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GEARING WITH VARIABLE GEAR RATIO APPLIED IN MECHANICAL SYSTEMS

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Keywords: gear design, variable gear ratio, FEM, stress

Abstract: The gearing with changing transmission gear ratio are used as synchronization component and for specific parameters. The gearing with changing transmission gear ratio is used in the practice, even though the "standard" gearing with constant transmission gear ratio are used more often. This article describes how to optimize the design of pitch curves of non-circular gear for given parameters. The non-circular gearing is consisting of two identical gear wheels. For a non-standard gearing was applied eccentric elliptical gear drive with continuously changing transmission gear ratio. The kinematic properties of this gearing are different from the properties of standard circular gears – spur gear. Thus, the gear ratio changes over the time of one revolution. The article is devoted to problems determining of the stress in a dangerous section of tooth foot using FEM.

1 Introduction

Gearboxes are one of the most used transmission mechanisms. They are based element in which machines transmit and transform mechanical energy and motion.

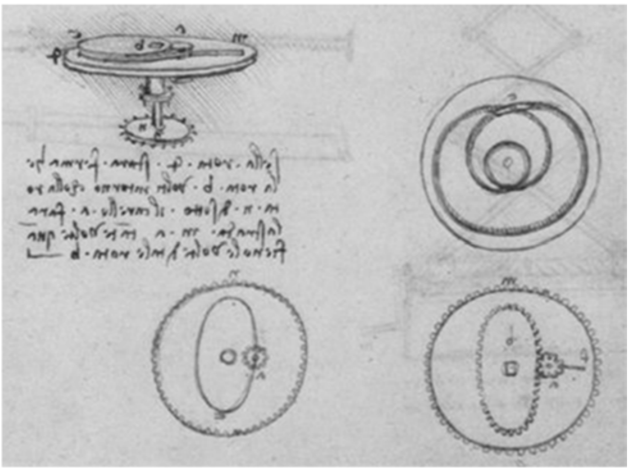

Figure 1 Sketched by Leonardo da Vinci [1]

The history of gears is probably as old as civilization itself. The earliest description of gears was written in the 4th century B.C. by Aristotle. He wrote that the "direction of rotation is reversed when one gear wheel drives another gear wheel". In practice, the most commonly used "standard" toothed gears, which can be characterized by a constant gear number and circular wheel shape. Non-circular gears are not very known, even though the idea of non-circular gears originates from the precursors of the engineering thought. These gears were sketched by Leonardo da Vinci (Figure 1). In late XIX, century Franz Reuleaux ordered at Gustav Voigt Mechanische Werkstatt in Berlin a series of non-circular gear models to help study kinematics. The gears made at those times had simplified tooth shapes and, for this reason, the meshing conditions were not always correct.

Noncircular gears are presented as a curiosity for the gear industry history, due to their complex design and manufacturing difficulties. Nowadays, performant modelling and simulation software, advanced CNC machine tools and nonconventional manufacturing technologies enable noncircular gear design and manufacture.

As mechanisms used to generate variable motion laws, in comparison with cams, linkages, variable transmission belts, Geneva mechanisms and even electrical servomotors, noncircular gears are remarkable due to their advantages, such as the ability to produce variable speed movements in a simple, compact and reliable way, the lack of gross separation or decoupling between elements, fewer parts in the design phase, the ability to produce high strength-to-weight ratios, etc [2,3].

The applications of non-circular gears include for example textile industry machines, for improving machine kinematics resulting in the process optimization [4-6], window shade panel drives, for introducing vibration which interfere with natural oscillations and cancel them out [7,8], high torque hydraulic engines for bulkhead drives

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[9,10], mechanical presses, for optimization of work cycle kinematics. They are also using as high-power starters, mechanical systems providing progressive torque for easier starting of the machines, where progressive torque helps to overcome the start-up inertia and as forging machines, for optimizing the work cycle parameters (reducing pressure dwell time). The use of noncircular gears in industry certifies their performances, leading to new ideas for improved working conditions. Non-circular gears have their application as well oval gear flowmeter [11]. Oval gear flowmeters are categorised as positive displacement flow technology. The positive displacement flow technology allows for precise flow measurement of most clean liquids regardless of the media conductivity.

A common challenge in the design of mechanical systems is the kinematic synthesis of a mechanism in order to satisfy a set of motion characteristics [12]. Frequent requirements are to guide a rigid body through a series of specified positions and orientations (rigid body guidance), to force a coupler point to move along a prescribed trajectory (path generation), or to cause an output member to move according to a specific function of the input motion (function generation) [13,14].

The first step in the noncircular gears virtual design process is the generation of the conjugate pitch curves, starting from a predesigned law of motion for the driven element or a predesigned geometry for the driving gear pitch curve.

By designing a pair of non-circular gears, which are able to perform a proper gear ratio function, the output member of a mechanism can be effectively forced to move according to a prescribed law of motion, when operated at a constant input-velocity. This mechanism is designed to obtain a specific motion law. Detailed knowledge of meshing conditions is a prerequisite for studying kinematic conditions in gearings, as well as the strength calculation of gearing.

2 Non-circular gearing

Generation of this non-circular gear was by developed starting from the hypothesis such as the law of driven gear motion, variation of gear transmission ratio and design of driving gear pitch curve. This model of non-circular gear was by designed for variable transmission ratio in the range u = 0.25 to 4.0. This transfer should be formed by two identical wheels with the number of teeth $z_1 = z_2 = 40$ and gearing module m_n = 4mm, the distance a = 160mm and for a one direction of rotation.

The first step in the noncircular gears design process is the generation of the pitch curves, starting from a predesigned law of motion for the driven element or a predesigned geometry for the driving gear pitch curve. For a non-standard gearing was applied eccentric elliptical gear drive with continuously changing transmission gear ratio. That is, the ellipse was used as the pitch curve (Figure 2). For the given distance, the pitch ellipse has a large half-axis a_e=80mm, which is half of the axial distance. The position of ellipse focus is determined by considering the desired continuously changing transmission gear ratio. For the given variable transmission ratio in the range u = 0.25 to 4.0 is position of the ellipse focal point (the centre point O of rotation) determined by the ratio lengths $x_1 : x_2$ equal to 1 : 4. The second half-axis be = 64mm is determined by the distance from the focus point a_e = 80mm for the transmission ratio u = 1.

