Development of an assistive instrument for lifting motion using driving springs

Hidetsugu TERADA*, Koji MAKINO*, Kazuyoshi ISHIDA* and Masahiro ICHIKAWA*

*Department of Mechatronics, University of Yamanashi
Takeda 4-3-11, Kofu, Yamanashi 400-8511, Japan
E-mail: terada@yamanashi.ac.jp

Received: 30 May 2017; Revised: 7 August 2017; Accepted: 24 August 2017

Abstract
To assist in lifting heavy weight objects from the floor, an assistive instrument that can be worn on the hip and knee joints has been developed. The instrument was driven by coil and flat spiral springs to operate for a long time. At this instrument, the assistive mechanism for the hip joint comprised a plate cam with a “sine-output” oscillating follower and a compression coil spring. The assistive mechanism for the knee joint comprised a non-circular gear, two grooved cams and a flat spiral spring. To eliminate the influence of abduction or adduction and a medial or lateral rotation, two types of freely rotating joints were attached to each of these mechanisms. Based on the analysis of the lifting motion of an object from the floor, it was concluded that the torque applied on the hip and knee joints depends on the individual differences in body height and a body weight, and can be estimated using a two dimensional model. A prototype of the proposed instrument was fabricated and tested. This instrument can be used to reduce the required muscle activities of the hip and knee joints by about 15% compared to while lifting without the instrument when subjected to a force of 98N by an object. Furthermore, the results showed that the assistive instrument exhibits sufficient performance in aiding the lifting motions.

Keywords: Assistive instrument, Lifting, Hip joint, Knee joint, Flexion, Extension, Driving spring

1. Introduction

Recently, in Japan, the number of elderly people has rapidly increased and the aging society causes various problems (Yoshimura, 2009). Moreover, at care centers and nursing homes, the number of nursing aid and nursing care personnel is insufficient. Furthermore, there are numerous cases that require flexible judgment by nursing personnel such as assisting while transferring a person from a wheelchair or while bathing, which are difficult to automate using robots. On the other hand, the nursing personnel often develop hip or knee joint injury during these occupational tasks. Therefore, to compensate for these issues, an assistive instrument for the lower limbs is required.

A variety of power assistive devices, that generally use electrical or hydraulic actuators, have been developed in Japan (Ishii et al., 2004 and Sato et al., 2010). However, these devices are heavy and not suitable to be used for a long duration. Previously, we developed a spring-driven instrument to assist the standing-up motion for elderly people (Terada et al., 2016). This device was designed to be light weight and to operate for a long duration. The behavior of the standing-up motion is similar to the motion of lifting an object from the floor. However, this instrument could not provide enough assistive power to enable a lifting motion. In this study, the lifting motion of a heavy weight object from the floor was analyzed to estimate the generated torques of hip and knee joints. Then, an assistive instrument was designed which can be used in any environmental conditions even without electric supply. Furthermore, to confirm the usefulness of the proposed design, a prototype is tested experimentally to characterize the reduction in the required muscle activity at the hip and knee joints as a result of using the instrument.
2. Two-dimensional model of a lifting motion

The values of mass, centroid position, and moment of inertia of physical each segment are required to simulate a human body motion. Dempster (1955) and Ae, Tang and Yokoi (1992) have proposed a method to estimate inertial properties of various body segments. Using these methods, a two dimensional (2D) model for the lifting motion was proposed to estimate the generated torques on the hip and knee joints. The horizontal position of combined centroid was calculated from the centroid of each human body segments and the centroid of the lifting object, as shown in Fig. 1. The position and the mass of each human body segment and object as well as those of that object lifted are defined as Table 1.

![Fig. 1 Two-dimensional model of a lifting motion.](image)

The moment torque of each body segment is calculated from the horizontal length and mass of each body segment to avoid falling down forward. Then, considering the torque balance of the human body, the horizontal position which is the centroid of the entire body and that of the entire body while lifting object are defined.

