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Study on Flexible Spacesuit Joints
Performed by Variable Shape Pneumatic Structures

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Chapter 1

General Introduction

Human activity field has been expanding significantly in the past several decades in terms of geographical and physical ranges. Environment of those fields is far different from our ordinary ground. For example, after human invented the airplane to fly the sky in 1903, we have utilized the sky at high altitude with cold temperature and thin air. Our frontiers extended out of Earth after the first human spaceflight in 1961. Astronauts stay for more than several months on the International Space Station in low earth orbit. They carry out extravehicular activities in outer space, which is harsh environment. Polar regions in low temperatures of 50 degrees below zero are another fields where human explored, and researchers stay there and enroll in activities on a regular basis. In addition, industrial activities have been utilizing various kind of plants handling with very high pressure, temperature and various chemicals. Human also started utilizing nuclear resources for power generation. As handling condition and material vary, hazardous risk simultaneously increase. There are substantial needs for dealing with chemically polluted environment in industry. Fire fighters should carry out rescue operations in severe heated and complex environment. We also need to immediately control biological hazard once new infectious viruses and bacteria are discovered. Chemical and biological weapons also increase the risk of facing hazardous environment. We also need to deal with radioactive substances for accidents of nuclear power plants.

On the other hand, today's society is anticipated to accept more diversified groups associated with development of social welfare service. Progress in medical treatment has allowed human to have the greater longevity with healthy conditions. To improve their quality of life, providing opportunity for public participation to elderly persons is gaining attention. Working opportunity for disabled persons is also expected more. In the meantime, social burden of nursery care is growing. Working environment for workers should be improved by reducing heavy physical labor tasks.

For supporting extension of human frontier, mitigating risks of hazardous environment and also providing more opportunities for diversified groups, functional garments take a great role for our further social activity prospects, which attracts strong interest.

1.1 Functions of garments

Garment is very essential elements for human living as well as food and habitation. Wearing garment distinguishes human from any other animal, which is a proof of intelligence for living in various environments. Functions of garments is divided into three parts: cultural indication, function addition and environment control as shown in Figure 1.1.

The first aspect is cultural indication. Although this thesis does not describe the point in detail, the aspect is very significant role taken by garment. Garments have been developed in individual group over the world from ancient times depending on the difference of ethnicity, gender, social status, or religion by means of color, design and style.

Garment has another role to shelter against surrounding environment such as temperature, pressure and atmosphere. Fire fighter suits, polar parkas and clothes for refrigerated warehouse protected workers from very high and low temperature conditions. An example to protect from high and low pressure conditions is atmospheric diving suits. Flight suits worn by high-altitude pilots is a garment to protect pilot from low pressure environment. And, as is described in detail later, extravehicular spacesuit is a full pressurized suit to protect vacuum environment. To protect dangerous goods such as radioactive, flammable, explosive, corrosive, oxidizing, asphyxiating, biohazardous, toxic, pathogenic, or allergenic substances, special clothes are prepared for each targeted dangerous hazard. Cleanroom suit is another example to shield inner and outer environment across the garment to protect contamination.

Garment has another function which can embed a variety of wearable devices on garment. For instance, wearable communication tools are beneficial for instant and continuous communications. Among those devices, physical assisting for limb movement is one of potential function for wide variety of field such as assisting locomotion and work operation of elderly or disabled persons. There are also significant potential needs for assisting heavy physical labor for all industrial fields.

Among function addition, assisting limb movement is getting concerned. There are a great variety of systems for assisting human limb movement. In some cases, human gets resistance from a garment to move his or her body due to mass of the garment or pressure difference across the garment. In that case, actual strength, which human can generate, may be degraded from the strength which can be generated by nude body (nude strength). Or, there are needs to enhance nude strength or support to keep long-time work as shown in Figure 1.2. To assist the degraded strength and to enhance nude strength, physical assisting system has been developed and considered to be widely used. The physical assisting systems is categorized into active type and passive type.

Active type of physical assisting system is powered by external energy resources such as electric or pneumatic energy. In case of powered assisting systems, by monitoring body action or signal from human muscle, the assisting system responds by amplifying strength using actuator devices. One typical example which is called powered exoskeleton consists of exoskeleton-like framework worn by a person and supply power of activation-energy of limb movement. The powered assisting systems have been widely developed for powered orthosis, assisting nursing care, assisting work ac-

tivities in industry and strengthening soldier [9]. Electric motors are often used for actuator devices because of their high power generation. Pneumatic actuator is also another candidate for actuating device because of its simplicity and light-weight. In addition, actuator materials are recently widely researched and getting attentions. The potential for practical applications is emerging in shape memory alloy, Ionic Polymer-Metal Composite (IPMC) [10] and dielectric elastomer [11, 12].

On the other hand, passive assisting system has function to compensate the degraded amount of strength due to garment mass or pressure difference by structural elaboration without using powered device. These structures are basically categorized to hard and soft types as shown in Figure 1.3. A typical structure employed for hard-type structures is bearing mechanism to improve flexibility of limb movement. As for soft type of structure, for example, devices have convoluted bellows which are composed of repeated corrugated shells.

Active assisting device, although it would generate high power, often has complex system with a great number of parts, which require maintenance and regular replacement. Total mass tends to be heavy due to its complex system. On the contrary, passive assisting systems have benefits for lighter and simpler systems. Improved passive assisting systems would be able to be attached with active assisting device to enhance strength. Therefore, developing passive assisting systems can be regarded as a fundamental development for all physical assisting device.