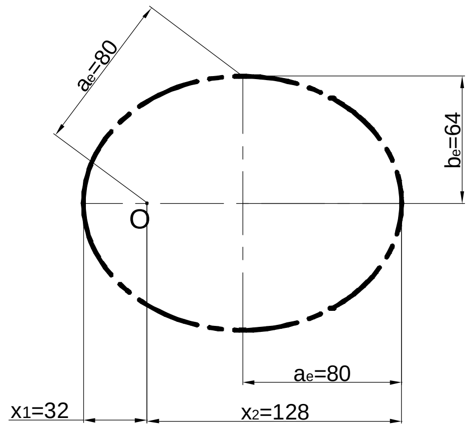

Figure 2 Pitch ellipse for gear ratio u = 0.25 to 4

In this case, one of the conditions of a correct mesh is that the measurements of the pitch on the ellipse pitch must be kept constant. A geometric separation of the pitch ellipse into 40 identical sections (the number of teeth $z_1 = z_2 = 40$) is mathematically much more difficult than in the case with the standard gear pitch circles.

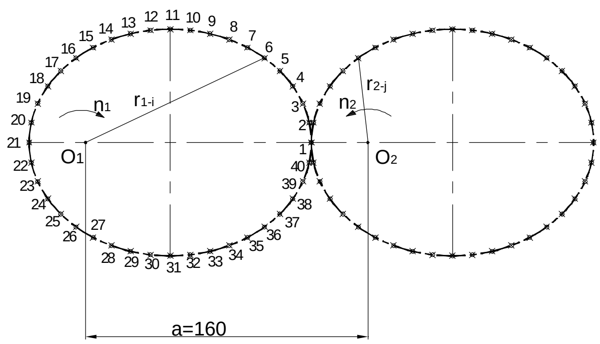

Figure 3 Designed of non-circular pitch curves

The Figure 3 shows the pitch ellipses of designed eccentric elliptical gear drive with continuously changing transmission gear ratio for a given parameters. Torque transmission ensures shape bonded between meshing gears. The gearing consists of two identical gears. The toothed number is shown for the drive wheel, for the driven wheel this numbering is the same. Wheels are designed for only one direction of rotation. The pitch ellipses must meet the condition that for each tooth the sum of the radii is equal to the axial distance:

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$$r_{1-i} + r_{2-j} = a = 90\ mm \qquad (1)$$

where r_{1-i} and r_{2-j} are a radius of mesh points.

3 Kinematic properties

In pursuit of kinematic ratios on the proposed gearings we assume from the right mesh conditions. Kinematic conditions were processed for a gear 1 (the centre of rotation at point O_1) and the gear 2 (with the centre of rotation at point O_2). The two gears are shown in a kinematic dependence one graph (on the horizontal axis of the wheel teeth first).

In Figure 4 is a course of continuously changing gear ratio in one mesh generated by elliptical gear, which continuously varies in the range from u = 0.25 through u = 1.0 until u = 4.0 and back. Thus, the gear ratio changes over the time of one revolution. A gear ratio value that is less than 1.0 signifies that this is an overdrive, and a gear ratio value greater than 1.0 signifies a speed reduction.

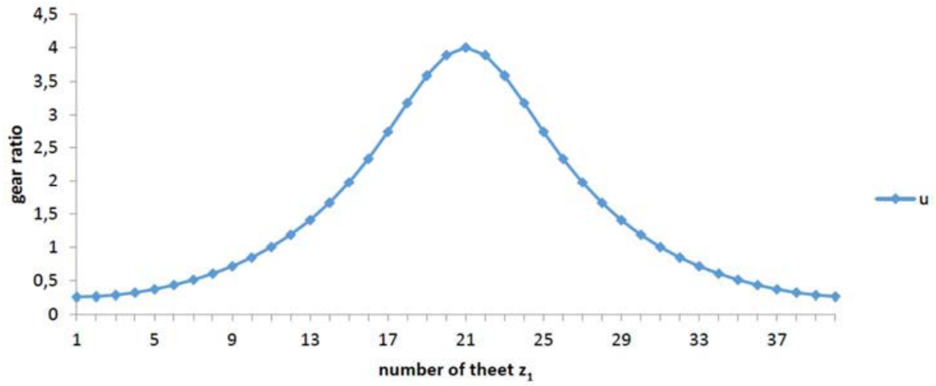

Figure 4 Gear ratio

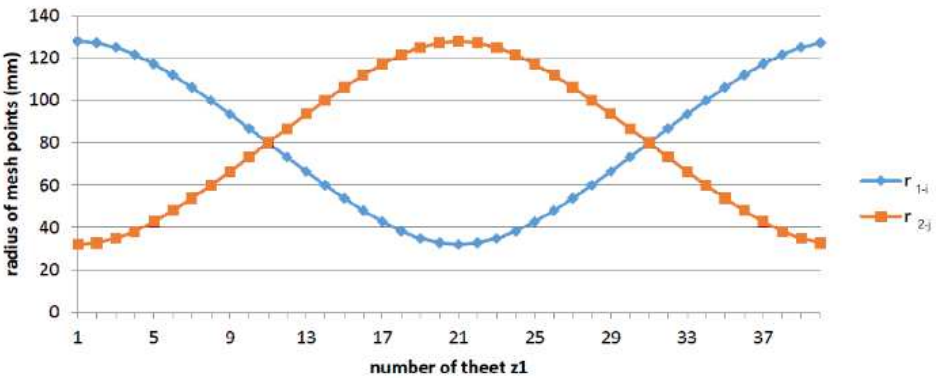

Figure 5 Radius of mesh points

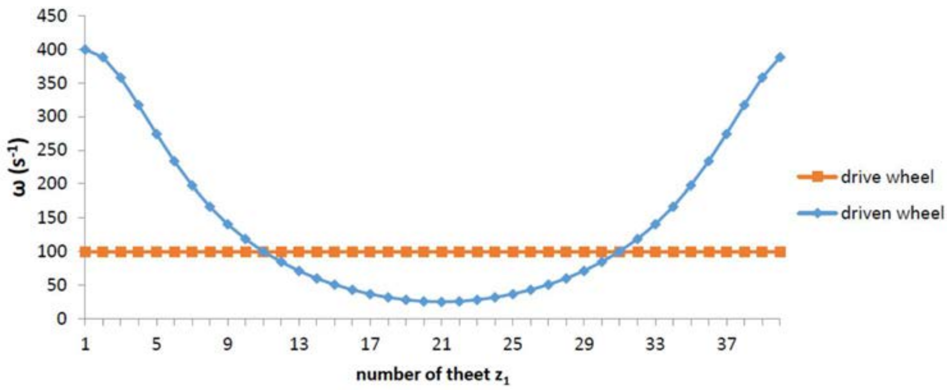

Figure 6 Rotational speed in non-circular gearing

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Figure 5 shows the progress of the meshing radii at the individual points of contact, designated as r_{1-i}, respectively. r_{2-i}, where index 1 applies to the drive wheel, index 2 for the driven wheel, index i or index j corresponds to the order number of the tooth.