Table 1 Definition of the centroid position and mass of each body segment shown in Fig. 1.

| Terms                                                                 | Symbols |
|----------------------------------------------------------------------|---------|
| Horizontal position of the centroid of each body segment (head, upper arm, torso, forearm, hand, thigh, shank, foot) | $x_i$   ( $i = 1, 2, \ldots, 8$ ) |
| Mass of each body segment (head, upper arm, torso, forearm, hand, thigh, shank, foot) | $M_i$ ( $i = 1, 2, \ldots, 8$ ) |
| Mass of the lifted object                                            | $m_L$   |
| Mass of the subject                                                  | $M_S$   |
| Horizontal position of the centroid of the lifted object             | $x_m$   |
| Horizontal position of the centroid of the entire body               | $x_G$   |
| Horizontal position of the centroid of the entire body with the lifted object | $x_F$ |
| Horizontal position of the center of the hip joint                  | $x_H$   |
| Horizontal position of the center of the knee joint                 | $x_K$   |
| Acceleration of gravity                                              | $G$     |

Using proposed definitions, the centroid of the entire body are defined as Eq. (1).

\[
x_G \cdot M_S = \sum (x_i \cdot M_i), \quad (i = 1, 2, \ldots, 8)
\]
Thus, the torque balance equation can be defined as in Eq. (2).

\[ x_f \cdot (M_S + m_L) = x_G \cdot M_S + x_m \cdot m_L \]  \hspace{1cm} (2)

Therefore, the generated torques of a hip joint and a knee joint, \( T_H \) and \( T_K \), are defined as in Eq. (3) and Eq. (4), respectively.

\[ T_H = (x_{H} - x_F) \cdot M_S \cdot G \]  \hspace{1cm} (3)

\[ T_K = (x_{K} - x_F) \cdot M_S \cdot G \]  \hspace{1cm} (4)

### 3. Estimation of the torque during lifting

Using the proposed calculation method, the generated torques for the hip and knee joints are estimated. For this estimation, the horizontal positions of the centroid of each body segment are measured using a 2D image-processing system. Figure 2 and 3 show the relationship between the flexion of the hip and knee joints and the estimated torque. In these figures, the sign of the torque indicates the direction of rotation wherein a positive value indicates extension and a negative value indicates flexion. The lifting motion involves various posture changes, as shown in these figures. Thus, based on the posture and circumstance of a lifted object, the lifting motion is classified into Phase 0 to Phase 5. On the hip joint, the generated torque in the flexion direction decreases rapidly during Phase 2. In Phase 3, that torque is increased for short periods of time due to the balance of the hip joint pose and the position of lifted weight. This is caused by the individual differences including the variation of the position of a lifted object and varying body-segment length. In Phase 5, a steady torque in the flexion direction was caused by the weight of the lifted object. In contrast, the
Torque on the knee joint decreases in the fully extended position and weight of the lifted object generates a torque on the knee joint in the extension direction after the full extension state. Therefore, it is necessary to include a motion limiter to resist the over-extension motion. Table 2 shows the estimated torque on the hip and knee joints of five subjects while lifting a 98 N object. The average torques, which depend on individual differences in body height and weight, are 217.5N-m on the hip joint and 151.7N-m on the knee joint.

Table 2 Maximum flexion angles and the maximum torques of the hip and knee joints for each subject as well as the average values and the standard deviations.

| Subject | Body weight (N) | Maximum flexion angle of the hip joint (degrees) | Magnitude of the maximum torque on the hip joint (N-m) | Maximum flexion angle of the knee joint (degrees) | Magnitude of the maximum torque on the knee joint (N-m) |
|---------|----------------|-----------------------------------------------|----------------------------------------------------|-----------------------------------------------|----------------------------------------------------|
| A       | 666.4          | 107.6                                         | 178.9                                              | 155.5                                         | 163.8                                              |
| B       | 686.0          | 90.0                                          | 257.0                                              | 180.0                                         | 238.6                                              |
| C       | 588.0          | 106.2                                         | 261.7                                              | 150.0                                         | 133.0                                              |
| D       | 666.4          | 145.7                                         | 267.5                                              | 179.4                                         | 107.2                                              |
| E       | 715.4          | 146.6                                         | 122.2                                              | 162.6                                         | 115.8                                              |
| Average | 664.4          | 119.2                                         | 217.5                                              | 165.5                                         | 151.7                                              |
| Standard deviation | 42.1          | 22.8                                          | 57.6                                               | 12.3                                          | 47.6                                               |
4. Structure of an assisting instrument

