Measuring Tape-Like Sampling Arm and Drill for Sampling Lunar Regolith

Abstract For the task of shallow lunar regolith sampling, we have designed a novel Measuring Tape-like Sampling Arm and Drill (MTS) which is a kind of flexible robot. The advantages of the sampling arm are that it has a small shrink volume, long working distance, light weight, low power consumption and good adaptivity, which is driven by one motor only. Furthermore, it has a large reverse torque with a short force arm, so the pushing force is large enough for the end-effector to penetrate into the lunar regolith. The sampling head which includes two opposite rotating parts has no-reaction screw force. The vibrator located in the sampling head can vibrate at frequencies ranging from 1 to 42Hz and can result in resonance of the sampling head. Therefore, it can improve the efficiency of thrusting, sampling and casting throw. We have carried out theoretical analysis, simulations and experimental studies and it is has been proved that the sampler can sample quantitatively at a depth of 10 cm in simulated lunar regolith (model CAS-1), cement and sand.

Keywords Lunar Exploration, Flexible Robot, Sampler, Drilling Techniques

1. Introduction

As the nearest celestial body, the moon is the key to answering some questions that are still open, such as the origin of the solar system and the beginnings of life on Earth. Furthermore, there is a lot of He in the regolith of the moon, which could supply the whole earth for ten thousand years [1]. Therefore, one of the most important tasks for lunar exploration is the collection of lunar regolith samples, in-situ analysis and possibly the return of the sample to Earth for further tests.

This indispensable tool, the sampler, is generally composed of a sampling arm and a sampling head, i.e., end-effector. Numerous researchers have studied and developed many sampling arms and drills or corers for planetary exploration. The existing samplers can be classified into two categories: rigid sampling arms and flexible sampling arms. The rigid sampling arms include single-rod arms and multi-stems arms. The structure of the Soviet samplers used in Lunar 16 (1970) and Lunar 20 (1972) is a single-rod [2]. The connecters used in multi-stems typed arms are generally joints, rack-pinion or spiral tube-screw. The structure of the lunar regolith...
sampler of “Surveyor 5” (1967) is a parallel connecting rods structure which can expand or retract by controlling one of its ends [3]. The Instrument Deployment Device (IDD) of NASA’s Mars Exploration Rovers (2003) and the JPL’s Rover are tandem joint type structure arms which have more freedom, although they possess too much weight and contraction volume[4,5]. The structure of the Jet Propulsion Laboratory & Honeybee Robotics rotary hammer ultrasonic/sonic drill system is a spiral tube-screw, which is about 2.7kg in weight and 15cm×10cm×30cm in size[6].

The arms of Mole-typed Deep Drillers (ESA in 2003[7], Japan in 2005[8] and “Earthworm” type samplers [9] from Japan (2007) are soft wires, and they have little hardness and cannot guide well. The samplers can only be used one time. The penetrator of the ESA/NASA Europa-Jupiter System Mission penetrator mission is delivered to planetary body surfaces by a Penetrator Delivery System in order to penetrate the surface (2010)[10]. It has good depth but poor position accuracy and needs complex control strategies. The structure of the deployable wood wasp drill is an elastic structure which has a light weight, low power consumption but a low sampling efficiency and small sampling volume (2006)[11,12].

From above, it is clear that the rigid arms have a large pushing force, good position accuracy and are easily controlled but with a large weight (several or dozens of kilograms), large contraction volume (maximum size for dozens or even hundreds of cm) and high power consumption (dozens or even hundreds of Watts). The flexible arm typed samplers have a small shrink volume, light weight and low power consumption, but they cannot be applied in accurate position sampling and need complex control algorithms and navigation technologies.

It is well known that a snake has a small size when coiled but has a long body when uncoiled, which inspires the origin of this paper, where we aim to present a novel sampler concept MTS (Measuring Tape-Like Sampling Arm and Drill ) that addresses the limitations of large mass, large volume, high power consumption, short working path and poor position accuracy. It can be used in lightweight lunar rovers for in-situ quantitative sampling.