The rotational speed on the drive wheel gear and the driven wheel gear is constant to standard spur gears. For designed elliptical gearing with variable transmission, the angular velocity of the driven wheel is not constant but is changed according to the continuous changing of the gear ratio. This is shown in Figure 6, if the angular velocity is on the drive wheel (ω_1 = 100 $^{s-1}$) and the driven elliptical wheel (ω_{2i}).

4 Stress of teeth solution by FEM

Create a geometric model of the gear is the first step to deal with tooth stress by FEM. Universal instructions to create geometry computer model does not exist [15]. The first part was to develop a functional model gear designated for the production of gears gearing for NC machine to electrospark cutting. To determine the computer model for studying deformation of the teeth using FEM was necessary to determine the material constants, define the type of finite element, and selecting appropriate boundary conditions.

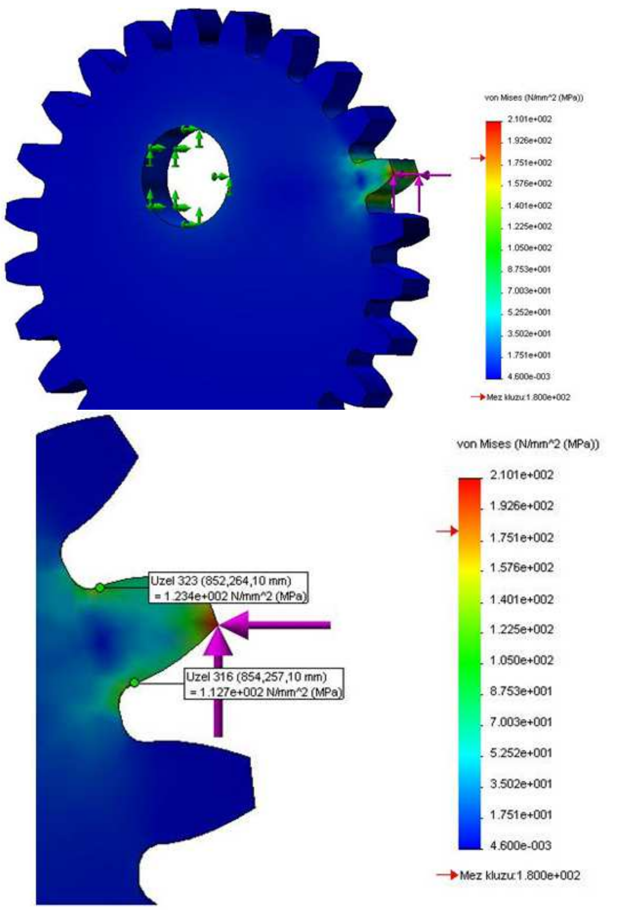

Figure 7 Sample solutions to stress in gear by FEM

The problem is solved with the gear continuously variable transmission numbers. The stress in a dangerous section of the tooth is solved using the finite element method for driving gear, the gear teeth to reach the number 0.25, 1 and 4.0.

In Figure 7 are results solutions to stress in gear by finite element method for tooth with gear ratio u = 1. Width of teeth is 10 mm, the driving torque is M_{k1}=100Nm.

Because of the gear of asymmetric profile, where the teeth of one gear wheel has different shape, the stress of teeth are different.

5 Conclusions

Non-circular gears are presented as a curiosity for the gear industry history, due to their complex design and manufacturing difficulties. Nowadays, performant modelling and simulation software, advanced CNC machine tools and nonconventional manufacturing technologies enable noncircular gear design and manufacture to be more feasible.

The main objective of this paper was to defined base kinematic properties of non-circular gear. Gearing was designed to meet continuous change of gear ratio during one rotation. The gearing consists of two identical gears and the basic shape of the gear wheel is formed by an ellipse. Wheels are designed for only one direction of rotation and the centre of rotation is one of the foci of ellipse. It is the gearing with variable transmission. Properties of this gearing are different from the properties of standard circular gears – spur gear. Thus, the gear ratio changes over the time of one revolution. A gear ratio value that is less than 1.0 signifies that this is an overdrive, and a gear ratio value greater than 1.0 signifies a speed reduction. For designed elliptical gearing with variable transmission, the angular velocity of the driven wheel is not constant but is changed according to the continuous changing of the gear ratio. The stress in a dangerous section of tooth calculation by standard is provided according to specific conditions. This calculation is not suitable for elliptical spur gear with variable gear ratio. The theoretical determination of the stress in the teeth is difficult for complex-shaped teeth. One way to determine the stress in a dangerous section of the tooth is a solution to this problem using the finite element method.

This elliptical gearing was by used in the drive mechanism for a new press for example. The new press kinematics result in a reduced pressure dwell time in comparison with a conventional press kinematic.

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THIN CLIENT IN MASSIVE RLS WITH CLOUD APPLICATION

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Keywords: remote laboratories, cloud computing, REMLABNET, thin client, datacentre
Abstract: T Many organizations, both large and small, are investigating the potential of thin client architectures for their companies. In general, a thin client is the one which does not have any local storage and we are using this because of their many advantages. Few years ago, we build our own virtualized cloud for REMLABNET and we still are taking benefits of this decision. This item handles with using Cloud computing platform for providing Remote laboratories. This work shows, how it is possible to save money if we use centralized system for more consumers. Every consumer can use access to centralized portal in the Cloud computing from Consortium REMLABNET. Every item is focused on environments of universities, where this cloud is existing, and this is what we want to use for remote labs. This is item from practice knowledge and experiences about system function and managing virtual platform and next construction this proposal.

1 Introduction

IT departments in universities are permanently under pressure to provide high quality services with reduced budget. On the other side, costs of energy for datacentres (DTCs) running and cooling call for radical changes in all universities compared to classic DTCs. Few years ago, we were using a prevailing standard in the decentralization and fractionating of services to several physical devices. This approach is nowadays under severe changes in direction to consolidation of DTCs denoted under cumulative term of virtualization. Virtualization has to offer decrease in energy consumption and increase in system performance without compromise on security of DTCs [1].

