, 6, 7-10].

There were expected to investigate the prominent characteristics of Sri Lankan bricks clay and disclose the feasibility of using in advanced industrial applications through the existing research.

2. Materials and Methodology

In the collection of clay samples, Maduragoda area was selected as the study area which is a famous area for the manufacturing of bricks in Sri Lanka because of the availability of adequate clays for such industry around the region of Maduragoda in

Kurunegala district. Therefore, this clay type is usually known as brick clays. A representative clay sample of brick clay is shown in the Fig.1.



Fig. 1. A brick clay sample

The collected clay sample were carefully stored in polythene bags until performing of the experiments.

2.1. Analysis of Natural Moisture Content

The initial weights of two selected raw brick clay masses were measured using an analytical balance. The weighed clay samples were oven-dried for 24 hours under the temperature of 110°C and the dry weights of such clay samples were measured using the same analytical balance. The moisture contents of those clay samples were computed using following equation [5-10].

$$\text{Moisture content} = [(W_R - W_D) / W_D] * 100\% \quad (1)$$

Where,

W_R = Weight of raw clay sample (g)

W_D = Weight of dried clay sample (g)

2.2. Analysis of the Acidity

A 5g mass of raw clay bulk was dissolved in 50ml of distilled water while stirring well and the P^H value of such clay solution was measured using a digital P^H meter. The temperature of the system was 25.4°C and the P^H value of the distilled water was 7.00 [6-9].

2.3. Analysis of the Properties of Bricks

A few of bricks were manufactured using this clay based on the investigations of water absorbance, bulk density, porosity and mechanical strengths of the bricks of such clays.

A separated raw clay portion was matured while adding water consecutively and placed in similar moulds which had been prepared 10cm*6cm*1.5cm of size each one.

The wetted clay blocks were allowed to solidify about seven days with the absence of direct sun light. After seven days the moulds were removed and the solidified clay bulks were oven dried for six hours under the temperature of 110°C and finally the dried clay blocks were fired using a muffle furnace under the average temperature of 800°C for nine hours [6, 7, 8, 10].

The fired clay blocks were allowed to expose to the normal environmental conditions for a few hours after extinguishing in the muffle furnace about a day.



Fig.2. Prepared bricks

One of prepared brick was oven dried again until getting constant weight and that weight was recorded. The brick was entirely immersed in water for 24 hours and the wetted weight of the brick was measured and recorded. The water absorption of such brick was computed using following equation [3, 7-10].

$$\text{Water absorption} = [(W_w - W_D) / W_D] * 100\% \quad (2)$$

Where,

W_w = Weight of wetted brick (g)

W_D = Weight of dried brick (g)

Bulk density of the brick was computed using following equation.

$$\text{Bulk density} = W_D / V \quad (3)$$

Where,

W_D = Weight of the dry brick (g)

V = Volume of the brick (cm³)

The porosity of the brick was computed as follows.

$$\text{Porosity} = [(W_w - W_D) / \rho * V] \quad (4)$$

Where,

W_w = Weight of wetted brick (g)

W_D = Weight of dried brick (g)

ρ = Density of the absorbed liquid by the brick (gcm⁻³)

V = Volume of the brick (cm³)

Apart from the physic-chemical analysis of raw clays and bricks, the compressive strength and splitting tensile loads of prepared bricks were measured using universal tensile strength testing machine. The relevant compressive strengths and splitting tensile strengths of bricks were determined using following equations [5-9].

$$\text{Compressive Strength} = F_C / A \quad (5)$$

Where,

F_C = Applied compressive load in the failure of the structure (N)

A = Surface area of the bed surface of the brick (m²)

$$\text{Splitting Tensile Strength} = 2 F / (\pi * H * L) \quad (6)$$

Where,

F = Maximum applied load at the failure of the structure (N)

H = Distance between two edges (bridges) (m)

π = Constant (3.142)

L = Splitting length (m)