2. Mechanical model of the sampling arm

The main part of the sampling arm is a tape spring, widely known as a carpenter tape, which is a straight strip section of an opening cylindrical shell. Before analysing the maximum pushing force of the sampling arm, it is necessary to introduce the geometry and general bending properties of the tape spring first. The major geometric parameters of an opening cylindrical shell are the shell thickness (t), cross-sectional radius of curvature (R), angle (θ) and length (ℓ) (see Fig.1). Its material parameters include Young’s modulus (E) and Poisson’s ratio (μ). These parameters work together to determine the mechanical properties of the sampling arm. The general coordinate system used throughout this study is shown in Fig.1.b.

The bending behaviour of a tape spring can be described as having two types of fold, equal sense bending and opposite sense bending as seen in Fig.2.

![Figure 1. Geometric parameters and coordinate system of the sampling arm](image1)

![Figure 2. Folding of a tape spring](image2)

Simplified schematic moment-rotation behaviour for a tape spring is shown in Fig.3. It should be noted that there are four key moment values identified for folding. For opposite bending the tape behaves linearly up to a maximum moment, M_{max}^o, then it snaps and behaves as a constant moment, M_{c}^o. It is the same for equal sense bending; there is a maximum moment, M_{max}^e, and a constant moment, M_{c}^e.

![Figure 3. Sketch map of equal and opposite bending moment](image3)

3. The design of the sampler

The maximum moment of a tape spring in opposite sense bending is much larger than that in equal sense bending. So the design uses a tape spring as a sampling arm and utilizes the significant moment difference to achieve a large pushing force and a small rolling force.
A novel type of sampler has been developed which mainly includes four parts: a box for retractable tape spring, open cylindrical shell, driving structure and a sampling head (see in Fig.4). As shown in Fig.5, the cylindrical shell rolls on the outside of the recovery wheel the inside of which connects with a contraction roll-up spring, and the inner end of the roll-up spring is fixed at the centre cylinder of the box. The contraction force of the roll-up spring drives the recovery wheel to rotate inward which also means the cylindrical shell is always in an inward contraction trend.

In Fig.4, the cylindrical shell is gripped by the convex and concave rollers the shape of which matches well with that of the cylindrical shell. The surfaces of the convex roller and concave roller are covered with soft rubber (or soft metal) to ensure the cylindrical shell can get the maximum moment ($M_{max}$).

![Figure 4. Structure of sampling arm: [1] Box for retractable tape spring, [2] Driving structure, [3] Open cylindrical shell i.e., sampling arm, [4] Sampling head.](image)

The concave roller is connected to the slots in both sides by bearings which are fixed in the brackets by strings. The convex roller which is coated with soft rubber and acts as friction wheel is fixed in the brackets by slots and connected with the motor by a coupling. The friction transmits the movement to the open cylindrical shell when the motor rotates. So the sampling arm moves down when the motor rotates counter clockwise, and vice versa. Only one 3W motor is used to achieve the sampling arm extension and contraction, so the power consumption of the arm is very low.

![Figure 5. Box for retractable tape spring: [1] box, [2] contraction roll-up spring, [3] fixed cylinder, [4] open cylindrical shell, [5] recovery wheel.](image)

Fig.6 shows the mechanism of the prototyped sampling head. A 0.5 W longitudinal vibrator and a 0.3 W torsional vibrator, the frequency of which can be adjusted, are embedded between a lid and a transducer horn from above to below in order. The 1 W sampling motor is fixed to the enclosure and it can drive the sampling spoons to rotate in different directions at the same speed using gears, so the sampling head shows no tortion when it is working. Besides, the sampling head has a small volume, light weight, low power consumption and it can sample a full scoop of about 5 cm$^3$ simply by rotating the spoon one circle, hence it can be used in shallow lunar regolith in-situ sampling quantitatively.

![Figure 6. Lunar regolith sampling head: [1] lid, [2] longitudinal vibrator, [3] torsional vibrator, [4] transducer horn, [5] sampling motor, [6] gears, [7] external sampling spoon, [8] internal sampling spoon.](image)

4. Theoretical analysis of the sampler

4.1 Statics analysis

We firstly obtained the opposite sense moment with regards to curvature of a tape spring made of isotropic material. The moment curvature relationship of a tape spring is subjected to equal and opposite end moments, considering a slightly distorted axi-symmetric cylindrical shell. By integrating the moments along the transverse axis for the whole cross section of the tape spring, the end moment was obtained by Walker et al. as[13,14].