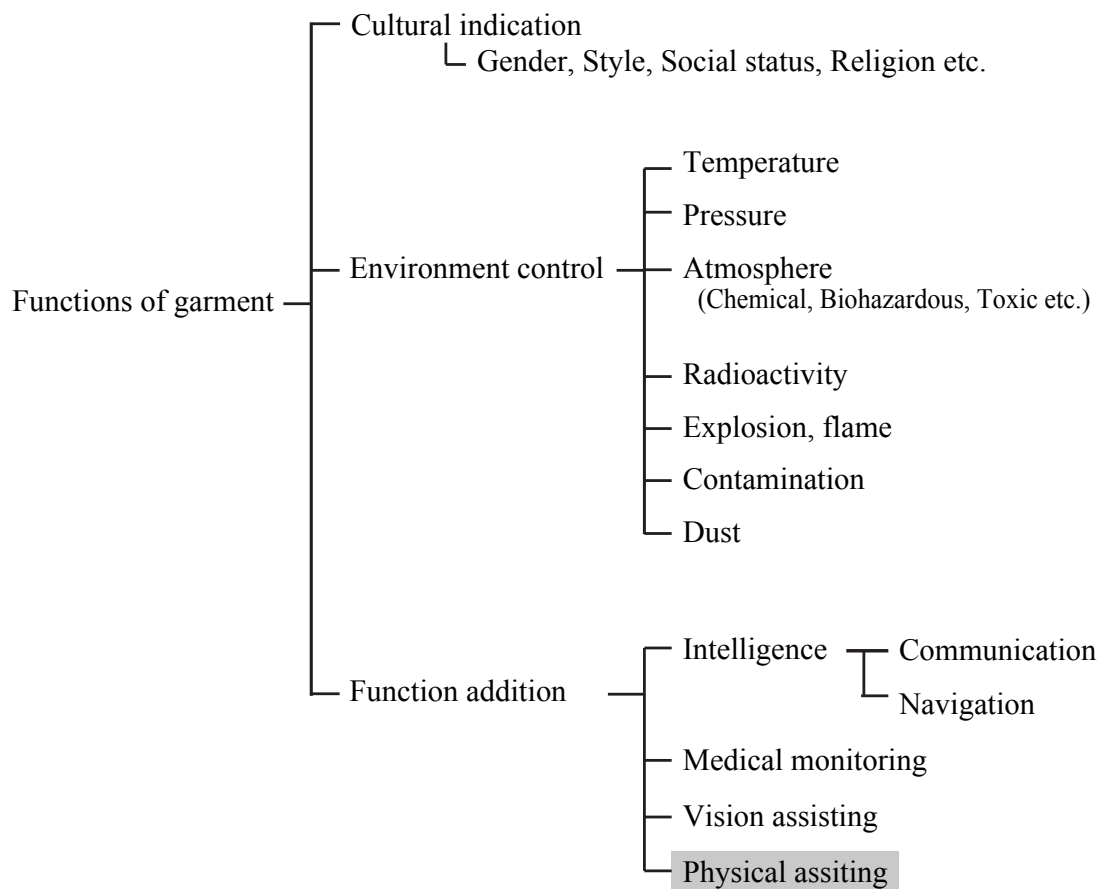


Figure 1.1. Functions of garments.

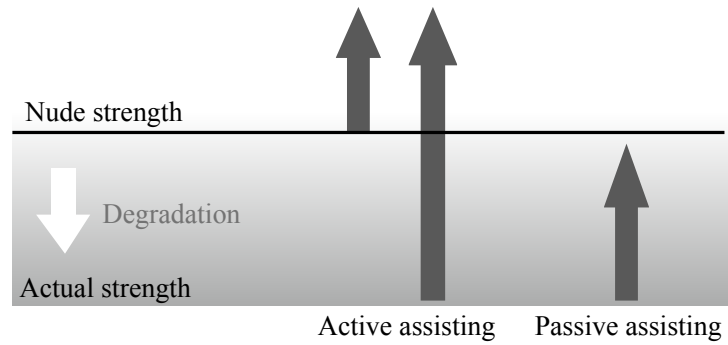


Figure 1.2. Definition of physical assisting.

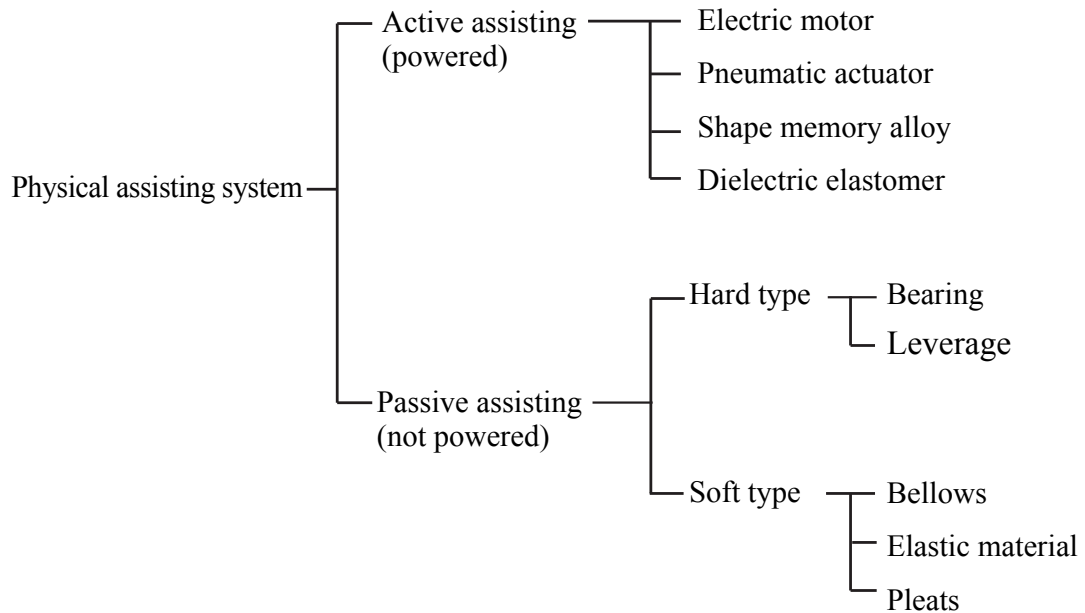


Figure 1.3. Active and passive physical assisting systems.

1.2 Insufficient mobility of spacesuit joint

Extravehicular spacesuit is one kind of garments which tackles with very harsh environment to protect human survival, such as vacuum, intense thermal condition, high-level solar radiation, protection from hazards by micrometeoroids and dust particles on the lunar surface, which is abrasive to the garment and degrade mechanical parts' friction. Also, it provides a wide variety of functions to astronauts from heat, atmosphere and pressure control to communication and navigations. In addition, providing mobility of limb movement is one of essential functions.

A spacesuit in outer space is subject to inflation caused by pressure difference across the garment, which results in degradation of joint mobility of spacesuits. In fact, general and finger fatigue during

extravehicular activities due to low flexibility of joints have been repeatedly marked by US astronauts in Gemini, Apollo and Shuttle Space Programs [13]. A number of measurement of joint flexibility of spacesuits has been reported [14, 15, 16, 17, 7, 18], which indicates that joint mobility is one of critical issues for developing spacesuits, and the level of mobility is not sufficient yet. Improving flexibility of spacesuit joint has been big challenge for several decades.

In addition, the poor flexibility does not only cause fatigue to astronauts but also reduces operation efficiency due to long preparation time prior to an every Extravehicular Activity (EVA) [16]. To compensate the low flexibility, the most of extravehicular spacesuits is operated under much lower pressure than spacecrafts. For example, current US Extravehicular Mobility Unit (EMU) is operated under 29.6 kPa in contrast to 101.3 kPa of cabin pressure of the International Space Station. Lower operating pressure brought duty of prebreathing and control risk of decompression sickness. Astronauts must “prebreathe” pure oxygen to help purge nitrogen from their bloodstream before entering such a low pressure pure oxygen environment to prevent decompression sickness. Prebreathing takes several hours to a half day prior to every extravehicular activity [19]. Consuming time for prebreathe shortens crew time, and it directly reduces operation time for other missions. In addition, risk controls of decompression sickness limits EVA schedule and severely limit capable astronauts who can carry out extravehicular activities.