The last two decades has seen the rise of the DTC computing practically in every application domain. The move to DTC has been powered by two separate trends. In parallel, functionality and data usually associated with personal computing have moved into the DTC; users continuously interact with remote sites while using local computers, while running intrinsically online applications, such as email, chat or manipulating data, traditionally these are stored locally, such as documents, spreadsheets, videos and photos. In effect, modern architecture is converging towards cloud computing (CC), a paradigm where the whole user activity is funnelled into the large DTC via high-speed networks. Simply speaking, cloud computing is a set of computers, services or infrastructure. Delivering services are meant to reduce the work of consumers every day, as well as service providers and IT specialists. Cloud computing allows more access services as it reduces infrastructure delivery time from weeks to hours and it offers reimbursement for provided sources and services only [2].

Main idea of our work and this paper, is for clients to use new methods on how to provide remote laboratories (RLs) [3]. On the Figure 1, we can see primary idea of this system. On the left side, we can see individual remote laboratories, experiments, with physical HW and SW, connected to our virtualized cloud. Core of our cloud is management system for monitoring, diagnosing and administrating remote laboratories and users, or our clients. This management system is named Remote Laboratory Management System (RLMS) REMLABNET and it is consisted of few modules. For example: diagnostic server, data warehouse and content management system (with schedule and calendar, communication server, etc.) [4].

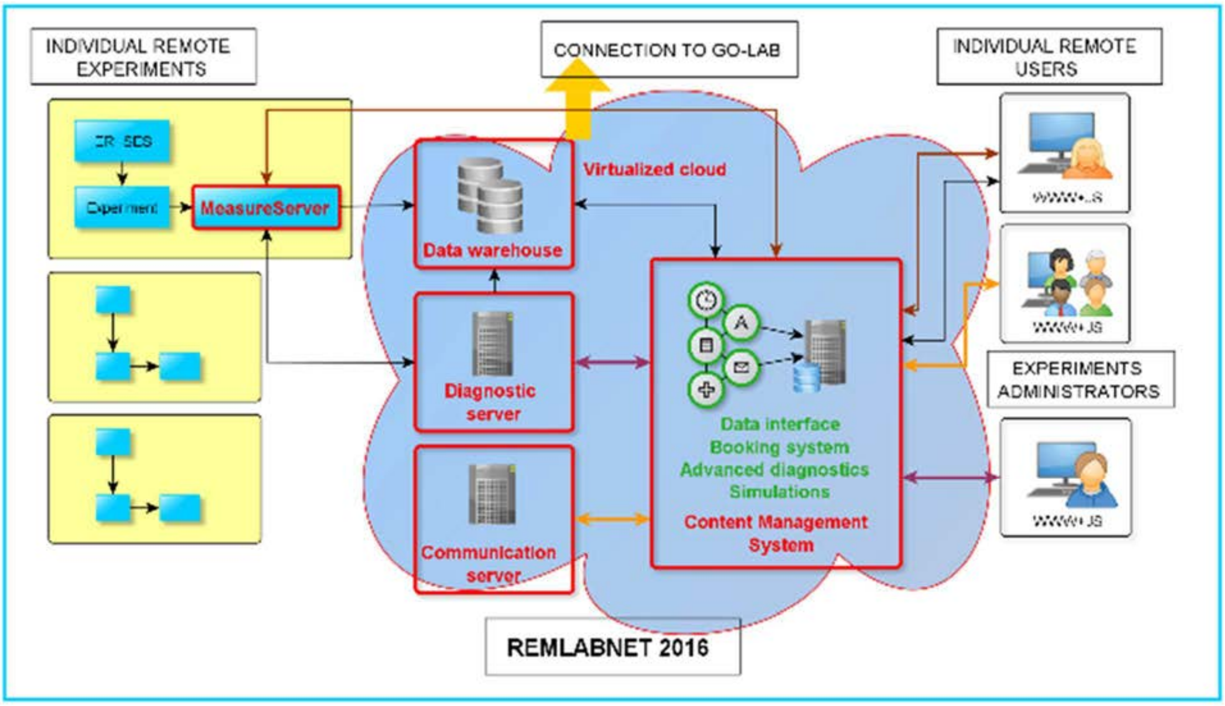

Figure 1 – Block diagram of CC-REMLABNET

2 Cloud computing concept

Of course, our work is primary oriented for remote laboratories, but our idea is to provide remote laboratories like cloud computing service. We were first in the world, who provided remote laboratories via CC technology. A new concept of our CC is figured on the Figure 2, where we can see all interesting parts of this idea.

First, we can see main parts of cloud computing. Each cloud is based on three primary services for use [5]:

IaaS – Infrastructure as a service is a standard service for providing all infrastructures.

PaaS – Platform as a service is a standard service for providing VMs with operating systems.

SaaS – Software as a service is a standard service for providing SW features for consumers.

Virtualized DTC contains physical and virtual servers, which serve a variety of services including web services, file services etc. The advantages of DTC are enabling application isolation from malicious or greedy applications cannot impact other applications co-located on the same physical server. Perhaps the biggest advantage of employing virtualization, is the ability to flexibly remaps physical resources to virtual servers in order to handle workload dynamics.

Server resources in a data centre are multiplexed across multiple applications and each server runs one or more applications. These applications are usually business critical applications with Quality-of-Service (QoS) requirements. The resource allocation needs to not only guarantee that a virtual container always has enough resources to meet its application's performance goals, but also prevent over provisioning in order to reduce cost and allow the concurrent hosting of more applications.

Our other aims are: To construct really stable and dynamically expandable Cloud computing for using remote laboratories. To create VMs and linkage for all parts in cloud, create communication links, virtual network for cloud computing inside, and all needed parts for Cloud computing concept. The goal of our work is new and acute topic of providing a new service for the consumers - completely functioning "Remote laboratory as a service" (RLaaS) [6].

It is very interesting for all clients of the Remote laboratories, because they can find this cloud concept and every remote laboratory. We created Consortium named REMLABNET and this is consortium of the three universities: Trnava University in Trnava (Slovakia), Tomas Bata University in Zlin (Czech Republic) and Charles University in Prague (Czech Republic). REMLABNET portal is on domain name or web site www.remlabnet.eu [7].