To assist under a range of situations, including those at the bed side and the bath side in a care center or hospital, an assistive instrument that uses a flat spiral spring and a coil spring as the power unit and does not include electric devices is proposed, as shown in Fig. 4. To avoid applying pressure on large blood vessels near the surface of the skin, the braces are arranged as shown. The assistive mechanism for the hip joint has an oscillating follower type cam to restrict the range in which assistance is applied. Then, the assistive mechanism for the knee joint a non-circular gear and grooved cams to provide a rotating motion with sliding backward translation; this complex motion is known as “roll-back motion”. The total weight of the instrument is less than 40.0N, and it can easily to worn for a long time. Then, to eliminate the influences of abduction or adduction and medial or lateral rotation, freely rotating joints are attached on the assistive systems of the hip joint. Furthermore, to avoid difficulty while walking, the range of joint angles at which assistance is applied to the hip joint is restricted to 30–150° and the range of joint angles at which assistance is applied to the knee joint is restricted to 15–180°.

5. Design of the assistive mechanism for the hip joint

A conventional oscillating follower type cam with a cantilever beam that has a spring characteristics provides insufficient assistive power (Terada et al., 2016). Moreover, a self-locking motion often occurs with a linear-motion follower-type cam. Therefore, to improve the assistive power and to avoid the self-locking motion caused by friction, the assistive mechanism of the hip joint comprises a plate cam and an oscillating cam follower, which has a follower lever and follower rollers that provides a linear sliding output motion, as shown in Fig. 5; this type of cam is called as the “sine-output-type oscillating cam” (Makino, 1976). When the subject flexes the hip joint, the roller on the oscillating follower lever is pushed and the corresponding lever is swung. Then, the opposite-side roller pushes the piston. This piston, which has a coil spring, slides horizontally and accumulates a flexion force. To avoid the self-locking motion, the vertical motion of the spring is eliminated as the offset motion. Furthermore, to adjust the angle at which the force begins to accumulate, an adjustment screw is attached to the piston. Then, upon extension of
the hip joint, the spring releases the accumulated force to generate an assistive torque. This mechanism is firmly attached to the leg using a hip brace and a thigh brace. To eliminate the influence of abduction or adduction and of medial or lateral rotation, the two types of freely rotating joints with tolerance angle as ±10° are integrated into this mechanism. The pressure angle of the cam follower is less than 70°, and the coefficient of friction is less than 0.1. Therefore, the friction loss is negligible (Makino, 1978).

To design the shape of the plate cam and the stroke of the piston, it is necessary to calculate the inverse kinematics using vector analysis (Terada et al., 2013). The vector geometry of a plate cam for the assistive mechanism for the hip joint is defined as shown in Fig. 6 and each vector is defined in Table 3. To avoid difficulty of hip joint flexion while walking, the angle $\theta$, at which an assistive force is applied, was restricted from 30–150°. To exhibit smooth rotation, the “modified sine motion curve” (Makino, 1976) is applied to the acceleration curve for the cam. The calculated parameters are defined as the non-dimensional displacement $S$; the non-dimensional time $T$; the start angle $\theta_s$, the end angle $\theta_e$ of the cam motion; the maximum stroke $r_{max}$ and the minimum stroke $r_{min}$ of the plate cam. The arbitrary motion stroke $r$ and the rotation angle $\theta$ are defined as shown in Eq. (5) and Eq. (6).

\[
S = \frac{r - r_{min}}{r_{max} - r_{min}} \quad (5)
\]

\[
T = \frac{\theta - \theta_s}{\theta_e - \theta_s} \quad (6)
\]

The non-dimensional displacement can be calculated as shown in Eqs. (7)–(9) using the acceleration curve of the “modified-sine acceleration curve”. The connecting conditions of these equations are different from the conventional curve.

