An Experimental Investigation on the Static Response of Rotating Disks

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ABSTRACT

An experimental study is carried out to investigate the static response of rotating annular disks of uniform and taper thickness and validate results obtained from theoretical analysis. The study of static response involves investigation of stress and displacement behaviour at a certain angular speed of the disk. When a symmetric disk is rotated at certain speed, radial and tangential stresses are generated in the disk due to action of centrifugal loading. As the speed increases, the stresses also increase and finally at a certain speed the stresses cross the yield limit. This particular speed is termed as limit angular speed. It is the purpose of this experimental study to determine the strain and displacement behavior and subsequently the stresses in disk rotating at and below limit angular speed. It is observed that the mounting arrangement of the disks greatly offset the resulting stress distribution, thus requiring more care in the mathematical modeling.

1. INTRODUCTION AND LITERATURE REVIEW

Estimation of rotating disk stresses at high rotational speeds has been a subject of longstanding attention to many a researcher, the reason being its vast industrial applications, starting from enormous flywheels to the smallest of gears. The most primitive form of a rotating disk is a thin cylindrical object and its state of stress is a function of its angular speed. It is well understood that stresses in disks are maximum near the axis of rotation and hence the thickness of the primitive thin cylinder is varied radially, without hampering its axisymmetric nature, so that more material is accumulated at the root of the disk. This leads to higher range of operating speeds apart from reducing the rotary inertia of the disk. The objective of the present experimental study is to obtain stress distribution in a rotating annular disk of uniform and taper thickness.

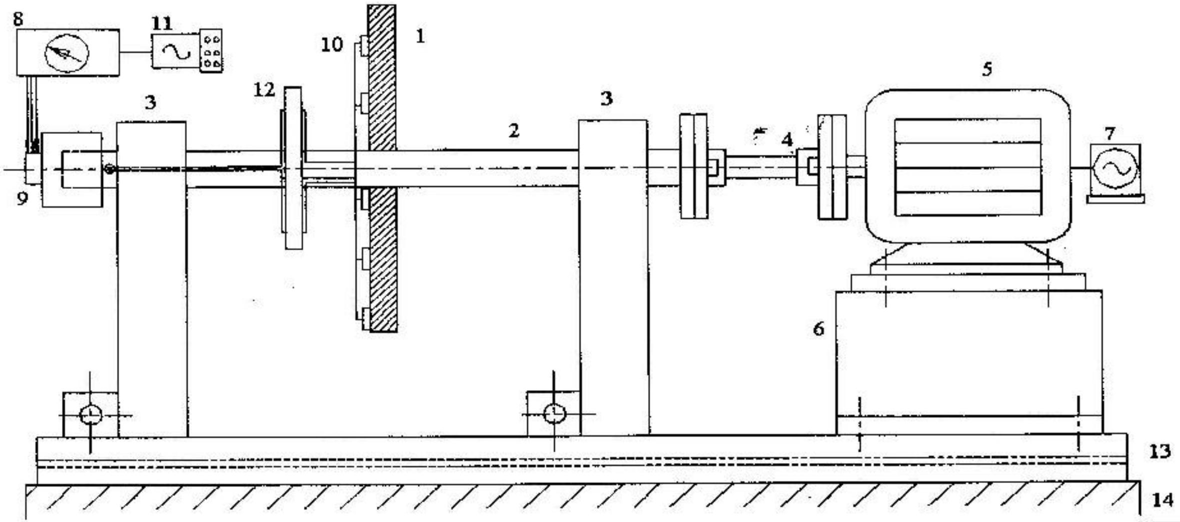
Several researchers have carried out experimental work to investigate the effect of different parameters on rotating disk behaviour and these experimental investigations provided excellent practical insights into the behaviour of rotating disks. The credit for initiating an experimental analysis probably goes to Robinson (1944) who tested solid and annular disks of both uniform thickness and taper profile and reported a semi-empirical criterion according to which bursting of disk is supposed to occur only when the average tangential stress was at par with the tensile strength of the material. Skidmore (1951) further extended the criterion by conducting a series of experiments and confirmed the results proposed by Robinson (1944). Waldren et al. (1965) conducted experiments on bursting of rotating disks made of vacuum melted steel and on comparing the theoretical and experimental results concluded that the large plastic strains that occurred were in proportion with the strain predicted using plasticity theory. Percy et al. (1974) observed permanent strain distributions along with instability and fracture conditions on disks made of the same material. Review also indicates that there exists some experimental study focused on the contribution of the location of the rotor burst zone. The basis of such studies was to determine the shape and size of segments into which the disks will burst to make an estimate of maximum translational energy contained in the fragments. Sato and Nadai (1963 a, b) performed experiments on cast iron disks of constant thickness and observed that the number of disk fragments at the failure increase with increase in the ratio of inner to outer radius. Mangano (1975) collected data and analyzed practical failure of 170 disks and found that rotors usually burst into 6, 4, 3 or 2 pieces.