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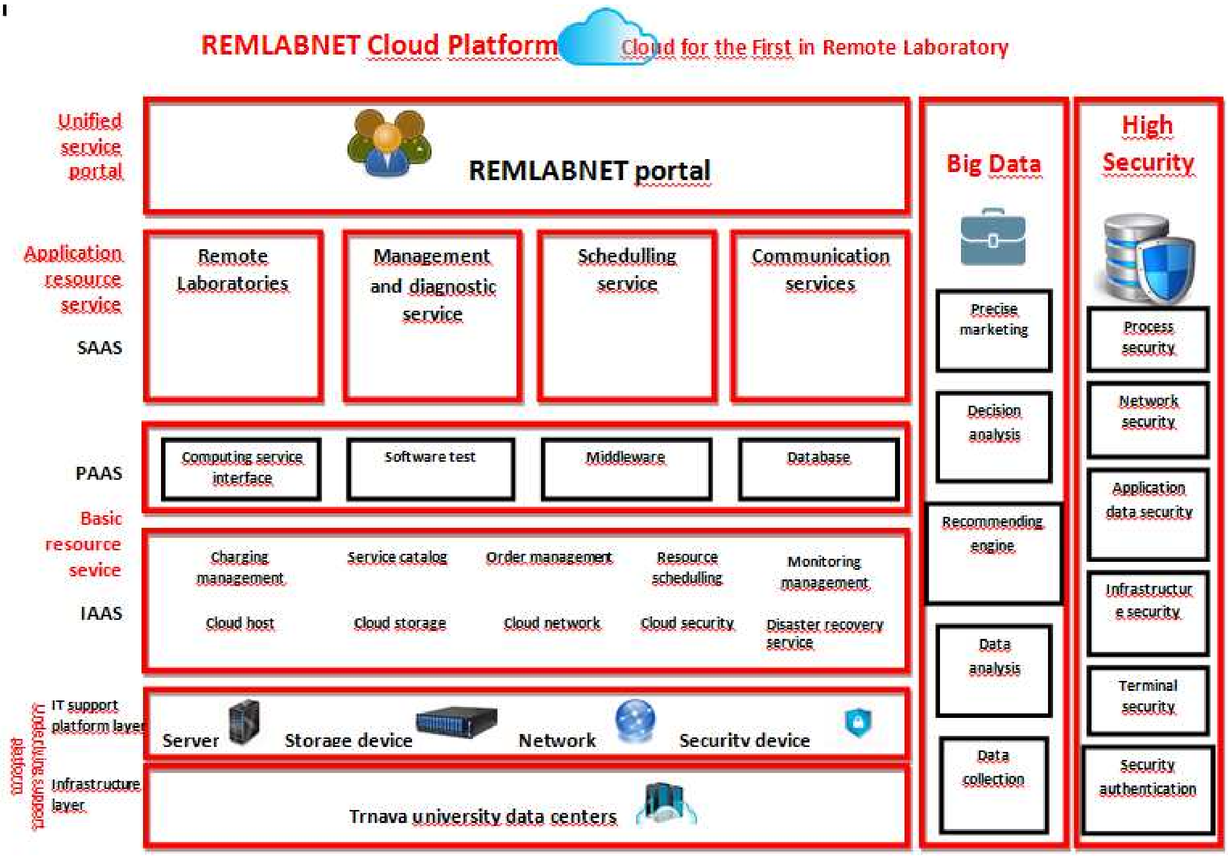

Figure 2 – Cloud computing concept in our Remote laboratory area

3 Thin client in RLs

In the world of client/server architecture, you need to determine, if it will be the client or the server that handles with the bulk of the workload. By client, we mean the application that runs on a personal computer or workstation and relies on a server to perform some operations.

Thick or thin client architecture is actually quite similar (Figure 3). In both cases, you can consider it as being the client application running on a PC, whose function is to send and receive data over the network to the server program. The server would normally communicate that information to the middle-tier software (the backend), which retrieves and stores that information from a database.

A thin client (TC) is a networked computer with few locally stored programs and a heavy dependence on network resources. It may have very limited resources of its own, perhaps operating without auxiliary drives, CD or DVD drives or even software applications.

Typically, a thin client is one of many network computers that share computation needs by using the resources of one server. A thin client often has low cost hardware with few moving parts and can usually function better in a hostile environment than a fat or rich client.

In global, we are talking about two categories:

1. thick client, called sometimes fat client,
2. thin client, where is difference in thin client or zero client (only HW configurations).

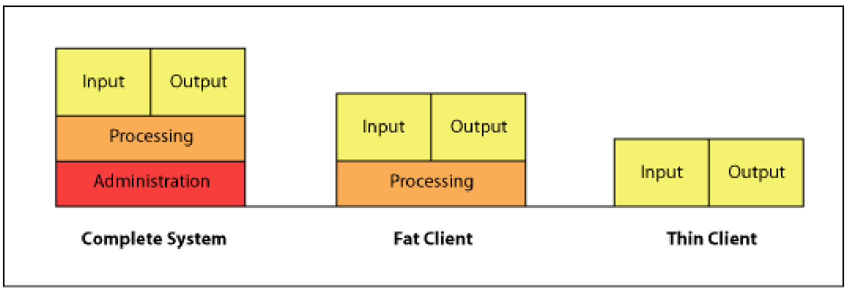

Figure 3 – Thin and Fat Client compared to a complete system [8]

We are arriving at decision to move to thin clients from local PCs for administration of the laboratories. TCs are sometimes called 'dumb terminals' and can offer a number of advantages:

- Increased security
- Easier upgrades
- Lower cost of ownership
- Reduced energy consumption
- Reliability

If we are looking at individual experiments, we can see these differences between Figure 4 and Figure 5, where we show using traditional PC architecture for laboratories and new concept with TC.

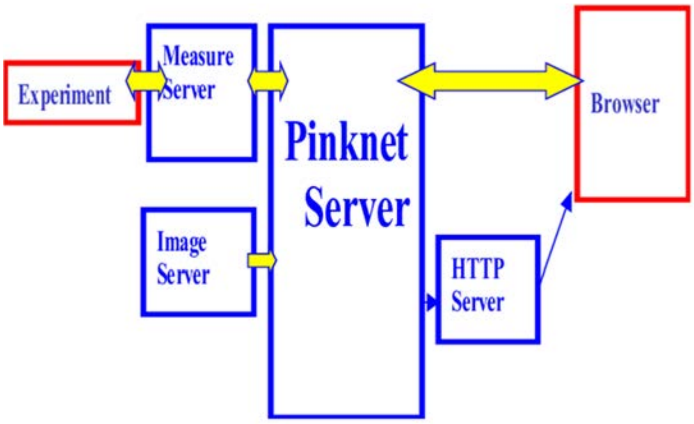

Figure 4 – Schematically representation of the remote experiment setup with Pinknet server and Image Server, Measure Server and HTTP Relay Server [9]