\[
S = \frac{\pi}{\pi + 4} \left( T - \frac{1}{4\pi} \sin 4\pi T \right) , \quad (0 \leq T \leq 1/8) \quad (7)
\]
Table 3  Definitions of vectors used to generate the shape of the plate cam and the piston motion.

| Terms                                      | Symbol | Length | Direction |
|--------------------------------------------|--------|--------|-----------|
| Constant vector between the centers of the plate cam and oscillating follower lever | C₁     | c₁     | θ₁        |
| Constant vector between the center of the oscillating follower lever and the piston origin | Cₚ     | cₚ     | π/2       |
| Oscillating vector of the plate cam side follower roller | B₂     | b₂     | θ₂        |
| Rotation vector of the follower roller at its origin | P₃     | p₃     | θ₃        |
| Relative motion vector of the follower roller at its center | P₄     | p₄     | θ₄        |
| Normal vector of the a follower roller at its center | P₄     | p₄ₜ     | φ         |
| Envelope vector of the follower roller     | B₅     | r₅     | θ₅        |
| Plate cam profile vector                   | P₆     | p₆     | θ₆        |
| Oscillation vector of the piston side follower roller | B₆     | b₆     | θ₆        |
| Linear-motion vector of the piston side    | Lₐ     | lₐ     | 0         |
| Linear-motion vector of the offset motion  | Lₐ     | lₐ     | -π/2      |

\[
S = \frac{9}{4(\pi + 4)} \left\{ 1 - \cos \left( \frac{8T - 1}{6} \right) \right\} + \frac{\pi}{\pi + 4} \left( T - \frac{1}{8} \right) + \frac{1}{\pi + 4} \left( \frac{8}{4\pi} - \frac{1}{4\pi} \right), \quad (1/8 \leq T \leq 7/8) \tag{8}
\]

\[
S = \frac{1}{4(\pi + 4)} \left\{ \cos \left( \frac{8T - 7}{2} \pi \right) - 1 \right\} + \frac{\pi}{\pi + 4} \left( T - \frac{7}{8} \right) + \frac{1}{\pi + 4} \left( \frac{\pi + 31}{32} + \frac{1}{4\pi} \right), \quad (7/8 \leq T \leq 1) \tag{9}
\]

Therefore, the motion stroke of the follower roller, which is the same as the stroke of the piston motion, can be calculated from these equations. The profile vectors of the plate cam are calculated using Eqs. (10)–(15) using a well-accepted cam profile calculation method, i.e., complex vector analysis (Terada, 2004).

\[
P₃ = B₂ + C₁ = p₃ \cdot e^{iθ₁} = b₂ \cdot e^{iθ₂} + c₁ \cdot e^{iθ₁} \tag{10}
\]
The plate cam profile at the center of the cam follower was calculated from the relative motion. Therefore, the motion vector rotates in the inverse direction. Thus, the plate cam profile is the envelope locus of a motion vector on the center of the cam follower, and the angle of a normal line $\phi$, is calculated geometrically from the differential value of the relative motion vector on the center of the cam follower. Furthermore, the rotating angle of the piston side follower is calculated geometrically as $\alpha$.

Next, the linear motion of a piston and the generated force $F_Y$ are calculated as shown in Eq. (16) and Eq. (17), using the spring constant $k_{mx}$.

\[
P_3 = B_2 + C_1 = p_3 \cdot e^{i\theta_2} = b_2 \cdot e^{i\theta_2} + c_1 \cdot e^{i\theta_1}
\]

\[
P_4 = p_3 \cdot e^{i(-\theta)} = p_4 \cdot e^{i\theta_1}
\]

\[
\dot{P}_4 = p_{4d} \cdot e^{i\phi}
\]

\[
B_5 = r_1 \cdot e^{i\left(\phi - \frac{\pi}{2}\right)}
\]

\[
P_6 = B_5 + P_4 = p_6 \cdot e^{i\theta_1}
\]

\[
\theta_A = \theta_2 + \alpha
\]

The plate cam profile at the center of the cam follower was calculated from the relative motion. Therefore, the motion vector rotates in the inverse direction. Thus, the plate cam profile is the envelope locus of a motion vector on the center of the cam follower, and the angle of a normal line $\phi$, is calculated geometrically from the differential value of the relative motion vector on the center of the cam follower. Furthermore, the rotating angle of the piston side follower is calculated geometrically as $\alpha$.