1.3 Development of spacesuit joint

Developing enhancement of joint mobility is an essential task for future space missions. In present, human space activities are mainly conducted in the International Space Station. A large number of experiments in wide variety of field has been carried out in micro-gravity environment and brought many valuable achievements. The next step will be planetary explorations such as lunar and Martian surface. To conduct planetary surface missions, spacesuit system will especially take more significant roles for astronaut survival and mission accomplishment than other previous programs. Compared to operations on the Earth orbit, operations on planetary surfaces require severe challenge, such as weight reduction, locomotion on uneven terrain and dust reduction. Among the requirements, high flexibility of joints and dexterous manipulation of hands will be highly demanded in order to grip tools and locomotion for survey for long-duration operations with minimum fatigue.

A new generation spacesuit will need to be capable under the pressure which does not have decompression risk without prebreathing. The targeted pressure is over 55.2 kPa. Performing high flexibility of spacesuit joint under such a higher operating pressure requires innovative development of new joint structure design.

To solve the problems of low joint flexibility, there are two possible approaches, which are incorporating active or passive physical assisting system as is mentioned in the last section. Equipping active assisting system on spacesuit probably strengthen joint flexibility. However, in terms of limitation due to power supply, reliability and weight increase, active assisting approach does not yet have sense of reality. On the other hand, developing passive assisting system is embedded into a uniform structure of joint part. It is simple, reliable and lightweight. Soft type is superior to hard

type in terms of weight, stowability and comfort.

Although many types of joint designs have been developed, little literature about comprehensive comparison or review on spacesuit joint structure or passive physical assisting structure is found. The thesis gained motivation to analyze or classify existing mechanical concepts of joint structure in order to derive hint to develop new designs.

1.4 Objectives and organization

The primary aim of the thesis is to advance the current understanding of mechanical concept of passive assisting structure to compensate degraded arm strength of astronaut wearing spacesuit. The two specific objectives are to:

- 1) Analyze and classify existing mechanical concepts of passive assisting structures and derive mechanical issues of insufficient flexibility.
- 2) Develop a new design of passive assisting structures by eliminating the mechanical issues of existing designs.

This thesis is composed of five chapters as shown in Figure 1.4. Chapter 1 describes introduction of this thesis. The topic begins with the rolls of functional garment surrounding our society. After categorization to active and passive physical assisting function, The need for improvement of passive assisting systems especially for spacesuits is shown. Current status of flexibility and related issues are described. Finally, this chapter describes objectives and organization of this thesis.

Chapter 2 explains the connection between the general objectives and the actual problems in this thesis. The problem of low flexibility for spacesuit joints is simplified to a problem of flexibility of a pneumatic membrane tube. After analyzing existing designs, this chapter proposes new design concepts of variable shape pneumatic membrane tubes.

Chapter 3 describes the first proposed design called Membrane-Beam Folding Structure (MBFS). The structure is composed of polygonal frames attached on a membrane cylinder regularly. The frames constrain the membrane cylinder to reduce axial tensile stress due to inner pressure on membrane. That allows extension and contraction with smaller external force. Enclosed volume is assumed to be maintained constant by extension of outer part and contraction of inner part at same ratio. Experiment is conducted to investigate bending property of the MBFS. After the experimental investigation, estimation is conducted through simplified analytical modeling. The chapter estimates feasibility of the design.

Chapter 4 describes the second proposed design called Isotensoid Excess Membrane Structure (IEMS). The proposed joint is intended to sustain excess material of garment by structural elaboration. The excess skin allows deformation of structure. Moreover, the proposed design has the axisymmetric shape, which takes advantage of maintaining volume. First finite element analysis is conducted to estimate equilibrium shape of isotensoid. Then, experiment to investigate flexion property and related properties is conducted.

Finally, Chapter 5 summarizes conclusions and recommendations from Chapter 2 to 4.

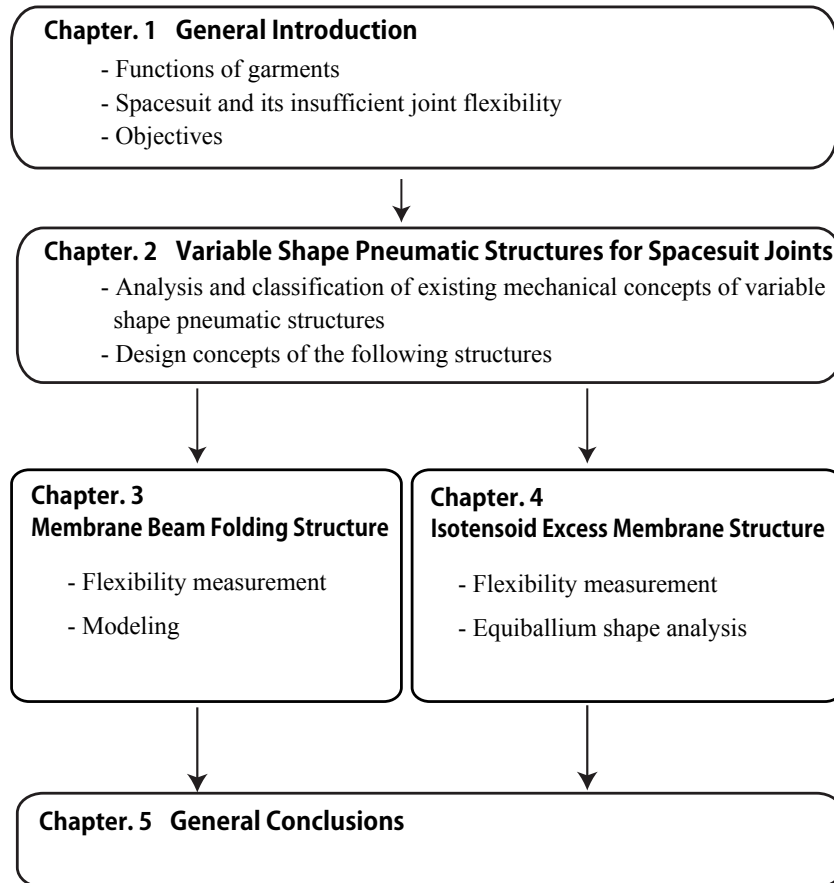


Figure 1.4. Organization of this thesis.

Chapter 2

Variable Shape Pneumatic Structures for Spacesuit Joints

2.1 Introduction

This chapter describes how the thesis approaches the aim, which is to acquire knowledge of passive physical-assisting device and enhance the potential of the technology. Spacesuit joint is one of the typical applications for passive physical-assisting device. Development of joint flexibility is highly required. However, spacesuit joints are parts of the spacesuit system and have multiple functions not only for flexibility. To tackle with the development of embedding passive physical-assisting function into spacesuit joint, the thesis establishes the simplified physical task to focus on mechanical problems. Namely, the thesis considers design of closed cylindrical membrane structure pressurized inside, which can bend in one plane with a minimal applied force as shown in Figure 2.1.