Fig. 3. Universal tensile strength testing machine

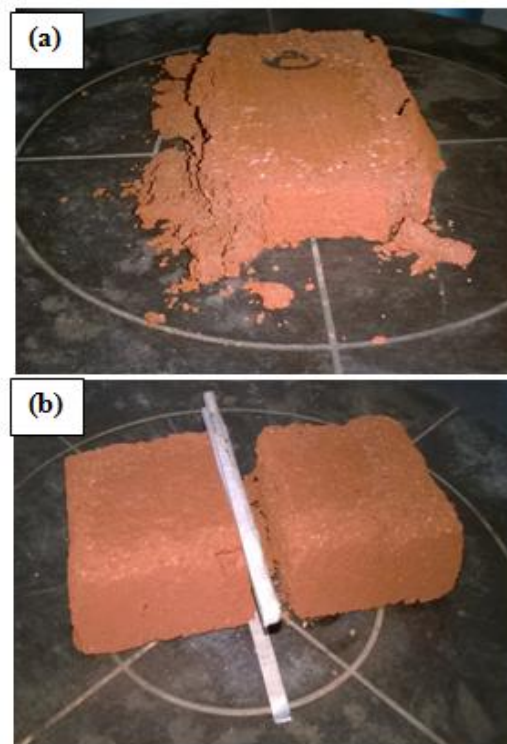


Fig. 4. (a) compressive breaking and (b) splitting tensile breaking of bricks

2.4. Analysis of the chemical composition of brick clay

Three sufficient clay portions were separated from raw clay bulk and those clay portions were oven dried for 24 hours under the temperature of 110°C until removal of the moisture based on following investigations.

- Portion 1- Mineralogical analysis (XRD spectrometer)
- Portion 2- Elemental analysis (XRF spectrometer)
- Portion 3- Analysis of organic and inorganic functional groups (FT-IR spectrometer)

A one of clay portion was crushed using a ball mill for 15 minutes until becoming powdered clay (< 0.037mm).



Fig.5. Ball mill (comminution machine)

The powdered clay sample was oven dried again for a few hours under the temperature of 110°C and the final representative clay sample was selected using the coning and quartering method as shown in Fig. 6. The selected representative clay sample was analyzed using X-ray diffraction (XRD) spectrometer.

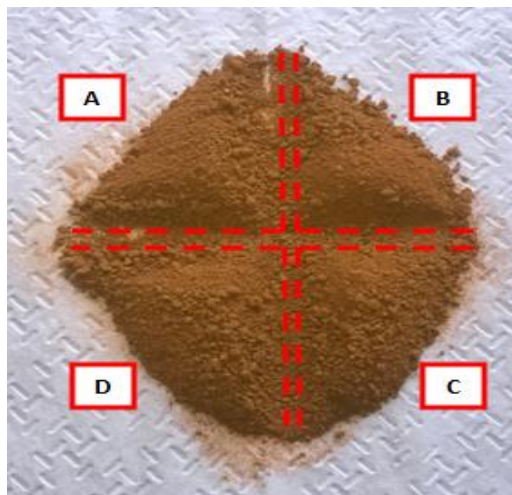


Fig. 6. Coning and quartering method

Coning and quartering method is frequently using method for the selection of final representative sample from large sample for some experiment or analysis. According to the limitations of that method the final representative samples would be a combination of two quarters which are located along a same diagonal (A and C or B and D).

One of other clay portion was crushed using a ceramic crucible while shuffling the portion well and oven dried for a few hours under the temperature of 110°C and the selected final representative clay portion through the coning and quartering method was analyzed using X-ray fluorescence (XRF) spectrometer.

The remaining clay portion was dissolved in a measuring cylinder using distilled water while shaking the system and the shaking period was continued for 15 minutes including a few of stirring activities. The well shacked system was allowed to settle down for 3 hours. The upper clay portion was sucked out using a medical dropper and collected that sucked portion (finest portion). The collected portion was oven dried for 24 hours under the temperature of 110°C. The dried clay portion was crushed using a ceramic crucible and the finest powder was produced. The final representative sample was selected using coning and quartering method and the final representative clay sample was analyzed using Fourier transforms infrared (FT-IR) spectrometer [1-7].



Fig.7. (a) X-ray fluorescence (XRF) spectrometer, (b) X-ray diffraction (XRD) spectrometer and (c) Fourier transforms infrared (FT-IR) spectrometer

3. Results and Discussion

The investigated physic-chemical and mechanical characteristics of raw clays and prepared bricks are given in the Table 1.