Next, the linear motion of a piston and the generated force $F_Y$ are calculated as shown in Eq. (16) and Eq. (17), using the spring constant $k_{mx}$.

\[
B_A + L_C = L_Y + C_p = b_A e^{i\theta_A} - j l_C = l_Y + j c_p
\]

\[
F_Y = k_{mx} \cdot l_Y
\]

Using these equations, the relationship among the flexion angle, the plate cam stroke and the acceleration curve were calculated, as shown in Fig. 7. Here, the assistive angle is restricted to the range of 30–150° to avoid difficulty while walking.

In addition, the relationship between the flexion angle, torque, and compression force from the coil spring are calculated, as shown in Fig. 8. Herein, the maximum assistive torque is less than 15% of the user-generated torque because the user should be able to resist the assistive motion using muscular strength, in case a problem, such as a fall or a loss of balance, arises.
6. Design of the assistive mechanism for the knee joint

In general, the motion of the knee joint is regarded as a rotation with backward translation wherein the bones come in contact with each other, as shown in Fig. 9 because the 2D shape of the condyle at the contact point has non-uniform radius, which are shown as I, II and III in Fig. 9. This motion at the center of bone is called “roll-back” motion and is defined as the motion of an imaginary rotation center (Smith, 2003).

To facilitate the rotation of the knee joint with the “roll-back” motion, the fundamental structure of the assistive mechanism for the knee joint comprises a plate cam, which has a non-circular gear and two grooved cams, and a flat spiral spring, as shown in Fig. 10 (Terada et al., 2016). In this mechanism, the rotational force was accumulated in the spiral spring during knee flexion, and that released when the joint is fully extended. In addition, based on the previously reported knee motion of Japanese people (Terada et al., 2012), the cam profiles and the center locus of the non-circular gear are generated according to the “asymmetrical-modified-trapezoid acceleration curve”. The theoretical relationship between the rotation angles of the knee joint, the roll-back motion displacement, and the assistive torque of the flat spiral spring are shown in Fig. 11. The maximum angle of the knee rotation at which the roll-back motion occurs is 120°, and beyond this angle, the knee joint rotates without roll-back motion. The assistive torque, which is about 23% of the user-generated torque, increases until the angle of the knee rotation is at 180°.

Fig. 8  Relationship between the flexion angle, the torque, and the compression force of the coil spring. The compression force and the applied torque are proportional to the plate cam stroke. Then, the maximum applied torque is limited to 15% of the user-generated torque.

Fig. 9  Roll-back motion of the knee joint. The knee flexion has a large flexion with a large translation to backward and external and internal rotations. During lifting, the external and internal rotations on a knee joint are absorbed using the elastic deformation of the braces.

[DOI: 10.1299/jamsm.2017jamsm0037] © 2017 The Japan Society of Mechanical Engineers
7. Verification of the prototype instrument

The merit of the instrument for lifting motion have been verified using a prototype of the instrument, as shown in Fig. 12. The instrument is attached to a healthy person (25 age, men) and the cam profiles are optimized for the subject (Terada, 2012). Because the influence of fatigue is significant, the test are repeated only once per subject. The coil spring constant on the assistive mechanism of the hip joint is 39.8N/mm and the flat spiral spring constant on the assistive mechanism of the knee joint is 0.73N-m/revolution. To avoid the risk of slipped disk and lumbago, the weight used to test the lifting motion is limited to 196N. Using the integrated myogenic potential evaluated as the normalized maximum voluntary contraction (%MVC) as the muscle activities, it is confirmed that the muscle activity of each joint
is reduced by about 15% compared with the same motion of lifting with a 98N object without the assistive instrument, as shown in Fig. 13 and Table 4. Based on these results, the proposed assistive instrument reduces the peak torque on the joints and shifts the timing of the peak torque toward the extension phase of the motion. However, the generated torques on the hip and knee joints are not proportional to the lifting weight, because the load distribution on the joints varies. Therefore, the reduction ratio of the muscle activity is insufficient for lifting 196N. Thus, the spring constant
and an assistive timing must be adjusted. Furthermore, we will investigate the effects of fatigue, analyze the repeatability, and assess the influence of individual differences.