A spacesuit can be regarded as a closed pneumatic structure. A pneumatic structure is a enclosed structure stabilized by pressurized air inside. Main material of joint part of current spacesuit is fabric material, this study deals with the structure made of so called membrane material. Membrane include fabric, rubber sheet and polymer sheet such as nylon or polyethylene, which is hardly sustain compressive stress and occur out-of-plane deformation instantly under compression stress. Note that the membrane used in this thesis has enough strength to sustain tensile stress under pressurization. In addition, providing flexibility for single axis joints, such as knees, elbows and fingers, is focused. The shape is regarded as a cylinder since extended configuration of a joint and connected limbs are described as a cylinder. The type of joint only allows flexion and extension in one plane. High flexibility of spacesuit means that less force is required for flexion and extension of joints.

This chapter is organized into three parts. In the first part, considerations on constituting principle of flexible pneumatic membrane cylinder are given. Then, in the second part, existing designs of flexible pneumatic membrane cylinder are analyzed and classified, and key problems of existing approaches are described. Finally, in the third part, two new designs are proposed with reviewing related literature.

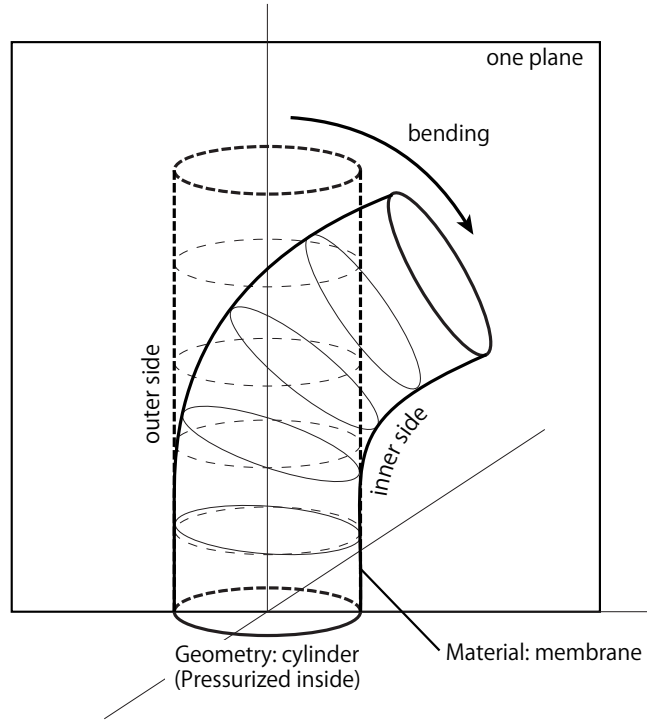


Figure 2.1. Illustration of the concerned problem.

2.2 Consideration on a flexible pneumatic membrane cylinder

Our study is to develop a flexible pneumatic membrane cylinder which performs the bending deformation with minimal applied force by designing its shape, material, structure.

In this problem, total potential energy of the system consists of hydrostatic pressure potential of inner gas, elastic strain energy of materials of the structure and gravitational potential energy of material. Amount of total potential energy in the system should be maintained with minimal change. Since gravitational potential energy does not change during flexion, the term can be neglected when considering flexibility.

The change of volume during deformation requires external mechanical work of gas compression resulting in increase of pressure potential of inner gas. To maintain internal volume constant, area of cross-section and length of neutral axis of a cylindrical structure should not be changed. Therefore, as shown in Figure 2.2, inner side region of cylindrical envelope must be contracted as well as outer side region should be extended at equivalent ratio. These contraction and extension of regions enable bending deformation while keeping inner volume constant. The contraction and extension movement should be performed with minimal applied force.

Thereby, to design the structure, the next two basic requirements are essential to be satisfied.

- (1) To maintain volume with minimal change
- (2) To elongate and contract outer and inner areas of bending with minimal force

The first requirement corresponds to maintaining pressure potential. The second requirement corresponds to strain energy of material.

The contraction and extension of pneumatic structure are not easily carried out without applying large external force. Membrane of a cylindrical structure is tensioned under pressurization. To carry out extension on membrane, material itself has to be elongated following material elasticity. Extending material requires huge amount of force since modulus of material is generally high to prevent significant inflation. On the other hand, to perform contraction on membrane, tensile stress on membrane carried by internal pressure should be totally removed. After stress is removed, membrane is able to start out-of-plane deformation generally called “wrinkling” which results in large deformation.

Bending property of inflatable cylindrical tube made of membrane materials is investigated by several researchers and physics-based models are addressed. Comer and Levy [20] developed a bending model from basic assumptions about the state of stress in material of an inflatable beam based on conventional Euler beam theory. Main et al. [1] further developed the accuracy of the model analysis. As the applied bending moments increases, the bending stress equilibrates the axial stress in the material skin and wrinkling begins in that region. Figure 2.3 shows model and experimental load-deflection curves. When no beam fabric is wrinkled, the model predicts a straight line relationship between beam load and tip deflection. After wrinkling starts, the stiffness of tubes dramatically decreases and deflections become large. The experimental results corresponded with the model prediction very well at higher aspect ratio between tube length and diameter, since a tube at lower aspect ratio shows shearing effect. Davids et al.[21] simulated cantilever bending tests by using a mixed-interpolation Timoshenko beam finite element based on virtual work principles accounting for material wrinkling, shear deformations and internal pressure. The simulation demonstrated that pressurized air significantly increases beam stiffness as the beam volume decreases due to deformation.

According to the analysis, it is clear that axial stress due to inner pressure and volume change over bending deformation are critical causes to increase stiffness of inflatable tubes. Inversely, eliminating the two causes is required to reduce bending stiffness and the causes correspond to the above requirements.

2.3 Classification of mechanical concepts of variable shape pneumatic cylinder

This section provides an overview of existing mechanical concepts of pneumatic cylindrical structures for performing good shape change capability and classifies those concepts as shown in Figure 2.4. The overview includes discussion how those concepts account for the above requirements.