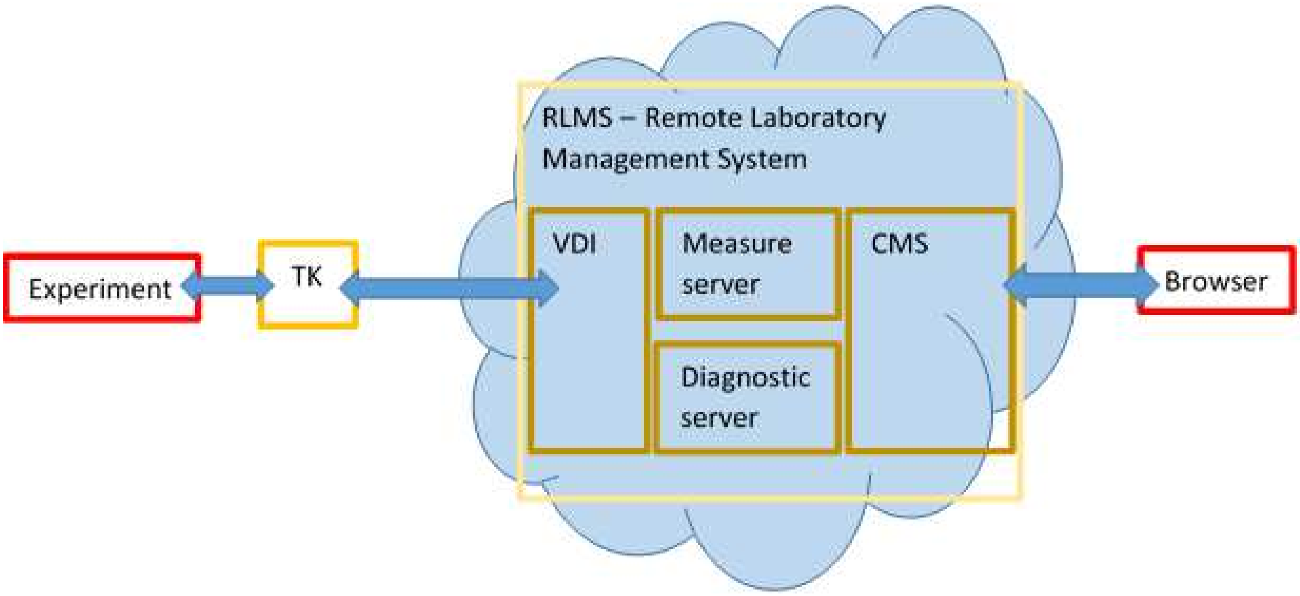

Figure 5 – New schema with using TC

We can see that all SW parts of experiment are moved to Cloud. Every part can start from template and this is good way for easier administration and possibility to connect different HW configuration of RLs to RLMS REMLABNET. In REMLABNET, we are trying two different styles of thin client. First is old system from Oracle SunRay (old Sun Microsystem SunRay) and second is thin client from Huawei. These are shown on Figure 6.

Figure 6 – Thin clients used in REMLABNET

The main architecture of TC in REMLABNET is figured on Figure 7. Every TC is connected to his own virtual machine (VM) via LAN (with separated VLAN) and Virtual desktop connector in REMLABNET named FlexControl. Flexcontrol system is set of SW instances for switching and management VMs. Every TC have allocated just one VM from virtual cloud. This allocation is handled by the MAC address to each TC.

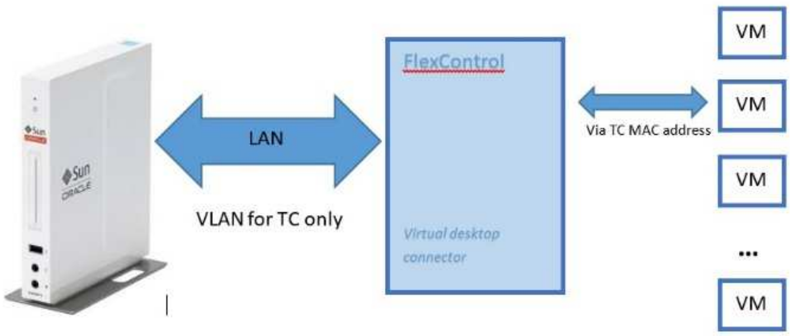

Figure 7 – Architecture of TC in REMLABNET

On each VM for TCs we have installed every needed part like Measure server, diagnostic server, and for some TCs also Image server. HTTP server and others are part of RLMS. FlexControl and VMs are part of virtualized cloud and can be used like cloud service.

4 Conclusions

Our idea of use Cloud computing was attested and discussed with experts in this research part. The way of our work is good and have a big progress. We can provide new service, Remote laboratory as a Service (RLaaS) in our cloud system and we are first to use thin client to communication between main experiment and virtual cloud. Our consumers are primary teachers, students and brainpower of the universities and high schools, but access is possible for all consumers via Internet. This show, how the university network is very overcast for communication and traffic. This claim, that network must be without failure and latency. And be secured too for management and research data protection. Security on the network is very important part, but it is without frame of this paper.

In this paper we showed our idea of construct Cloud computing system with important parts like using thin client. Our work is oriented to save money in education and research if everyone builds their own Remote laboratories. We have connected many laboratories from Zlin University, Trnava University, Charles University and other in the world. Our work is in simple terms „Bring Technology to Service!“.

Acknowledgement

The paper was published thanks to the University of Trnava, where the virtualized cloud is running. Thanks also belongs to IdeaCreative, which helps create cloud services and underlines research in the area of RLs. The biggest thanks belongs to Swiss National Science Foundation (SNSF) -“SCOPES” No IZ74Z0_160454, which helped the start of the REMLABNET environment and with the help of which several RLs were built. We would also like to thank prof. Javier Garcia-Zubia, for his valuable advice and support for our solutions.

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DETERMINATION OF POISSON NUMBER AT THIN ROD-SAMPLES WITH NON-STANDARD CROSS-SECTIONS BY PENDULUM METHODS

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Keywords: Tensile modules, Poisson number, vibration methods, thin samples, non-standard cross-sections
Abstract: The paper describes the measurements of modulus of elasticity of thin samples and related Poisson number by one device – Searle´s pendulum. We have focused our attention mainly to non-traditional samples with non-standard (i.e. other than circular) cross-sections.

1 Introduction, what is Poisson number?

The Poisson number μ belongs to basic physical constants characterising the elastic properties of solids. It is defined as the ratio of the relative transverse shortening and the relative longitudinal prolongation. This can be expressed by means of elastic modules as

$$\mu = \frac{E}{2G} - 1 \quad , \tag{1}$$

where E means the tensile modulus (or Young´s modulus) and G is the shear modulus.