Review work indicates that although there exist a host of analytical studies, experimental investigation on static response of rotating disks is rather scarce. Hence this experimental study is aimed at investigating the strain, displacement and subsequently the stress fields in rotating disks to validate the existing theory.

2. EXPERIMENTAL SETUP

The general assembly of the experimental set up is shown in fig. 1. The set up comprises of a rotating annular disk mounted on a rigid shaft. The shaft is simply supported between two bearings and connected at one end to a 3-phase induction motor by means of a universal joint. The motor speed is controlled by using a variable speed A. C. drive. The motor is supported on a base plate and the complete assembly is mounted on a machine bed. The other end of the shaft carries a rotating electrical connecter. The connecter is used to transmit signal from sensors attached to the rotating disk to a strain indicator. It consists of a static and a rotary component and is described in detail in a separate section.

Between the two bearings, an annular disk is mounted on the shaft, which carries five set of strain gauges at five different radial locations as illustrated in table 1. Each set consists of a pair of strain gauge, one attached radially and the other tangentially (fig. 2). For five sets, the arrangement provides 15 terminal points, and hence a manually operated relay switch is arranged to allow signal transmission for each strain gauge separately. Output signal from electrical connecter is supplied to strain indicator. At lower angular speeds, however, the readings in the strain indicator are insignificant and as a result, output from strain indicator is relayed to an oscilloscope, where output in terms of voltage is recorded. A dummy gauge is used to allow for the provision of balancing the strain gauge bridge circuit.



1. Rotating disk 2. Shaft 3. Bearing support 4. Universal Coupling 5. 3-ph induction motor 6. Base plate for motor 7. A.C. drive 8. Strain indicator 9. Totating connecter 10. Strain gauges with cables 11. Osalloscope 12. Relay switch 13. C.I Machine bed 14. Reinforced concrete slab

Fig. 1 General assembly drawing of the experimental setup

2.1 Specimen geometry

An annular disk of uniform thickness made of commercial mild steel is used as the first specimen. Outer diameter of the disk is 500 mm and it is mounted on an adapter sleeve with dimensions corresponding to a 25 mm diameter shaft, the annular inner part of the disk being tapered to match the outer surface of the adapter sleeve. However, the sleeve thickness being negligible compared to disk dimension, the average inner diameter of the disk can be considered equal to the outer diameter of the adapter sleeve at its midpoint (31.3 mm). The disk thickness is 20 mm.

A second specimen is fabricated in the form of an annular taper disk, the outer and inner diameter of which are 500 mm and 25 mm respectively. The disk thickness at the annulus is 39 mm and reduces linearly to 15 mm at the outer periphery. The disk is directly mounted on the shaft and gripped with threaded bush-adapter sleeve assembly on both sides to transmit rotational motion. The details of the specimen are shown in fig. 2, along with the strain gauge locations.

Table 1 Locations of the gauges [in mm]

Strain gage location	Radial	Tangential
L1	R1=237.5	T1=246.5
L2	R2=187.5	T2=196.5
L3	R3=137.5	T3=146.5
L4	R4=86.5	T4=96.0
L5	R5=37.5	T5=46.5