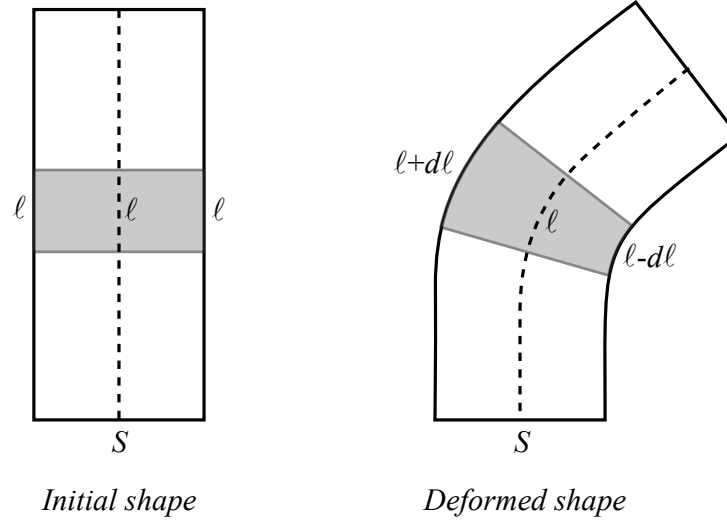


Figure 2.2. The change of lengths of neutral axis and inner/outer side regions over bending deformation; the left image is initial shape and the right is deformed shape.

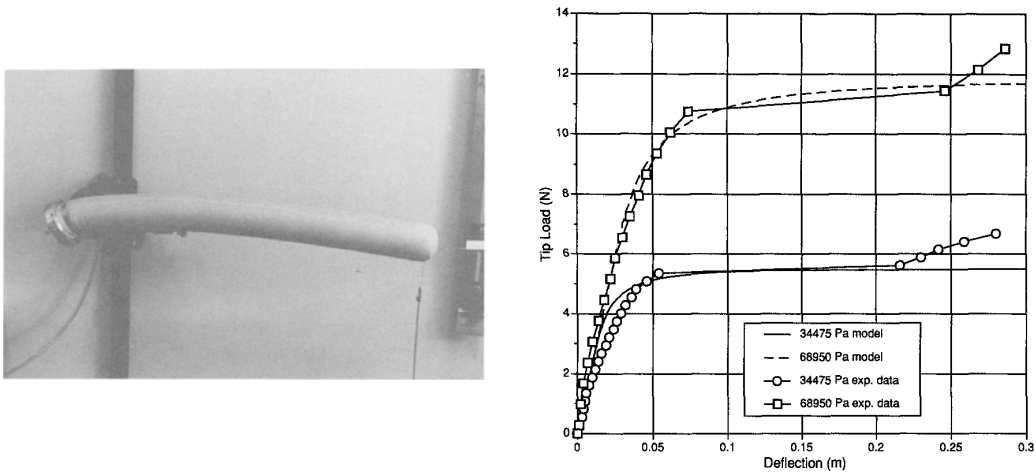


Figure 2.3. The bending analysis of inflatable tubes by Main et al. [1]; (a) Photograph of bending test, (b) Model and experimental load-deflection curves for the beam with aspect ratio at $l/d = 6$.

2.3.1 Restraint group

In this type, cable or other alternative material is attached on a cylindrical membrane tube to prevent inflation in axial direction and to reduce bending rigidity as shown in Figure 2.5. Then, the element changes tensile stress condition by creating wrinkled areas, which can be easily contracted and elongated. The attached elements are generally called restraint. In the most cases, the restraint is a cable element attached in longitudinal direction of a cylindrical membrane tube. Paired restraint cables are attached on both lateral sides. Restraint carries longitudinal load due to inner pressure instead of membrane; thus, longitudinal stress on membrane becomes relaxed. That means membrane

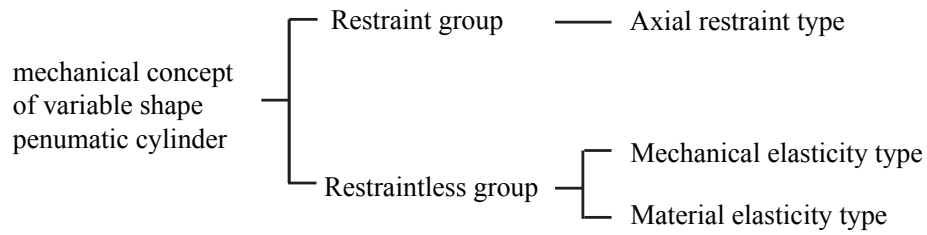


Figure 2.4. Classified types of mechanical concepts of variable shape pneumatic cylinder.

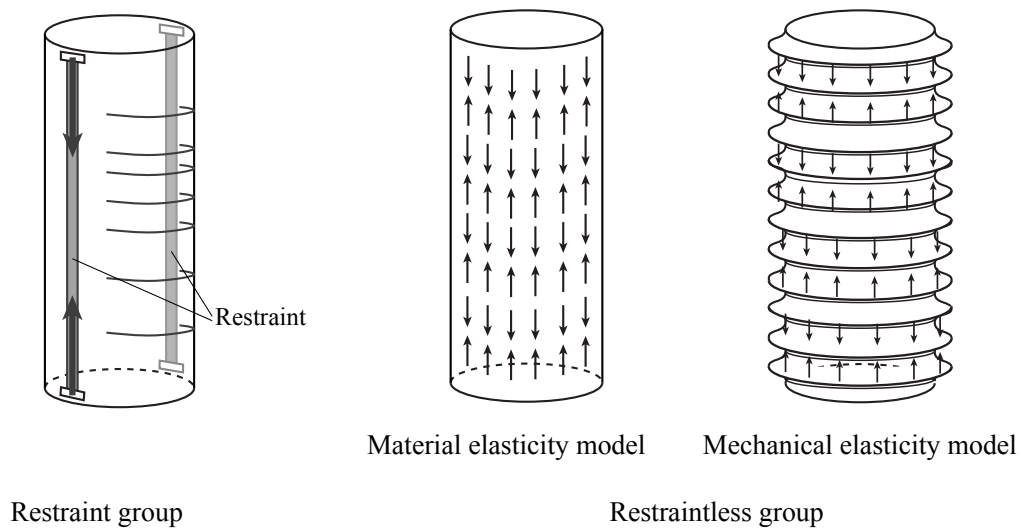


Figure 2.5. Schematic image of restraint and restraintless group of mechanical concepts.

has capability to perform contraction and extension without inducing material strain or removing tensile stress. Note that a restraint cable should have enough tensile strength to sustain longitudinal pressure force.

Restraint group is widely presented in the joints of past and present spacesuits and spacesuit prototypes. Flat pattern joint is a major example which is actively used in knee and elbow joints on current the U.S. EMU. The joint is composed of flat patterned membrane elements sewn each other to shape cylinder with many pleats perpendicular to longitudinal direction on outer side of joints, and restraint cables are attached in both lateral sides. The pleats are extracted when the joint is flexed. Fabric and rubber molded convolute (bellows) joint with restraint cables used on the old A7L and A7LB Apollo lunar suits are also classified into the restraint group.

Drawback of the type is related with the restraint cables in longitudinal direction. Most of the

models have restraint cables in longitudinal direction. Those cables become stiffened due to pressurization, and it resists flexion.