In our task we have used a device that is able to measure both of these relevant quantities. This device is so-called "Searle´s pendulum", designed by American physicist G.F.C. Searle [1]. This device is commonly used to measure Young's modulus of thin specimens with classical circular cross-sections. We have extended this use to the measurement of samples with other cross-sections (some of them with a hollow character), and we used the vertical arrangement of the system to measure the shear modulus of elasticity G, too. It also presents a convenient way to determine the Poisson number μ.

2 Theoretical analysis and experimental procedures

Searle´s pendulum - in its classic form - is based on two flywheels connected by the sample being measured to create an oscillating system after deflection. The device can be used in two configurations: horizontal and vertical (Fig. 1a) and 1b)).

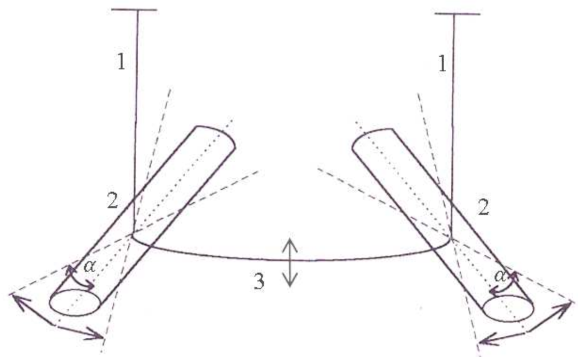


Figure 1 a) Horizontal flywheel set.
1 – hanging threads, 2 – cylinder flywheels, 3– measured wire (arrows indicate the direction of the oscillations, α is an angle of deflection)

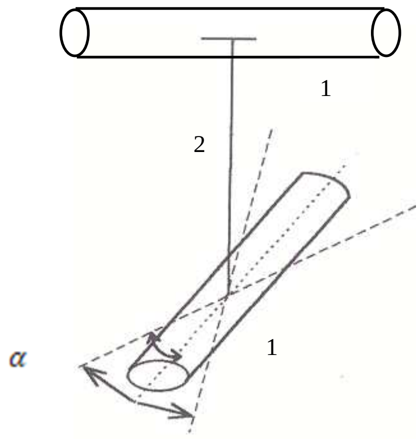


Figure 1 b) Vertictal flywheel set.
1 – cylinder flywheels, 2 – measured wire (arrows indicate the direction of the oscillations)

The principle of operation is essentially the same in both configurations: the vibration energy of the sample is "spilled" into the vibration of the flywheels, and vice versa. The sample performs bending oscillations in the first case and torsional ones in the second case. This can be illustrated by following scheme:

horizontal configuration → bending vibrations → E

and

vertical configuration → torsion vibrations → G

Dynamic analysis of pendulum operation results in relationships for flexibility modules [2]

$$E = \frac{4\pi^2}{T^2} \cdot \frac{lJ}{2J_A} \quad (2a)$$

and

$$G = \frac{4\pi^2}{T_G^2} \cdot \frac{lJ}{2J_A} \quad (2b)$$

Here J is the momentum of inertia, l presents the length of sample and T_E and T_G are the periods of oscillations, respectively. J_A represents the area moment of inertia (see below).

But now here we must differentiate the type of flywheels, too. In the case of cylindrical flywheels, as well as in our picture, the moment of inertia is given by the known relationship

$$J = m\left(\frac{L^2}{12} + \frac{R^2}{4}\right). \quad (3a)$$

The parameters R and L are the diameter and length of flywheels, and m means their (single) mass. The next possible cases would be:

Square flywheels

$$J = \frac{1}{12} m(A^2 + B^2), \quad (3b)$$

(A and B are the length and the width of the prism), and dumbbell flywheels

$$J = \frac{1}{4} mL^2, \quad (3c)$$

(L means the length of dumbbell).

The quantity of J_A represents the area moment of inertia with respect to the bending axis. This variable is different for otherwise shaped cross-sections, and we will pay more attention to it in the next part of the article; it will be summarized in Table 2 in detail.

3 Experimental part

3.1 The experimental assembly for measurements

We used the apparatus, that photo is – in horizontal arrangement – illustrated in Figure 2.

It consists from two homogeneous steel rollers in the role of flywheels, each having a mass m = 0.72 kg, a length L = 137 mm and a radius R = 14,6 mm. Size of moment of inertia of each of them, determined from the relation (3a), had a value of $J = 1.15 \times 10^{-3}$ kg.m^2.

We had used both - horizontal and vertical - configuration of device.

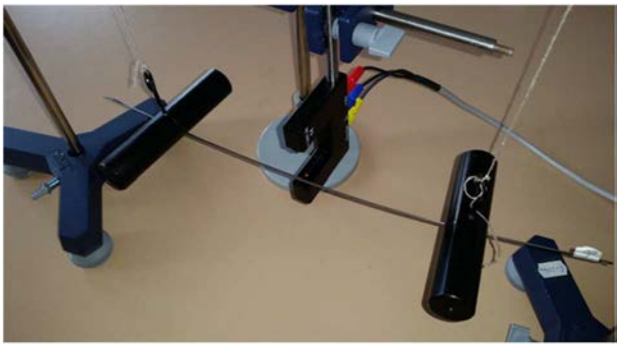


Figure 2 Experimental assembly in horizontal arrangement. Vibrating wire sample crosses the infrared beam of an optical

sensor (prismatic body with the shape of figure U in the centre of operation)

The arrangement of this assembly when measuring the torsion module is analogous, but it is oriented in the vertical direction.

We have performed measurements of several types of samples. First we measured samples with a classic full circular cross-section, then we measured samples with unconventional cross-sectional character.

3.2 Samples with classic (circular) cross-sections

We performed test measurements for standard circular cross-sectional samples. We used thin wires of different materials and cross-sections.

The moment of inertia of such samples is

$$J_A = \frac{1}{4}\pi r^4 \qquad (4)$$

After its incorporation into relations (2a) and (2b) we get expressions for the modules of elasticity

$$E = \frac{8\pi l}{r^4 T_E^2} \qquad (5)$$

and

$$G = \frac{8\pi l J}{r^4 T_G^2} \qquad (6)$$

The pendulum times T_E and T_G, that are important in these measurements were scanned electronically.

All the samples had the same "active" length (i.e. the distance between the points of attachment to flywheels) $l = 0{,}30$ m, and the same radius $r = 1$ mm. A review of results of measurements of them is given in Table 1.

