Importance of drill string assembly swivel in horizontal drilling

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Význam rotačného spojníka pri horizontálnom vŕtaní

A part of the drill string – the swivel (rotational connector) – accomplishes an important task in the horizontal drilling. Its malfunctioning makes it impossible to draw in (install) large diameter and length pipelines. The causes of the connector break-down during the horizontal drilling are investigated in the paper. The drilling has been made for twenty inches gas pipeline installation during reaming operations. A trouble was encountered making good work conditions of a system consisting of the drilling machine drill string reamer swivel tube shield of Cardan joint and the gas pipeline 500 m long. In this case, the swivel brokes down and the planned operation was not finished. The assessment of improper drilling conditions, selection of operation system components, and drilling parameters and the insufficient technological supervising have created an excessive risk of failure. A proper application of technical analysis would considerably decrease the hazard of failure which cause large costs, delays and decrease of confidence to the drilling contractor and pipeline installation.

Key words*: swivel, drill string, horizontal drilling, gas pipeline*

Work conditions of drilling assembly

After making the 559 m long horizontal borehole under a water canal in the Zachodniopomorski District, a pipeline section of 508 mm in diameter was dragged. The horizontal directional drilling HDD was made mostly in soft and very soft layers (muddy sands, fine and medium-grain sands with gravel interbeddings). These rocks can be easily drilled but at the same time they create problems with the loss of stability of the borehole walls (landslides). Prior to making a pilot borehole, rock strata were probed in four places along the pipeline to a depth of 11 to 18 m. The registered dogleg changes indicated high values of the angle, considerably exceeding values planned for this trajectory of drilling. In the zones of rapid changes of curvature, drilling problems were observed to occur. In such situations the whole mass of the string rests upon the well's walls, in the course of which key seats form and sticking may result. An excessive friction occurs between the well's wall and the string, making the torque value increase. Therefore it is crucial to decide in what HDD conditions and what external loads of the strains in the drilling string material may reach the critical level, i.e. higher than admissible. This recognition is important for a rational and safe exploitation of the drilling string (Bednarz, 2004). The tensile strength of the suspended string at the outlet of the well during drilling operations is lower than indicated by the measured depth (length) (MD). This is caused by leaning of the string against the well's wall, especially at a high angle of deviation. During tripping out operations the force is higher than in vertical wells. Moreover, during the HDD drilling the torque is also higher than that for vertical wells. In the vertical wells, the maximal tensile strength is a limit load, whereas in horizontal wells it is a torsional strength. Unlike simple expected loads in the vertical wells which require only simple calculations, in the horizontal wells the influence of friction is big enough, therefore it has to be accounted for and simulated. The drilling string design for a HDD should be made on the basis of a number of factors. Owing to the increasing length of the HDD, it is also the behaviour of the drilling string in the horizontal wells. The drilling string is limited by the well's walls and accordingly assumes a few shapes along its length, i.e. straight, sinusoidal, unstable sinusoidal and spiral. The shape of the drilling string in a HDD exhibits a very dynamic character of stresses. During the first drilling operations microcracks form in places called in mechanics "stress concentrators". Their origin may be related the technological manufacturing process of pipes (nonmetallic intrusions, defects of microstructure, bottom of thread groves, etc.), and the exploitation in a broad sense (cuts, notches and pits accumulated during handling and drilling). Most failures of the drill pipe are a result of the material fatigue. The amount of fatigue damage which results in the exploitation depends upon the tensile and torsion load in the pipe at the dogleg, the severity of dogleg and corrosion. At the place of a failure the dogleg was observed to change from 0.25^o to $1.59^o/10$ m at the planned value of $0^o/10$ m (Bednarz et al. 2002).