$$M = \int_{s/2}^{s/2} (M_f - N_f w)dy = sD(k_e + \frac{\mu}{R} - \frac{1}{k_e} + \frac{\mu}{R})F_1 + \frac{1}{k_e}F_2 + \frac{1}{k_e}F_2$$

(1)

Where, $s$ is the width of the tape spring; $M_i$ is the bending moment per unit length and $N_f$ is the axial force per unit length; out-of-plane deflection, the y-axis corresponds to the longitudinal direction; $D$ is the bending stiffness, in which E is the Young’s modulus, $t$ is the thickness of the tape spring, $\mu$ is Poisson’s ratio. $R$ is the initial transverse radius of the tape; $k_e$ is the longitudinal curvature; $s, D_i, F_1, F_2$ and $\lambda$ are calculated...
from \( s = 2R \sin(\theta / 2) \),

\[
D = \frac{Et^3}{12(1 - \mu^2)},
\]

\[
F_1 = \frac{2 \cosh \lambda - \cos \lambda}{\lambda \sinh \lambda + \sin \lambda},
\]

\[
F_2 = \frac{F_1}{4} - \frac{\sinh \lambda \sin \lambda}{(\lambda \sinh \lambda + \sin \lambda)^2},
\]

\[
\lambda = \sqrt{\frac{3(1 - \mu^2)}{t/k}},
\]

respectively. \( \theta \) is the initial subtended angle of the tape spring.

As shown in Fig. 7, the bottom of the sampling arm is connected to the sampler by two screws and the distance from the screws to the cross-section of the shell is \( d \). The sampling arm only bears a reverse force when it moves down.

Therefore, the pushing force \( F \) of the sampling arm can be calculated from the equations below and the maximum pushing force is then obtained by maximizing Eq. (2).

\[
F = \frac{Est^3(k_1 + \mu k_2 \lambda R) \sinh \lambda + \sin \lambda}{12l(1 - \mu^2)}.
\] (2)

The maximum pushing force is calculated as

\[
F_{\text{max}} = M_{\text{a}} \sin \theta / (M_{\text{a}} / 1 > M_{\text{a}} / d),
\]

where \( \ell \) is the effective moment of the arm, i.e., the sampling arm can work well when the pushing force is less than \( F_{\text{max}} \).

4.2 Dynamics analysis

The contact between the sampling head and the lunar regolith can be considered as a rigid-powder coupling system. The sampling head’s circumference unit can be equivalent to a unilateral shock vibration broken system as shown in Fig. 8.

Vibration equation Eq.(3) of the system can be given when the sampling head applies the incentive signal \( F(t) = kA \sin(\omega t) \) to the lunar regolith according to the vibration theory.

\[
m\ddot{x}(t) + c\dot{x}(t) + kx(t) = kA \sin(\omega t)
\] (3)

Let \( c = 2m\omega_n \zeta \), \( \omega_n = \sqrt{k/m} \), \( \beta = \omega/\omega_n \), where \( m \) is the effective mass of the sampler head, \( \omega \) is the frequency of the external excitation, \( \omega_n \) is the resonance frequency of the vibration system; \( A \) is the amplitude of the external excitation; \( \zeta \) is the damping ratio; \( k \) is the equivalent stiffness of the lunar regolith and sampling arm which directly determines the resonant frequency; the steady-state solution can be obtained from the equation above as a harmonic function:

\[
X(t) = \frac{A}{\sqrt{(1 - \beta^2)^2 + (2\zeta \beta)^2}} \sin(\omega t)
\] (4)

Where, \( \beta \) is the ratio of the frequency.

Let \( X = |H(\omega)|A \), then

\[
|H(\omega)| = \frac{1}{\sqrt{(1 - \beta^2)^2 + (2\zeta \beta)^2}}.
\]

When \( \omega = \omega_n \), \( H(\omega) \) reaches the peak value. The maximum value \( H(\omega_n) \) can be obtained from the derivative of \( H(\omega) \) when it is equal to zero. Under such conditions, resonance occurs. As a result, the lunar regolith that comes into contact with the sampling head achieves the maximum amplitude, the internal friction angle value is decreased, the bond force is reduced and fluidity is enhanced. It is an easy moment for sampler penetration.