Instability is another drawback for the type. Uncontrolled pneumatic forces are exhibited as a phenomenon referred to as “snaking.” The cylindrical structure deflects out of longitudinal alignment, forming “S” shape.

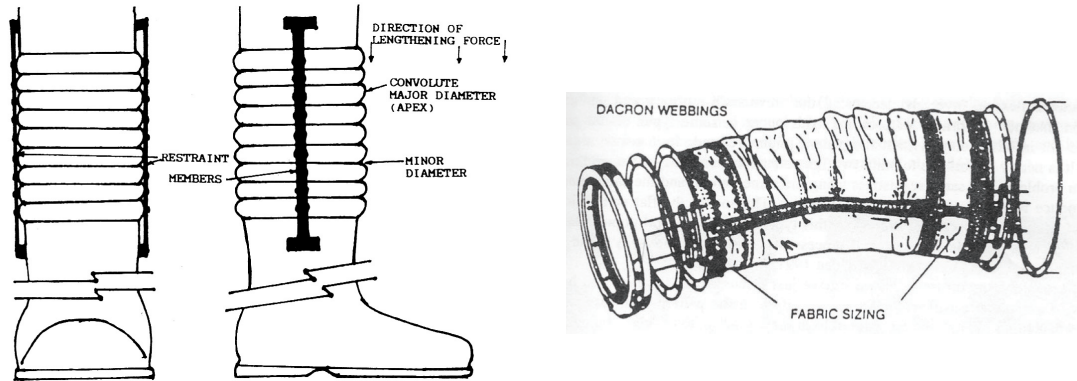


Figure 2.6. Examples of restraint group: a) convolute joint applied for Apollo , b) flat pattern joint used in current US EMU [72].

2.3.2 Restraintless group

While the restraint type sustains pressure load by locally attached restraint elements, joints of this group sustain pressure load by total structure without restraint elements.

In the group two elasticity models are classified: mechanical elasticity model and material elasticity model.

In mechanical elasticity model, wall of structure is shaped with patterned shape instead of plain membrane to store mechanical energy with certain amount of elongation. Patterned structure takes a roll like helical spring in longitudinal direction, in which tiny strain in each element accumulates and performs large elastic deformation. To bend the cylindrical model, the patterned structure in outer side and inner side area of bending of the cylindrical structure is extended and contracted, respectively. Extension is realized by additional deformation of the patterned wall, which needs less force than inducing material strain. A typical example of the type is bellows or toroidal convolute without restraint cables. Wall of bellows is patterned with U-shaped corrugation. Under pressurization, cylindrical bellows would expand in certain amount. Additional force changes the shape of the wall resulting in wall extension and contraction. Note that material used in the model should be selected to keep the shape against pressurization. This type is not appropriate for operating under high pressure inside since material stiffness should be higher as well as bending property becomes poor.

Material elastic model is another type which use such elasticity. An experimental investigation is taken by Tanaka et al. [22] and Korona et al. [23] by testing mobility of elastic gloves. Extension

in outer side area is achieved by stretch of material itself. Material used to form a cylinder is elastic material which has low Young modulus. Therefore, the subjected force to extend outer area is smaller. Figure 2.8 explains comparison between a plain membrane cylinder made of high modulus and low modulus. A membrane cylinder made of low modulus corresponds to material elastic model. Due to pressurization membrane is extended in certain amount. Material elastic model requires less compression and tensile stress to perform the same amount of strain. It should be taken into account that material elastic model has greater initial inflation when pressurized. In addition, it should be noted that exerting force inevitably increases when operating pressure is high because material with higher Young modulus has to be selected for the cylinder in order to keep its shape. Large bending deformation seems difficult in higher pressure difference is applied.

Both mechanical and material elasticity models assume that certain amount of force to exert to elongate spring-like wall or elastic wall is necessary. Thus, both models cannot get rid of resistant force for bending the cylinder completely.



Figure 2.7. A typical example of restraintless group of mechanical concepts: rubber bellows.

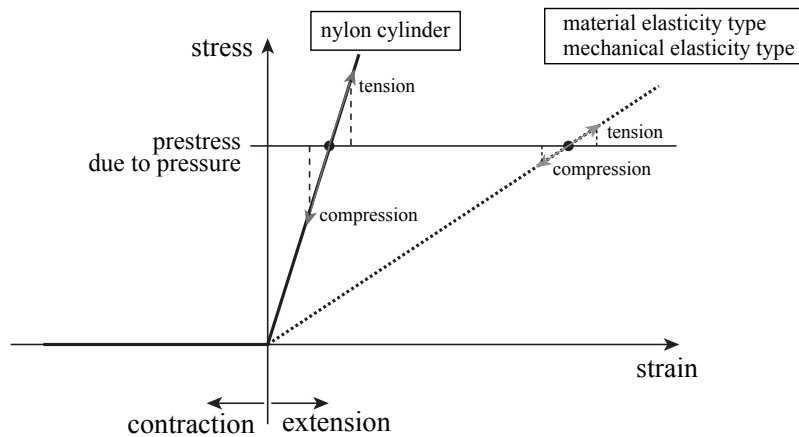


Figure 2.8. Comparison of stress-strain diagram between nylon cylinder and elasticity types.

2.4 New approach on variable shape pneumatic tube

From the consideration of present variation of variable shape pneumatic tubes, it is confirmed that both types are faced with individual problems, and need further improvement.

In the most cases of restraint group, longitudinal restraint takes a role to change tensile stress condition on pneumatic tube. The restraint cable itself becomes stiffened and hinders flexion motion. Therefore, the author gained motivation to focus on the new type of restraint which does not run in longitudinal direction. The new type of designs is explained with the name of Membrane Beam Folding Structure later.

On the other hand, as for restraintless group, as operating pressure is high, bending force inevitably increases. This is because both mechanical elasticity and material elasticity type have to deform fully tensioned due to pressurization material. Therefore, initial shape optimization of structure to investigate possibility to avoid facing with full tensile force is focused. Tensile stress carried on membrane varies depending on geometry of the structure under the same inner pressure. For example, it is known that tensile stress is proportional to curvature of curved surface in an air-pressurized structure.

Changing shape and reduce tensile stress in axial direction is effective way to contract material with minimal external force. However, for expanding material it is not effective since excess material is not yet exist and need big force to make strain. To perform expansion with minimum material, adding excess skin in direction for expansion is necessary. The new type of restraintless type named Isotenoid Excess Membrane Structure is proposed and explained in the following section.

2.5 Membrane Beam Folding Structure

Possibility of restraint which is not attached in longitudinal direction is investigated. A core idea is restraining the shape of cross section at each plane of a cylinder, which functions as restraint in longitudinal direction since a membrane cylinder is forced to be folded in axial direction as shown in Figure 2.14. The idea means that a membrane cylinder is able to be restrained in longitudinal direction without attaching longitudinal restraint cables. The idea is possible to eliminate problems of existing mechanical concept with axial restraint. There is a number of folding patterns for a cylindrical tube to fold it into axial direction. It is possible to select the most appropriate folding patterns.