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The first part of the drilling was made with a bit 200 mm in diameter (pilot), to be later reamed three times to the diameter of 450 mm, 650 mm and 900 mm, respectively. Owing to the uncoordinated operation, the idle time was about two weeks. At that time, no casing was tripped to the well 900 mm in diameter and about 500 m long. Some rock slide was observed, a reamer 900 mm in diameter was used and a pipe 508 mm in diameter started to be tripped inside (Fig. 1). At a distance of 40 m and 60 m from the target side the pipe got stuck in the process of tripping. After doing away with this problem the pipe was finally tripped inside (according to the reports, the tractive force was 18 tons, and the torque 10 000 Nm). At a distance of 60 to 445 m the tripping operation went on smoothly, with a continuous increase of the tractive force to 35 tons and the constant torque 10 000 Nm. At a distance of 445 m the pipe was stuck. First, the advancement decreased to 0.4 m/h to finally stop. At that time the tractive force was ca. 35 tons and the torque 25 000 Nm. At the end of the dragging operation at a distance of 445 m to the end the pipe got stuck. Further attempts to move and trip the pipe were futile. At a tractive force of about 100 tons the rotary swivel was damaged.

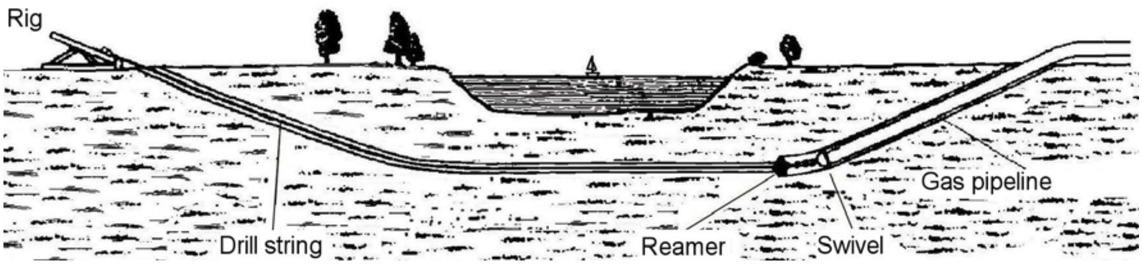

Fig. 1. Horizontal directional drilling.

It should be stressed that, according to the technical characteristics, the maximal tractive force of the rig is 750 kN (76.5 tons). The pipe was withdrawn without problems.

The swivel

The location of a swivel in the protection of conductor head of a gas pipeline with the visible Cardan joint pin is presented in Fig. 2. A crack of the shaft is visible in the front part of the swivel. The leading head with the partly cut off casing and the intermediate connector, presented in Fig. 3, is ribbed on the gas pipeline side.

Fig. 2. A swivel in the protective pipe of leading head of the gas pipeline

Fig. 3. Leading head and intermediate connector.

A considerable frictional wear of the inner casing pipe head was observed at about 1/3 of circumference in the front part, overlapping with the reamer inside casing along a section ca. 30 cm long. Moreover, the casing pipe was deformed in the sections of increased curvature after the reamer-head set was installed. An oval was formed of 480 mm/510 mm. The cut swivel with destroyed bearing elements, severed shaft, sealings and grease are presented in Fig. 4. The 92 mm diameter main shaft in the back part had bearing in the immobile roller body of external diameter with an angular bearing (double purpose bearing). The back nut of the bearing of external diameter was fixed in a position with a plate screwed to a flat back surface of the nut touching the relevant protrusion on the shaft.

The second angular bearing was in the front of the main bearing, separated with a distance protrusion of decreased inner diameter in the body (Fig. 5). A roller bearing was disposed before the oblique bearing, the outer race of which was sealed with two O-rings. The bearing was fixed from the front with an inner ring screwed to the body, sealed with a similar O-ring and blocked by an imbus.

Fig. 4. Cut swivel with destroyed bearing elements, severed shaft, sealings and grease.

Fig. 5. Internal elements of the damaged swivel.

The shaft was sealed in an externally threaded ring M180x2 with the use of a string seal, pressed with a flat ring and 6 screws M8 disposed in a flat surface on its circumference. The screws were blocked by means of bolts fastened perpendicular from the outer roller surface of the ring. By making a longitudinal cut of the swivel it was possible to measure the geometrical dimensions of the swivel elements. The size of the specific components was defined with the use of special measuring apparatus. Some of the parameters are quoted in the chapter 3. It should be stressed that the high degree of damaging of the bearings and the absence of a considerable part of the shaft hindered detailed measurements of some elements of the bearing. This limits the possibility to clearly state their arrangement in the oblique bearing disposed between the screwed ring and the angular bearing in the front part of the swivel.