Table 4  Muscle activity reduction ratio in each joint with the assistive instrument compared with the same motion executed without the assistive instrument.

| Mass of the lifted object (N) | Reduction ratio of the muscle activity (%) |
|------------------------------|--------------------------------------------|
| Hip joint                    | Knee joint                                 |
| 98                           | 15.0                                      |
| 196                          | 4.8                                       |

8. Conclusions

To assist a lifting motion, an assistive instrument using driving springs has been developed. Considering the motion analysis of the lifting motion, it is confirmed that this instrument can reduce the required muscle activity in the hip and knee joints by about 15% when lifting a 98N object. Based on the results, the spring constant of the assistive mechanisms have to be adjusted for each lifted object. Furthermore, the assistive timing is fixed for the lifting motion from the floor in this prototype and therefore, this prototype is insufficient for other arbitrary lifting motions. In future work, we will develop a mechanism to adjust the spring constant and the assistive timing. Furthermore, we will investigate the effects of fatigue, assess the repeatability, and analyze the influence of individual differences among users.

References

Ae, M., Tang H. and Yokoi, T., Estimation of inertia properties of the body segments in Japanese athletes, Biomechanisms, vol.11, (1992), pp23–33.

Dempster, W. T., Space Requirements of the Seated Operator, WADC Technical Report, (1955), pp55–159.

Ishii, M., Yamamoto. K., Hyodo, K., Matsuo T. and Takahashi, K., Development of power assist suit for assisting nurse labor, improvement of sensing system and mechanism, Proc. of the Welfare Engineering Symposium 2004 , (2004), pp123–126 (in Japanese).

Makino, H. and Takano, M., Machine kinematics, Corona Publishing Press, (1978), pp.112–113(in Japanese),

Makino, H., Automatic assembly machine kinematics, Nikkan-Kogyo Shimbun Press, (1976), pp.28-29, 160-161, 180-188, 212-216(in Japanese).

Sato, H., Kawabata, T., Tanaka, F. and Sankai, Y., Transferring-care assistance with robot suit HAL, Transactions of the Japan Society of Mechanical Engineers- C, Vol. 76, No.762, (2010), pp227-235 (in Japanese).

Smith, P. N., Refshauge, K. M. and Scarvell, J. M., Development of the concepts of knee kinematics, Archives of Physical Medicine and Rehabilitation, Vol.84, (2003), pp 1895–1901.

Terada, H., Kobayashi, M. and Imase, K., Development of a trochoidal gear reducer with slipping rollers type torque limiter, New Advances in Mechanisms, Transmissions and Applications, Vol. 17, Springer International Publishing, (2013), pp.49–56.

Terada, H., Zhu, Y., Suzuki, M., Cheng, C. and Takahashi, R., Developments of a knee motion assist mechanism for wearable robot with a non-circular gear and grooved cams, Mechanisms, Transmissions and Applications, Vol.3, Springer International Publishing, (2012), pp. 69–76.

Terada, H., Ishisa, K., Chiba, H. and Imase, K., Fundamental Analysis of a liniear type trochoidal gear (3rd report), kinematic analysis of an internal gear type trochoidal corner curve rack, Journal of the Japan Society for Precision Engineering, Vol. 70, No.2, (2004), pp.209–213(in Japanese).

Terada, H., Makino, K., Ishida, K. and Ichikawa, M., Development of an assisting instrument of standing-up motion using driving springs for elderly persons, Mechanisms and Machine Science New Advances in Mechanisms, Mechanical Transmissions and Robotics, Vol. 46, Springer International Publishing, (2016), pp.417–425.

Yoshimura, N., Progress of research in osteoarthritis. Epidemiology of osteoarthritis in Japanese population -The ROAD study, Clinical Calcium, Vol.19, No.11 (2009), pp1572–1577 (in Japanese).