5. Finite Element Method (FEM) analysis of the sampler

5.1 The statics Finite Element Analysis of the sampler

Based on the design mentioned above, the paper studied the relationship between the shell angle, shell thickness, shell length, shell radius and maximum pushing force, respectively, using the ANSYS tools. Then a mathematical formula was estimated based on the results of the FEM (Finite Element Method) analysis and shell theory.

The deformation of the sampling arm after force has been applied is shown in Fig. 9. The cross-section being clamped by a convex roller and a concave roller is defined as the constraint surface, and the vertical pushing force is imposed at the centre bottom of the sampling head. The main parameters of the shell are shown as follows.

- Young’s modulus \( E = 210 \text{GPa} \),
- Poisson’s ratio \( \mu = 0.3 \),
- density \( \rho = 7850 \text{kg/m}^3 \),
- length \( \rho = 7850 \text{kg/m}^3 \),
- cross-section angle \( \theta = 90^\circ \),
- radius \( R = 15 \text{mm} \).

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Figure 9. The FEM model of sampling arm deformed by thrust

Only one of the parameters, such as angle, thickness, length and radius is changed each time to study their relationships with the maximum pushing force. In Fig.10, figure (a) shows that the shell angle and the maximum pushing force maintain a roughly linear relationship, figure (b) shows that the shell thickness and the maximum pushing force is basically a third-order relationship, figure (c) shows that the shell length is inversely proportional to the maximum pushing force, figure (d) shows that the shell radius and the maximum pushing force also maintain a linear relationship.

Figure 10. Relationship of the sampling arm parameters with the maximum pushing force

Due to the complexity of the equation (2), it is necessary to provide a simple and efficient formula to calculate the maximum pushing force. The mathematical formula of the estimated maximum pushing force is shown in equation (5) as

\[ F_{\text{max}} = k_a R \theta t^3 / l \]  

Where \( R \) is the radius of the sampling arm; \( \theta \) is the angle of the sampling arm; \( t \) is the thickness of the sampling arm; \( l \) is the length of the sampling arm; \( k_a \) is the real constant that can be determined by the physical character of the sampling arm.

5.2 Dynamic Finite Element Analysis of the sampler

We also investigated the dynamic finite element analysis using the ANSYS tool. Firstly, we obtained the vibration analysis results from the sampling head as show in Fig.11 and Table.1.

Figure 11. Vibration modal of sampling head

(a) Angle with the maximum pushing force
(b) Thickness with the maximum pushing force
(c) Length with the maximum pushing force
(d) Radius with the maximum pushing force
| Modal (Order) | 1  | 2  | 3  | 4  | 5  |
|-------------|----|----|----|----|----|
| Frequency (Hz) | 1031 | 1041 | 6465 | 18378 | 20035 |
| Modal (Order) | 6  | 7  | 8  | 9  | 10 |
| Frequency (Hz) | 25237 | 45048 | 51598 | 54505 | 64108 |

Table 1. Vibration modal of sampling head

It can clearly be seen that the resonance frequencies are more than 1 kHz and the contribution of the sampling head’s vibration to the penetration is tiny because the resonant frequency of the soil is typically about several to dozens of Hertz.

Sampler’s structural dynamics modals with different sampling arm length were also established and analysed respectively using ANSYS software. First order (horizontal vibration), second order (axial vibration) and the third order (horizontal vibration) vibration are shown in Fig.12 respectively.

![First order](image1)
![Second order](image2)
![Third order](image3)

Figure 12. Vibration simulation of sampler

Each vibration order with different sampling arm lengths are shown in Fig.13 from the simulation analysis. Through the formula fitting it is clear that there is a power function between vibration frequency and the sampling arm’s length for each vibration order.

![Relationship](image4)

Figure 13. Relationship of sampler length and vibration frequency

6. Experimental results

6.1 Prototype and measure & control system

According to the above design, the experimental prototype of a “coiled snake” type sampler was developed. The main performance parameters are shown in Tab.2.

| Weight | 0.95kg |
|--------|--------|
| Volume | 0.18~0.9×0.9m³ |
| Supplied voltage | 12V |
| Mean power | 3.5W |
| Working path | 0~1m |
| Sampling depth | 0~10cm |
| Sampling volume | 5cm³ |
| Deploying velocity | 3.5mm/s |

Table 2. Main performance parameters of sampler

In addition, the corresponding signal acquisition and control system was developed, the experimental device is shown in Fig.14. The accelerometer which is fixed on the sampling head is used to measure the vibration amplitude of the sampling head. NI USB-6251 data acquisition card and Labwindows CVI software are used in the measure & control system.