This zone is of less significance in view of the transmitted longitudinal loads. The main and some detailed dimensions of elements of the swivel are given in Fig. 6.

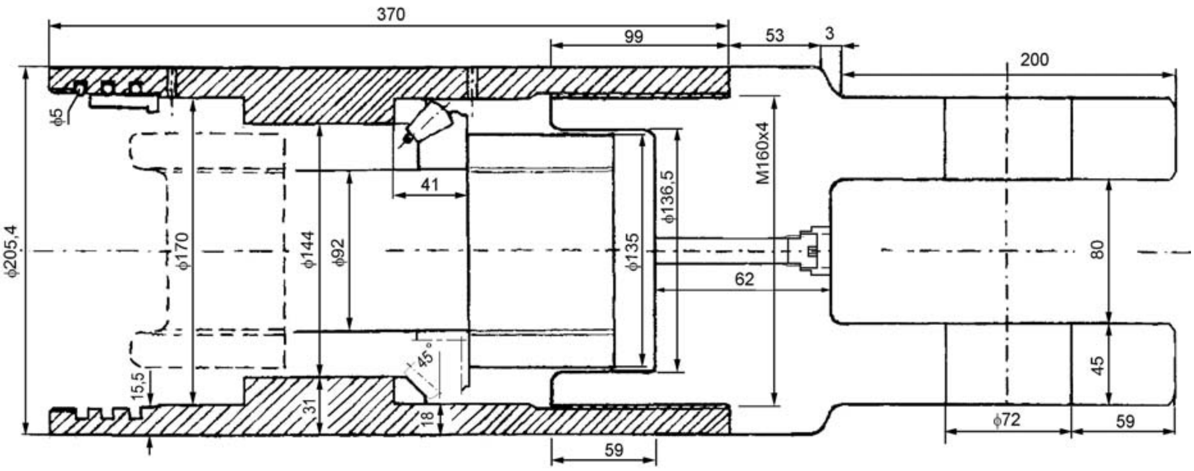

Fig. 6. Design of a swivel – based on the measurements of damaged elements of the swivel

The location of the shaft at the moment of its final failure was denoted with a dashed line. The angular roller bearing <45^{o} (22 barrels) disposed in the back part of the body of the swivel of 170 mm of external diameter and ~92 mm of inner diameter was supposed to transmit the whole longitudinal force during the gas pipeline tripping operation. The round nut of 135 mm in diameter was touching the inner race of the bearing with its flat surface.

Moreover, in the course of analyses of the whole set, it was also the pin joint at the end of the swivel which was measured. The openings in the swivel's lug turned out to have assumed an oval shape;

- lug no. 1 – 71.56 mm/ 72.48 mm,
- lug no. 2 – 71.32 mm/ 72.50 mm.

These results show that the swivel was worn to a medium degree.

Wear observation and metallographic analyses

Cutting the tractive swivel revealed a considerable damaging of the swivel's elements, particularly the bearings. The elements of the bearing got stuck most probably because of the huge load in the angular bearing system. This, in turn, was caused by blocking of one of its elements because of excessive abrasive wear or cracking. The bearing blockage was accompanied by a drastically increased friction, cracking of the inner race of the bearing (Fig. 7).

Fig. 7. Elements of bearings fixed at the end of the shaft.

This also resulted in a damaged bearing cage, cutting of the barrels and finally dragging of these elements through a contraction in the body of the swivel and disposing them against the elements of the auxiliary bearing.

Owing to the increasing friction and emission of heat, the contacting surfaces started to blur, the steel became softer, then it melted in the zones of highest loads. The strength of the material of the tractive shaft lowered in the vicinity of the melting zone. Cracked inner races of both bearings, plastic deformations of barrel, many of which are stuck in the melted bearing material, are presented in Fig. 7. Owing to the considerable lowered strength, the shaft was twisted and severed. The color and consistency (density and viscosity) of the semi-solid lubricant from the back part of the swivel had a regular form, although was strongly contaminated with friction products, abrasive wear of the bearing elements and to a small degree with sand. The quantity of grease in the back part of the swivel (as compared to the front bearing) was 0.55 l making up ca. 80 % of free space in this part of the swivel. A plate was cut off the melted end of the shaft and bearing (Fig. 7) for metallographic specimen analyses. A polished surface of the sample plate with visible zones of thermal impact is presented in Fig. 8. This material underwent a high temperature in the course of friction processes in the stuck bearing. The metallographic analyses were performed on a section of the bearing and the shaft which got melted and welded during the break down. From this section, samples (No. 1, 2, 3, 4, 5) were collected for microscopic analyses. Lines, along which the hardness tests were carried out with a Vickers hardness tester at 300 N are also marked in the figure.