Table 1 The results of measurements of samples with standard circular cross-sections, diameter r = 1 mm

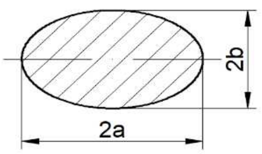
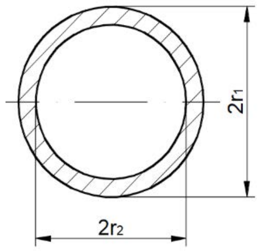
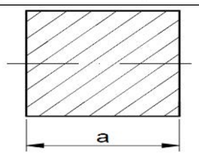
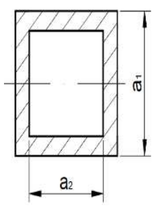
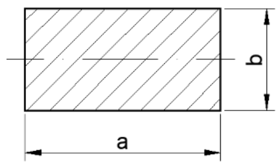
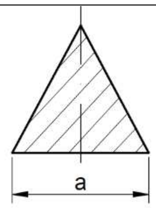
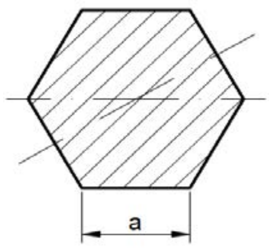
Sample	Period of bending oscillations T_E (s)	Period of torsion oscillations T_G (s)	Tensile modulus E (GPa)	Shear modulus G (GPa)	Poisson number	
					Measured μ	Table valued parameter
Steel	0.238	0.374	202	81	0.28	0.28-0.31
Copper	0.303	0.499	123	45	0.34	0.34-0.35
Alluminium	0.399	0.654	71	26.5	0.32	0.27-0.30
Brass	0.338	0.557	99	36.5	0.36	0.35-0.37
Polyamid (nylon 6,6)	2.32	3.88	2.1	0.75	0.40	0.39-0.41
PVC	1.97	3.41	2.9	1.01	0.42	0.40-0.42
Polystyrene	1.73	3.94	3.7	1.37	0.35	0.34-0.37
Polypropylene	2.51	4.31	1.8	0.66	0.37	0.36-0.41
Polyethylene	2.66	4.53	1.6	0.56	0.41	0.42-0.46

3.3 Samples with unconventional cross-sections

Using this method, we had measured the tensile modulus of several unconventional metal and plastic samples, and one wooden specimen, moreover.

An overview of these patterns being possible is summarized in Table 2.

Table 2 Overview for different possible cross-sectional shapes of wire samples

Cross-section		Area moment of inertia
Ellipse		$J_A = \frac{1}{4}\pi a^3 b$
Circle - hollow		$J_A = \frac{1}{4}\pi(r_1^4 - r_2^4)$
Square - full		$J_A = \frac{1}{12}a^4$
Square - hollow		$J_A = \frac{1}{12}(a_1^4 - a_2^4)$
Rectangle		$J_A = \frac{1}{12}ab^3$
Triangle (equilateral)		$J_A = \frac{\sqrt{3}}{96}a^3$
Hexagon (regular) side and/or top axis		$J_A = \frac{5\sqrt{3}}{16}a^4$

In the last column there are presented the relations for the calculation of area momentum of inertia, that stands out in expressions for receiving the modules *E* and *G*.

An overview of the measured samples, including the relevant geometric parameters and the values being obtained, is given in Table 3.

Table 3 Parameters and results of measurements of samples with non-standard cross-sections

Sample	T_E (s)	T_G (s)	*E* (GPa)	*G* (GPa)	μ	μ_{tab}
Steel – circle hollow r_1 =1,1 mm; r_2 = 0,75 mm	0.195	0.312	204	79.8	0.28	0.28-0.30
Polypropylene – circel hollow r_1 = 1,5 mm; r_2 = 1 mm	1.12	2.00	1.7	0.61	0.39	0.36-0.41
Polystyrene – ellipse *a* = 1,5 mm; *b* = 1 mm	1.58	2.61	2.3	0.84	0.37	0.36-0.40
Polyethylene – ellippse *a* = 1,5 mm; *b* = 1 mm	1.89	3.22	1.6	0.55	0.45	0.42-0.46
Polystyrene – square full *a* = 2 mm	1.49	2.43	2.4	0.9	0.33	0.36-0.40

Polyamide – square full a = 1,5 mm	2.77	5.28	2.1	0.71	0.41	0.39-0.41
Polypropylene – square hollow a_1 = 2 mm; a_2 = 1 mm	1.74	3.39	1.8	0.62	0.45	0.36-0.41
Polypropylene – rectangular a = 2,7 mm; b = 1 mm	1.56	2.60	1.7	0.63	0.38	0.36-0.41
PVC – rectangular a = 2,5 mm; b = 1,2 mm	1.25	2.12	2.8	0.97	0.44	0.38-0.43
Polyamide (nylon) – triangle a = 2,5 mm	2.14	3.59	2.1	0.75	0.40	0.38-0.42
Polyamide – hexagonal a = 1,5 mm	2.97	5.54	2.0	0.73	0.37	0.39-0.41

3.4 Simplification of the evaluation process

We have determined the Poisson number through the elastic modules E and G.

However, there exists also an easier way: After substituting expressions for E and G from (6) and (7) to (1), the expression will be considerably simplified to form (6) and (7) to (1), the expression will be considerably simplified to form

$$\mu = \frac{T_G^2}{T_E^2} - 1 \qquad (7)$$

In this way, we should eliminate "uncomfortable" quantities E, G and J_A from the calculations.

4 Conclusion

Searle's pendulum, though a simple and fairly accurate device to measure elastic modulus, is used relatively few (unfairly in our opinion). Additionally, it should be mostly in the cases of "slow oscillating" samples of circular cross section.

The equipment being described is simple and illustrative, completing the range of pendulum-based methods for the measurements of elasticity constants. It does non-require intricate measuring equipment and works without destruction, practically. Even extremely thin samples can be measured without a risk of damage or permanent deformation. The activity of pendulums is stable, the system phases do not "tune out" or dump even after several tens or hundreds of oscillations.

The results are sufficiently precise, as evidenced by the fact that the measured values are well correlated with the table values. The measurement error did not exceed 8% accuracy to statistical calculations [3].

This method can be used successfully in the wires, plastic and textile industries (investigation of elasticity of thin materials [4]), in botany (elasticity of stalks) and the like. As so as a demonstration chapter in university textbook (section of "Vibrating Movements" or "Solid State Physics"), or a task for laboratory exercises [5].

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