2.5.1 Variation of folding patterns for thin-walled objects

Folding has been long studied and developed associating with Origami, an ancient Japanese art of folding paper. Geometry and mathematics of folding two- and three-dimensional objects have been broadly studied [24]. In case of two-dimensional objects, folding methods of circular or flat shells are comprehensively studied by Nojima [25, 26]. Miura-ori is also one of the folding patterns to fold flat sheet (see Figure 2.10(a)) [27]. Folding patterns on three-dimensional geometry, such as a corn, a

cylinder and any other curved surface, have been developed by many researchers [28]. A cylindrical shell can be folded into axial direction by several folding methods. Not only for folding methods for very thin-walled material, recently, Miura and Tachi [29] proposed rigid-foldable cylindrical polyhedra, called star-polyhedron, which is foldable for structures made by rigid panels and hinges.

Horizontal folding is one of folding pattern groups in which major folding valley lines are horizontal as shown in developments of Figure 2.11. The folding pattern consists of trapezoid or triangle elements, and base angles of them, α and β , are defined particularly so that cylinders are folded perfectly. In case of Figure 2.11(a), the pattern consists of one kind of trapezoid elements, and the sum of base angles, $\alpha + \beta$, are defined by π/N , and ratio of α to β is arbitrary. Note that N is the number of times by which horizontal folding line is divided. For example, the value of N in Figure 2.11(a) is 6. When α and β is equal as shown in Figure 2.11, angle of α equal to $\pi/2N$. When segment of a line BC in Figure 2.11(a) is 0, Figure 2.11(a) becomes Figure 2.11(c), and Figure 2.11(b) becomes Figure 2.11(d). A cylinder folded according to horizontal folding pattern in Figure 2.11(d) is in Figure 2.13(b) as is mentioned later.

Another folding pattern is Spiral folding of cylindrical shells as shown in Figure 2.12 which is a derivative pattern of Horizontal folding; major valley lines are inclined. The cylinder patterned with spiral folding is axially folded and simultaneously twisted. Triangulated cylindrical shells, on which triangle rigid shell elements are connected by hinges making cylindrical shape following the spiral folding pattern, are studied to investigate folding properties.

It is known that many folding structures exist in nature, for instance, petals, insect wings and leaves are compactly folded to enhance packaging efficiency [30, 31, 32] (see Figure 2.10(f)). Miura-ori is found in nature [33].

Mechanical properties of folding thin shell structures such as packaging capability and strength have been studied. For example, Schenk studied mechanical properties of folded textured sheets where local texture is created by means of a tessellated fold pattern, such as Miura-ori [34]. Guest and Pellegrino [35] proposed a variation of cylindrical foldable shell by twisting Yoshimura-pattern to enable a valid state in three-dimensional and also in the flat-folded state.

Based on those researches, folding structures find a number of applications in commercial products like folding maps, drinking bottles, containers, solar panels of space satellites, and shock-absorbing beams for automobile body [36]. The world's first deep space solar sail demonstration spacecraft "IKAROS" developed by JAXA embedded folding technology to deploy 20 m-span thin-film sail, and successfully deployed the sail in outer space [37, 38]. Kuribayashi et al. [39] developed new type of stent grafts for medical treatments, made of single foldable foil which is TiNi shape memory alloy to deploy by SMA effect. A related future application is morphing structures. Morphing structures are a relatively new field, with many concepts still in an early stage of development. Such as morphing wings where the shape for the wings of an aircraft is altered to change their aerodynamic qualities [40].

In our laboratory, Gubarevich [41] designed new concept of a single-axis spacesuit joint, and applied curved shell folding structures into the outer and inner part of the joint to give flexibility as shown in Figure 2.9.

One of the patterns to fold a cylindrical shell completely along the axis is called Yoshimura pattern (diamond pattern) that has been studied in connection with a post-buckling behavior of a cylindrical shell. Figure 2.10(c) shows the local buckling under axial loading. The pattern is composed of a number of rhombic elements. Sides of a rhombus are mountain lines, and the horizontal diagonal corresponds to a valley line. The post-buckling behavior is theoretically non-extensional [42]. Following minimum energy theory the buckling behavior occurs. Interestingly in nano-scale observation, Yoshimura buckling pattern is confirmed by calculation. Surface of multiwalled carbon nanotubes (MWCNT) shows Yoshimura pattern under bending loading [3, 43](see Figure 2.10(e)). The pattern and other patterns are formulated by Nojima [25] as a folding pattern of a cylinder (see Figure 2.10(d)).



Figure 2.9. Single axis joint with curved shell folding structures [41].

2.5.2 A new design concept with transverse restraint elements for restraint group

Yoshimura folding pattern is one of the most appropriate patterns for giving axial restraint function because of its simplicity. The cylinder folded by Yoshimura folding pattern is explained as a patterned polyhedron. The polyhedron has cross sections shaped with a regular n -polygon at equal interval along axial direction. Constraining the shapes of the cross section by cables or rigid beams will perform as axial restraint cable as shown in Figure 2.14. In addition, a cylinder with Yoshimura folding pattern is able to keep area of cross section as is folded, which is important since arm or finger is inserted inside. Also, since Yoshimura pattern is one of the horizontal folding pattern, folding direction (restraint direction) corresponds to axial direction of cylinder. It is also reasonable to perform the same function with axial restraint cable.