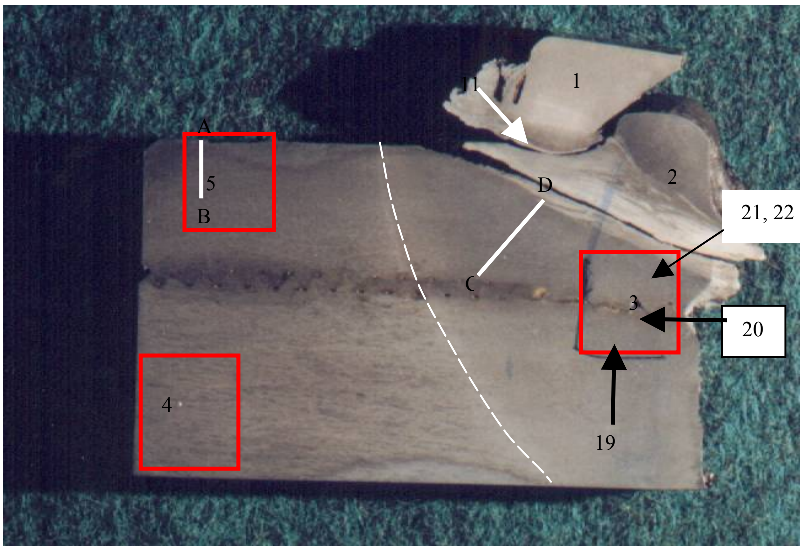

Fig. 8. Section of the shaft, barrel, bearing race and the fastening nut from the damaged area with indicated places where the samples were collected for the microscopic analyses (1, 2, 3, 4, 5). A-B and C-D – lines along which hardness tests were made.

Moreover, the boundary of influence of heat over 700 ^{o}C was marked with a dashed line. The race was made of bearing steel having a martensitic structure with divorced cementite. In the zone of welding the materials mix up (Fig. 9). Except for the zone of heat impact, the barrels have a martensite structure with a cementite separation, typical of bearing steel. Judging from the structural character, the nut is made of a similar material as the shaft. The temperature of nut reached over 750^{o}C in the zone of heating. This can be proved by the presence of a structure of martensite bainite and retained austenite (Przybyłowicz 1999). The material of the shaft beyond the heat impact has a bainite-ferritic structure of hardness of about 250 HV. In the heat affected zone (sample 3) the hardness is slightly higher 260–270 HV. The measurements of hardness in the heat affected zone up of the nut from point C to D revealed an increase of hardness 240 to 312 HV, which is justified by the formation of martensitic structures. The nut underwent strong friction forces against the cooperating elements (Fig. 10), resulting in visible deformations of the surface.