![Prototype](image5)

Figure 14. Prototype of sampler and Measure & control system

6.2 Statics experiments of sampler

We selected a type of opening cylindrical shell as the sampling arm whose parameters are as follows: elastic modulus $E = 210$ GPa, Poisson ratio $\mu = 0.3$, density $\rho = 7850$ kg/m³, shell thickness $t = 0.2$ mm, cross section circle of angle $\theta = 90^\circ$, section radius $R = 15$ mm. Experiments show that the arm has about 6N maximum pushing force when it is deployed to 0.8m and this is enough for the sampling head to thrust 10cm deep into the simulated lunar regolith (modal CAS-1, similarly hereinafter), sand or cement and sample quantitatively (5 cm³) as shown in Fig.15. Fig.16 shows the experiments of casting throw samples. The deploying and retracting speed is 3.5mm/s, i.e., it takes about 286s to deploy the arm for 1m. It takes about 2.5s to collect the sample and cast throw the sample respectively. So
the total time for one sampling period is about 10 minutes. In addition, the experiments show that the soil compaction affects the drilling depth and drilling speed to a large extent. A powdered substance such as a lunar soil simulant, cement and sand are used for the experiment and are in a loose state because the lunar soil on the moon is very loose.

Figure 17. Vibration amplitude of sampling head

The length of the sampling arm is changed from 40–80cm with a step increase of 10cm. The vibration frequency of the torsional vibrator embedded in the sampling head is scanned from 1 to 42 Hz at each step length. Thus, we can obtain the relationship between the first-order resonant (horizontal vibration, the maximum amplitude vibration) frequency and the length of the sampling arm, as shown in Fig.18. Experimental results match well with those of the dynamics finite element method simulation. We can also obtain the formula of the first-order resonant frequency as \( f_0 \approx k_1 \cdot \ell^2 \) using a curve fitting method. Where \( f_0 \) is the first-order resonant frequency, \( k_1 \) is the coefficient and \( \ell \) is the length of the sampling arm.

Figure 18. Relationship curve of sampling arm length and the first-order resonance frequency

Experiments of frequency scanning from 1–42 Hz were also performed when the sampling head penetrates into a simulated lunar regolith not more than 10 cm deep. The relationship between the vibration amplitude of the sampling head and the vibration frequency is shown in Fig.19. Thus, it is easy to obtain the relationship between the maximum vibration amplitude and the frequency at various drilling depths as shown in Fig.20. The resonance frequency and drilling depth meet quadratic multinomial relations after fitting.
Next, comparative experiments with and without vibrations have been conducted in a simulated lunar regolith and cement respectively. The drilling speed is about 2.3 and 2 mm/s without vibration and the maximum drilling depth is 65mm and 58mm in a simulated lunar regolith and cement respectively. The drilling speed is about 3.5 and 3.1 mm/s while the torsional vibrator keeps working at a resonant frequency and the maximum drilling depth is 93 mm and 87 mm respectively. Experimental results demonstrate that the vibration method can improve the drilling speed and drilling depth by about 55% and 46%, respectively.

Finally, experiments of casting throw sample have been conducted with and without vibrations. It took 23 and 25 seconds to empty the simulated lunar regolith and cement without vibration but only took 8 and 10 seconds to empty them with vibration. So the vibration method can improve the casting throw efficiency by about 62.5%.

7. Conclusions

This paper presents a novel minitype sampler – Measuring Tape-Like Sampling Arm and Drill (MTS) for the shallow lunar regolith, which has a small shrink volume, long deploying length, light weight and low power consumption. We have carried out a theoretical analysis, simulations and experimental studies, which verify that it is strong enough to thrust into and sample quantitatively in a simulated lunar regolith (model CAS-1) and cement about 10 cm deep. Dynamic analysis and experimental results also indicate the vibration of the torsional vibrator embedded in the inside of the sampling head can highly increase drilling and casting throw efficiency.

In the future, the vibration frequency fuzzy adaptive control algorithm will be studied in order to improve the sampler drilling efficiency. Moreover, the in-situ identification methods based on the vibration signal will be considered in order to extend the function of the sampler.

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