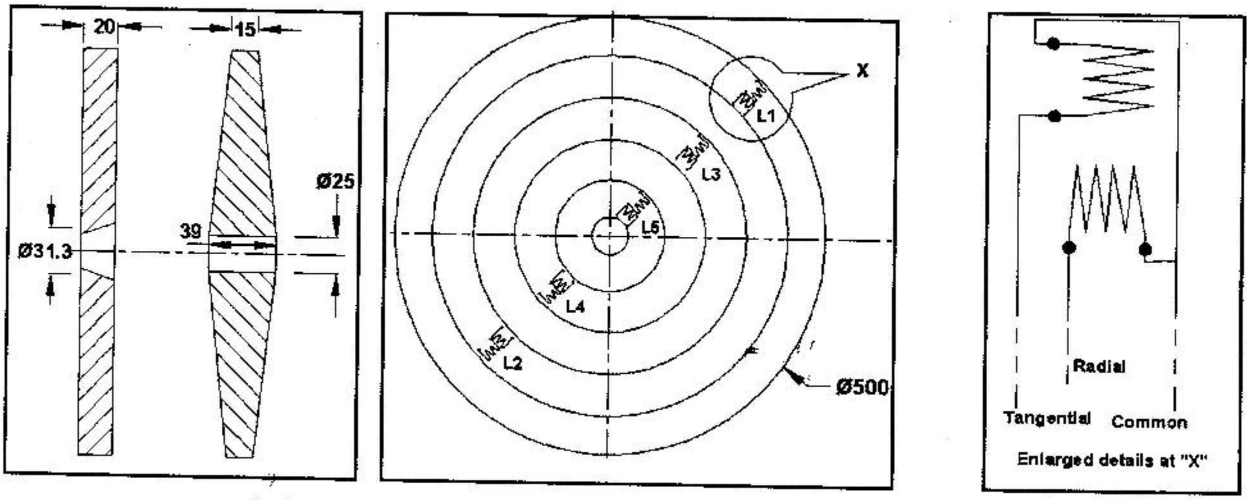
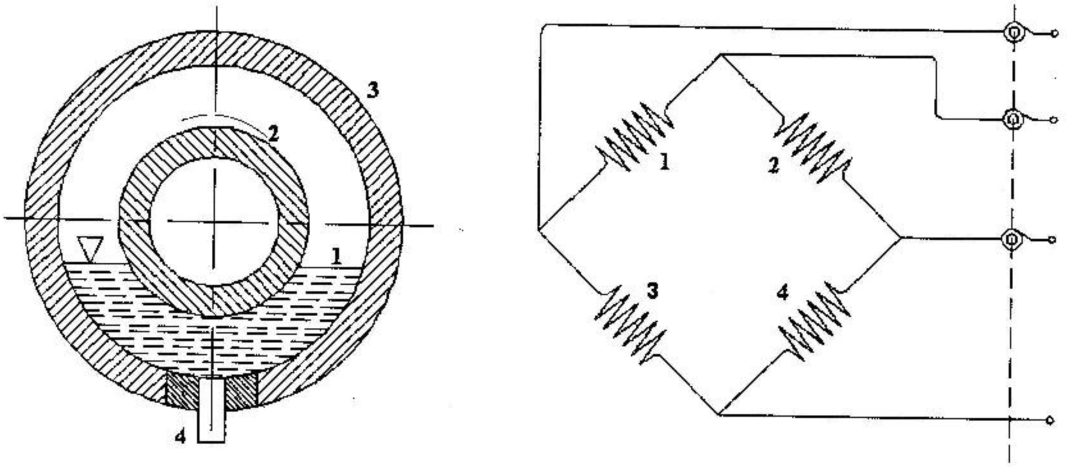


Fig. 2 Details of disks and Strain gauge locations



1. Mercury pool 2. Rotating ring
3. Stationary housing 4. Terminal

1 & 2. Gauges on rotating disk
3 & 4. Gauges/resistances in the measuring circuit