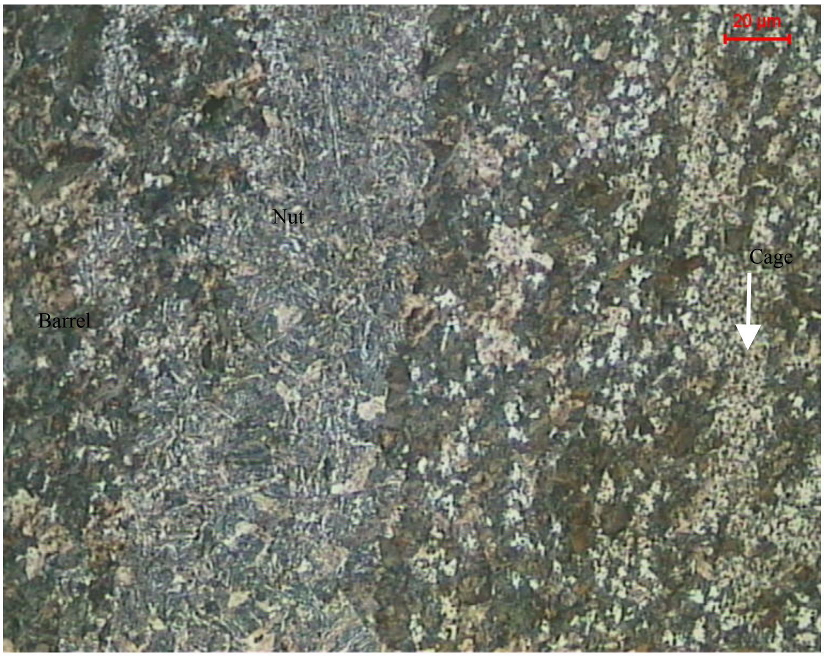

Fig. 9. Structure of the zone of welding of bearing cage, nut and barrel. Visibly mixed materials. Sample no. 2.

Fig. 10. Strongly deformed surface of the nut. Sample no. 5.

Concluding remarks

It can be stated from the results of measurements and analyses that the swivel in the technological set used for tripping in the gas pipeline was subjected to a considerable longitudinal force, and from the moment the bearings were stuck, a great torsional torque. During the undisturbed process of tripping, the swivel under the high tensile loads was subjected to the torsional torque, being a result of friction in the bearings. A small variability of torsional stresses resulted only from a periodic presence of bending moment caused by the curvature of the hole. These moments did not produce significant torsional stresses in the shaft. The swivel was attached to the tractive joint which was welded to the conductor head of the front pipe of the gas pipeline with a deflected intermediate joint. These two pin joints made an angular coupling with an asynchronic character of rotations of the „driven shaft" to the "driving shaft". The swivel was a driving element in this "coupling", but only in the sense that the torque absorbed by friction in the bearing can produce some torsional stress. „The driven shaft", i.e. the tractive swivel, basically remains immobile, owing to the increasing fixing moment as the gas pipeline elements are introduced to the hole. The longitudinal force was basically transmitted by one angular bearing of angle of 45^o. For such high loads exerted by the longitudinal force (after technical characteristic of the rig the maximal tractive force was 750 kN (Tab. 1) it would be more justifiable to use a bearing of an angle less than $45\ ^o$.

Tab. 1. Technical characteristics of the HDD rig .

Parameter	Dimension
General size (length x height)	9,3 m x 2,78 m
Maximal tractive force	750 kN
Maximal pushing force	400 kN
Length of the drill pipe	6,0 m
Torque of head / rotary speed	
gear 1	50 kNm / 40 r/min
gear 2	25 kNm / 80 r/min
bore of head spindle	ID = 0,07 m
thread connection	5 ½'' IF
Make-up torque of thread connections	60 kNm
Screw-out torque of thread connections	75 kNm
Control: joystick	electro-hydraulic
air conditioning	4 kW
measuring devices	analog-digital
Undercarriage: caterpillar	
width	1,045 m
inclination angle	$9 - 22^0$

A very high temperature in the zone of the damaged bearings was confirmed by analyses of metallographic specimen of the material sampled from the swivel.

Conclusions

1. The swivel was analyzed on the basis of geometric measurements of its elements, after cutting the swivel and disassembling owing to the lack of technical-operational documentation,.
2. The material and metallographic analyses were made for the bearing material, which underwent high temperature, especially in the vicinity of the race of the inner main angular bearing.
3. The analysis reveals that the failure was most probably caused by a break-down of the bearing. The stuck bearings and the rotating shaft caused heating of the cooperating elements (nut, bearing cage, ring and barrels) and their welding. The temperature in the area of the rubbing elements was high enough (over 1000 ^oC) to cause welding of various materials and a flash typical of friction welding. High temperature caused heating of the shaft in the ring area to over 700^oC and its tearing off. At such temperatures the forces responsible for tearing off are considerably low, owing to the low yield point value.
4. As far as longitudinal forces were attainable by a HDD rig, the design of the swivel bearing node does not fully correspond to the load conditions of the drill string.
5. According to the technical characteristic, the maximal longitudinal tractive force of the rig is 750 kN (ca. 75 tons) exceeds the dynamic capacity of the used bearing.

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