Table 1. Physical, chemical and mechanical characteristics of brick clay

Physical, Chemical or Mechanical Character	Magnitude
p ^H	6.57
Natural moisture content (%)	21.45
Water Absorption of Bricks (%)	19.88
Bulk Density of Bricks (gcm ⁻³)	3.15
Porosity of Bricks (%)	62.7
Compressive Strength of Bricks (MPa)	17.25
Splitting Tensile Strength of Bricks (MPa)	0.31

According to such characteristics it seems that the clay is neutral clay (slightly acidic), relatively higher dense due to the presence of highly dense coarse particles such as the sand particles, high porosity, intermediate moisture content and the water absorption capacity of brick was slightly reduced and the bricks are strengthen enough with the mechanical forces such as the compressive force and splitting tensile force. In the further clarification and analysis of such characterizations the followings forecasts could be made regarding the applications of such bricks clay [1,2,6,7,8,9,10,11,12].

- Manufacturing of some hard materials such as abrasive materials because of the higher mechanical strength.
- Advanced water treatment applications because of the higher porosity which is an important character for the process of adsorption due to the increasing of surface area with the porosity.

The X-ray diffraction (XRD) spectrum of brick clay is shown in the Fig.8.

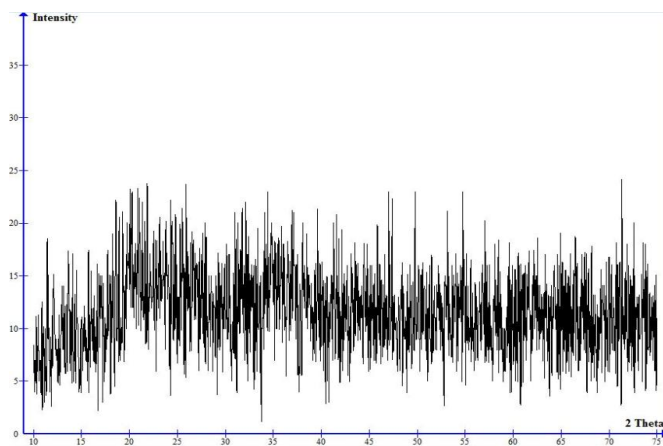


Fig.8. X-ray diffraction (XRD) spectroscopy of brick clay

The observations and the basic assignments of the mineralogy of the brick clay have been summarized in the Table 2 [2, 3, 4, 5, 6 - 9].

Table 2. Description of XRD spectrum of brick clay

Major Intensity Peaks at 2 θ	Indicated Mineral
12, 25, 37	Kaolinite
21, 27, 36, 39, 43, 50	Quartz
20, 35, 55	Muscovite

According to the above description the presence of kaolinite, muscovite and quartz is confirmed as the mineralogy of brick clays. The advanced description on the results of further chemical analysis has been done in following sub sections.

The X-ray fluorescence (XRF) spectrum of brick clay is shown in the Fig.9.

Table 3. Elemental composition of brick clay

Element	Content (%)
Fe	84.38
Ti	5.92
Ca	7.56
Ba	2.14

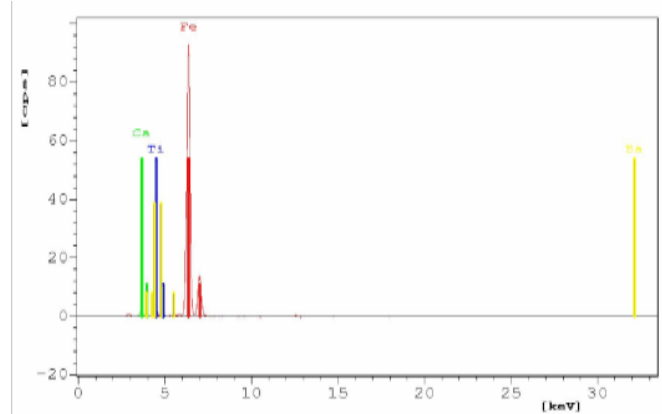


Fig.9. X-ray fluorescence (XRF) spectroscopy of brick clay

The elemental composition of brick clay is shown in the Table 3.