(a) Construction

(b) Bridge circuit configuration

Fig. 3 Details of rotating connector assembly and circuit arrangement

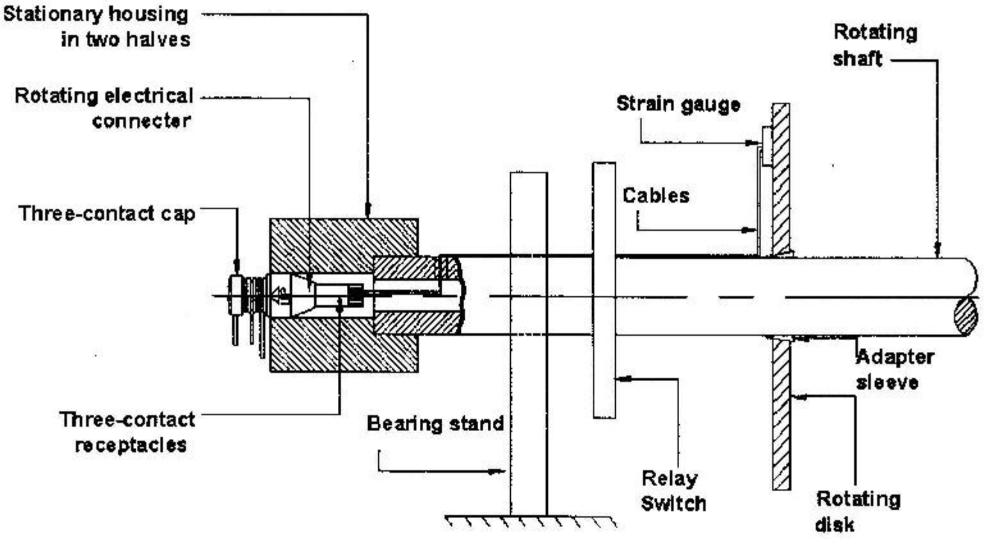


Fig. 4 Rotating electrical connecter arrangement

2.2 Details of rotating electrical connector

When transducers are to be mounted on rotating members of machines, some means must be there to provide excitation power to the transducer and to take away the output signal. For continuous high speed application slip rings are commonly used with noble metal rings and block type silver-graphite brushes. However for strain gauge applications, a small variation in sliding contact resistance may be comparable with the small strain gauge resistance change to be measured. Hence instead of conventional slip ring a rotating disk dipped into mercury pool is used for this purpose as shown in fig. 3. The commercially available unit corresponding to the operating speeds had only 3 terminals and hence a full bridge for strain measurement could not be used. The arrangement of the bridge circuit is also shown in the figure. Static end of the connecter carries a three-contact cap which serves as output plugs. Input is supplied at the rotating end of the connecter from disk/relay switch by means of a three contact receptacle. The connecter-shaft assembly is shown in fig. 4. The connection between static and rotary part is made through a pool of liquid metal molecularly bonded to the contact, which provides a low resistance, stable connection. Under rotation, the fluid maintains the electrical connection between the contacts without any wear. Photographs of the experimental setup are provided in fig. 5.

2.3 Instrument specification

A.C. drive (frequency controller):

Model no.: VFD037M43A, Make: Delta electronics, Taiwan

Input: 3-phase, 480 V, 50/60 Hz., 8.5 Amp.

Output: 3-phase, 0 to 480 V, 8.2 Amp., 6.2 kVA, 5 H.P., Frequency range: 0.1 to 400 Hz.

Induction motor:

Model no.: 01013409DGA, Make: NEC motors Pvt. Ltd. India

3-star phase connection, 415 ± 6% V, 3.8 Amp. 1.5 kW (2 H.P.), 1425 r.p.m. (η:76%).

Strain Gauge:

Gauge type: CEA-240UZ-120, Lot no.: A44AD569

Code: 113712-6835, Make: Micro Measurements Div., Measurements Inc., USA

Resistance (in ohms): 120 ± 0.3% at 24° C, Gage factor: 2.08 nom at 24° C

Transverse sensitivity: (+0.4 ± 0.2)% at 24° C, Temp. range: -75° C to 175° C

Strain indicator:

Model no: Lo 1808 Make: Philips PR 9307

Voltage: 110/245 V, Max. Power: 11 VA, Frequency range: 45-400 Hz

Oscilloscope:

Model no: TDS-210, S. no.: B071515, Make: Tektronix

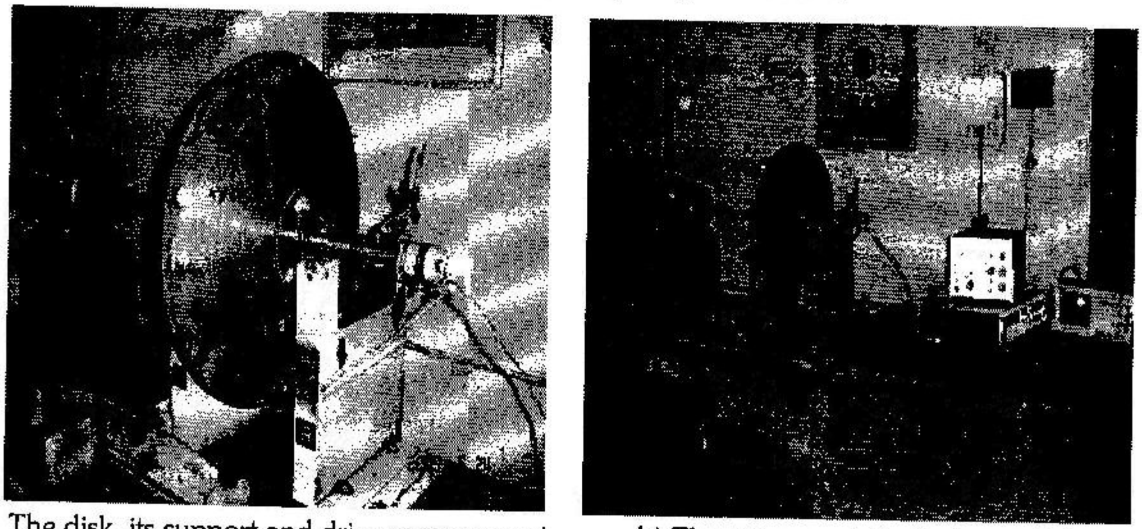
Voltage range: 120-240 V, Frequency range: 45 to 66 Hz, Max. power: 25 W, Max. VA: 35

Rotating electrical connecter:

Model no.: 305, Make: Mercotac Inc., USA

3 conductors, 0 to 250 V, 4 Amp., 100 gm-cm rotational torque, Max. Freq.: 200 MHz,

Max r.p.m.: 1800 Contact resistance < 1m , Temp. range: 7° to 60° C



a) The disk, its support and drive arrangement b) The set-up and the instrumentation

Fig. 5 Sample photographs of the experimental setup

2.4 Testing procedure

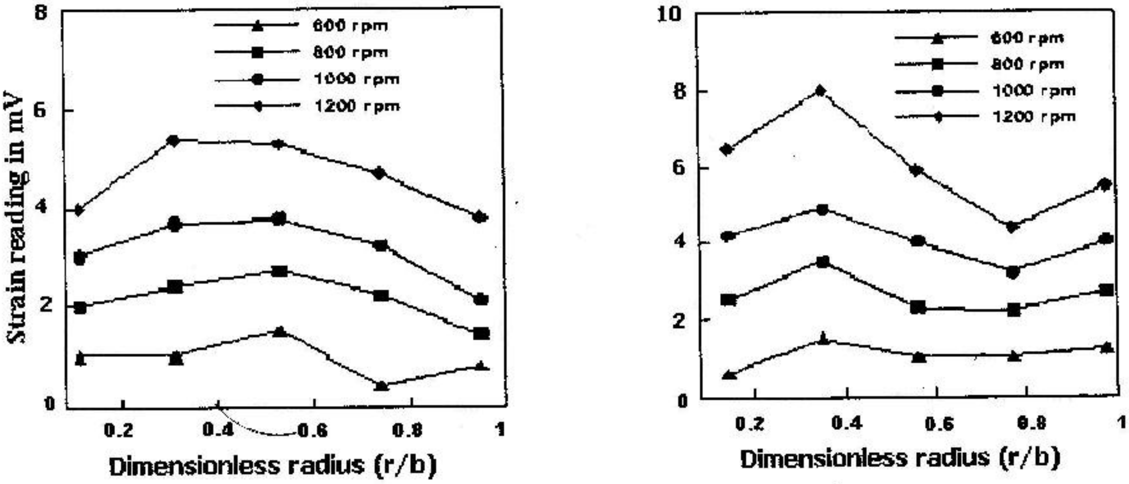
Following are the steps carried out while conducting the experiment:

1. Before starting the test, arrangement is made at the relay switch to ensure that proper sensor location is selected. It is further ensured that at the selected location (say L1), proper strain gage (say T1 at r = 246.5 mm) is opted and corresponding cable is connected to the rotating electrical connecter. The bridge is balanced by using the bridge balancing resistor.
2. After balancing the bridge, the disk-shaft assembly is rotated at a specified r.p.m and readings are noted from the strain indicator. Experimentation begins with outermost sensor location (L1) being selected for the first reading followed by rest of the locations. Initially tangential stress is predicted followed by the radial stress at each location simultaneously. The output is obtained in terms of voltage (mV) reading. The experiment is repeated for each strain gauge and then the rotational speed is changed.
3. As the strain corresponds to the stress states at selected locations, the strain in terms of voltage readings are plotted in fig. 5 for the uniform disk. It is well understood that the plot will readily give an insight into the actual stress distribution.
4. Calibration of strain gauges is carried out separately on a standard beam bending test setup. The resulting chart can be used to obtain the gauge factor using which the corresponding strain reading at each location can be calculated.

3 TEST RESULTS AND DISCUSSIONS

Radial and tangential strains at each strain gauge location are noted at different rotational speeds ranging from 600 rpm to 1200 rpm at an increment of 200 rpm and plotted in fig. 6 (a-b). Results are also obtained from theoretical analysis [Bhowmick et al. (2004)], assuming free boundary conditions at both the edges, and they are furnished in fig. 7. It is quite evident that except at low speed (600 rpm) the radial strain of rotating disk agrees with the theoretical results (fig. 7 (a)).

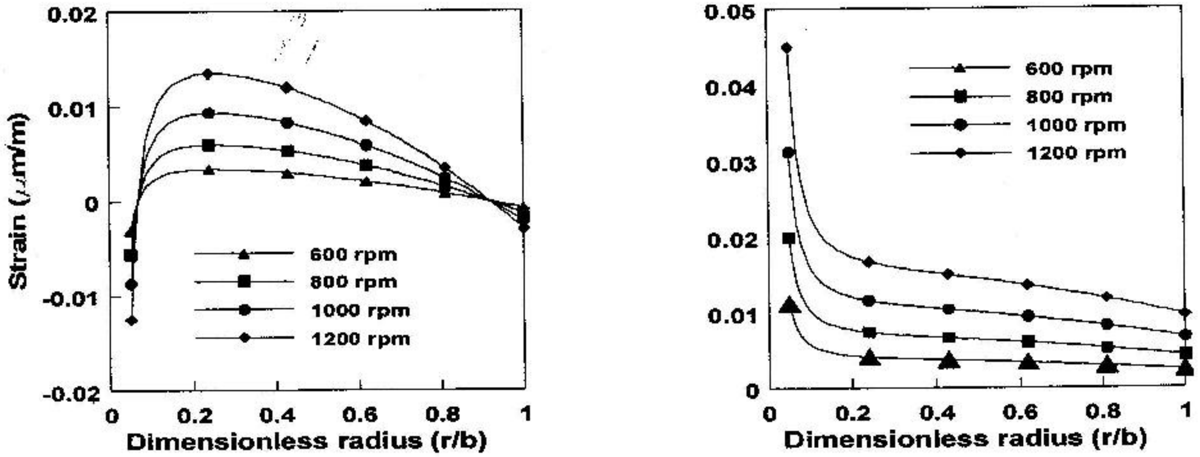
The tangential strain of the disk at each location except near the axis also nearly agrees with the theoretical ones (fig. 7 (b)). The reason of deviation near the axis is due to the fact that the theoretical formulation of annular disks is based on free inner and outer boundary conditions where as in practice the disk is shrink-fitted on the shaft through adapter-sleeve assembly.



(a) Radial strain

(b) Tangential strain

Fig. 6 Plot of experimental values for the uniform disk



(a) Radial strain

(b) Tangential strain

Fig. 7 Plot of theoretical strain values for a uniform disk

In case of the taper specimen, the problem of shrink fit does not arise as the disk is directly mounted over the shaft and drive is transmitted by gripping the disk on both sides with threaded bush. The investigation provides similar results to those of uniform disk and hence is not reproduced here to maintain the brevity.

4. CONCLUSION

The present study is carried out to investigate the static response of rotating annular disks of uniform and taper thickness and validate results obtained from theoretical analysis. The experiment shows that, except at low speed, the nature of the radial and tangential strain distribution matches with the trend obtained from the theoretical calculations. It is observed that the mounting arrangement of the disks greatly offset the resulting stress distribution, thus requiring more care in the mathematical modeling.

5. ACKNOWLEDGEMENT

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