According to the above results the Fe was found as the most abundant element in brick clay with Ti, Ca and Ba as trace elements. Usually those elements are found in clays as their oxides because of the reaction between such metallic elements and water or moisture. The presence of CaO indicates the higher ion exchanging capacity and heat resistant capacity of such clay. In addition that it is possible to expect some content of Fe mineral in brick clays [5, 6, 7, 8-12].

The Fourier transforms infrared (FT-IR) spectrums of bricks clay are shown in the Fig.10 and Fig.11.

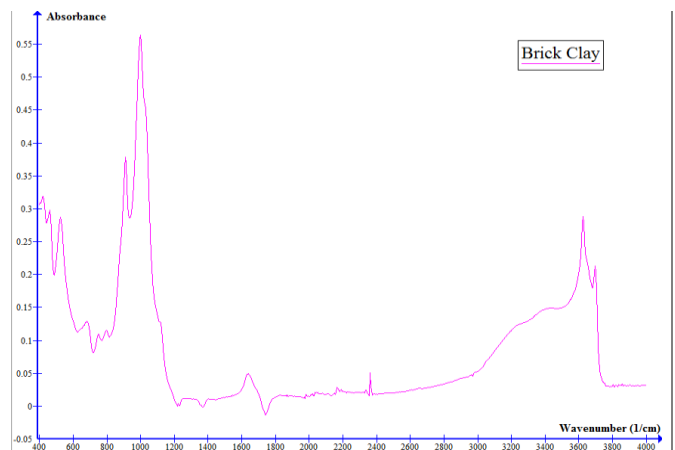


Fig.10. Fourier transforms infrared (FT-IR) spectroscopy of brick clay (absorbance)

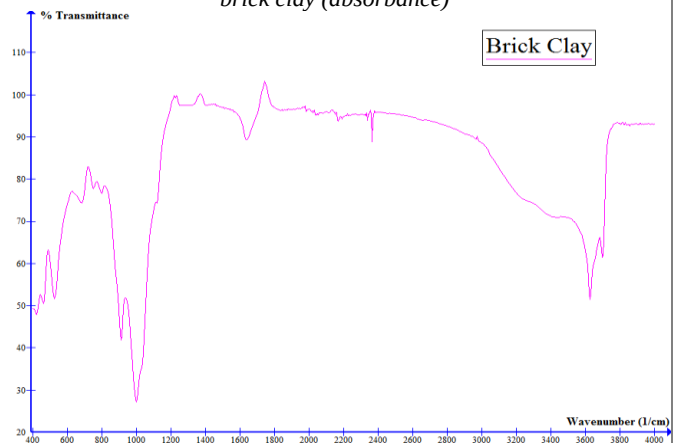


Fig.11. Fourier transforms infrared (FT-IR) spectroscopy of brick clay (transmittance)

The above FT-IR spectrums showed the presence of major peaks of wave numbers at 3702 cm^{-1} , 3629 cm^{-1} , 1001 cm^{-1} , 909 cm^{-1} , 530 cm^{-1} , 469 cm^{-1} and 420 cm^{-1} that denote the presence in order of OH stretching of inner surface hydroxyl groups, OH stretching of structural hydroxyl groups, Si-O stretching, OH deformations of inner hydroxyl groups, Al-O-Si deformation and Si-O-Si deformation. Those inorganic functional groups confirmed that the presence of kaolinite, muscovite and quartz in brick clays [3, 5-12].

In the further clarifications regarding the kaolinite, it seems that the brick clay may have higher adsorption for metals such as Co, Cu, Fe, Ni, Mn, As, Hg with some other toxic metals and some of pathogens from some solvents. Therefore, it is possible to suggest the development of this brick clay or some more clay types which are having those components for the water treatment applications and some other advanced metal extraction processes.

4. Conclusion and Remarks

This brick clay type is much dense while having higher porosity and intermediate water absorption. According to the chemical composition, there were found some higher Fe content with Ba, Ca and Ti with kaolinite, muscovite and quartz as the mineralogy. This clay type should be an appropriate material for the manufacturing of hard composites, refractory materials, ion exchanging material and adsorber for a series of advanced industrial applications.

5. Acknowledgement

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