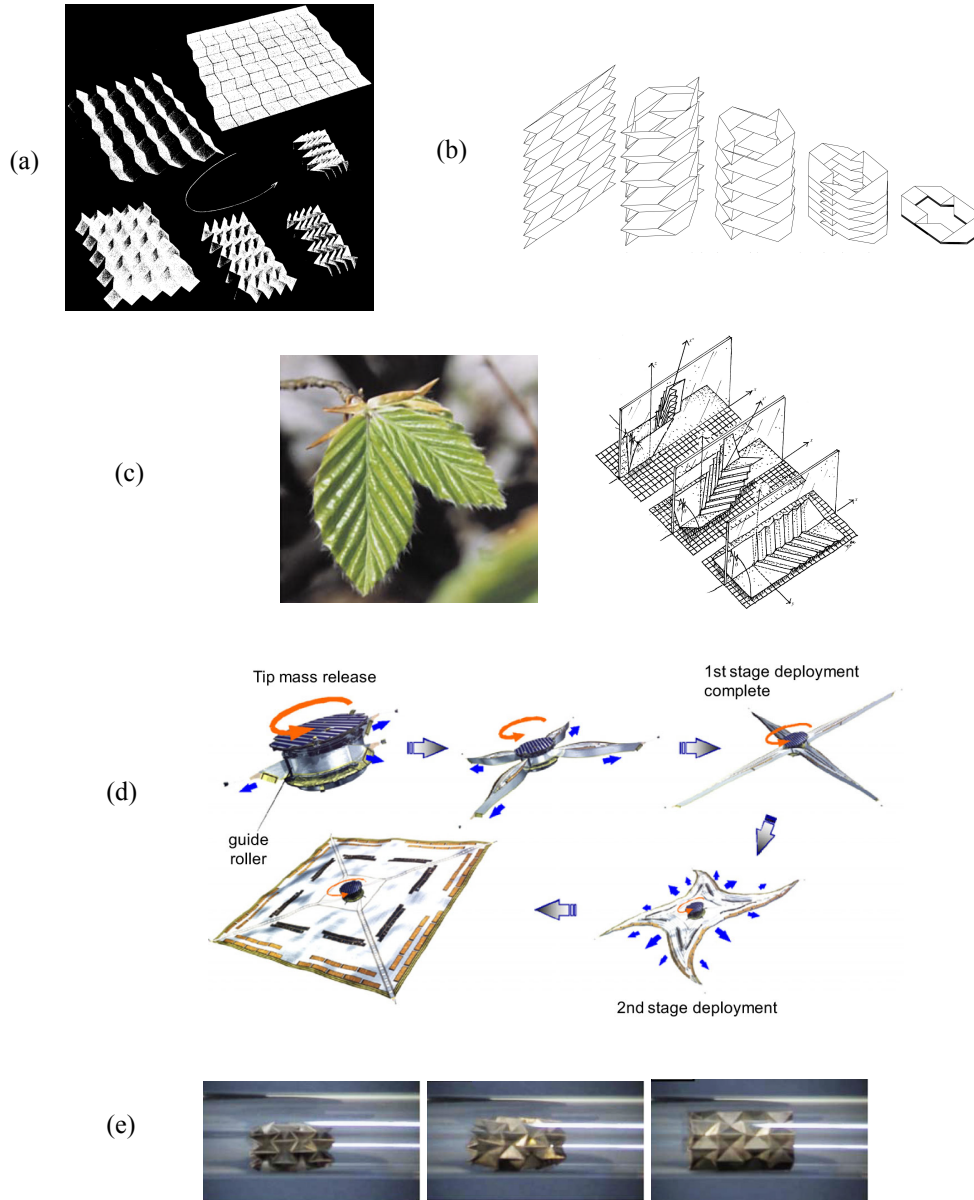


Figure 2.10. Examples of folding structures; a) Miura-ori [27], b) star-polyhedron [29], c) an example of folding from leaves [30], d) deployment procedure of IKAROS [37], e) folded foils in application for stent grafts [39].

To constrain the shapes of the cross sections, rigid beams were selected since rigid beam has higher bending rigidity against pressure than cable element. The design is called Membrane Beam Folding Structure, in short, MBFS. In order to reveal a feasibility of MBFS, the property of restraint in axial direction and bending property is investigated in Chapter 3.

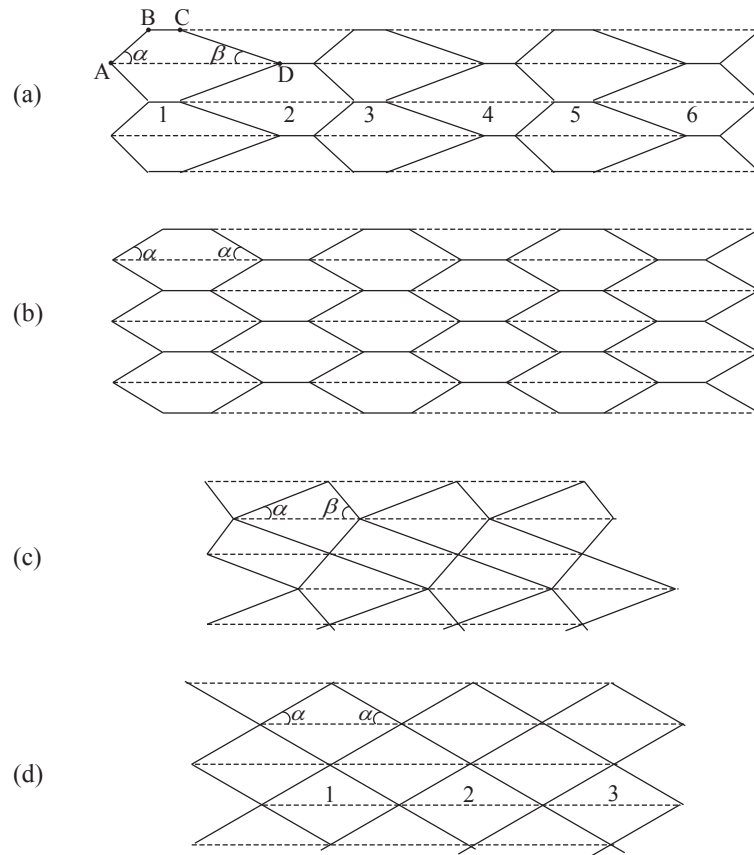


Figure 2.11. Examples of horizontal folding patterns.

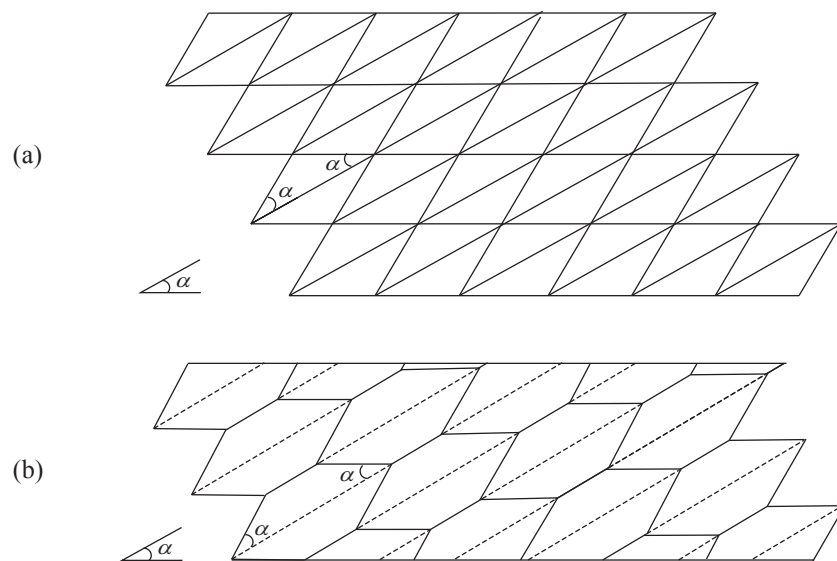


Figure 2.12. Examples of spiral folding patterns.

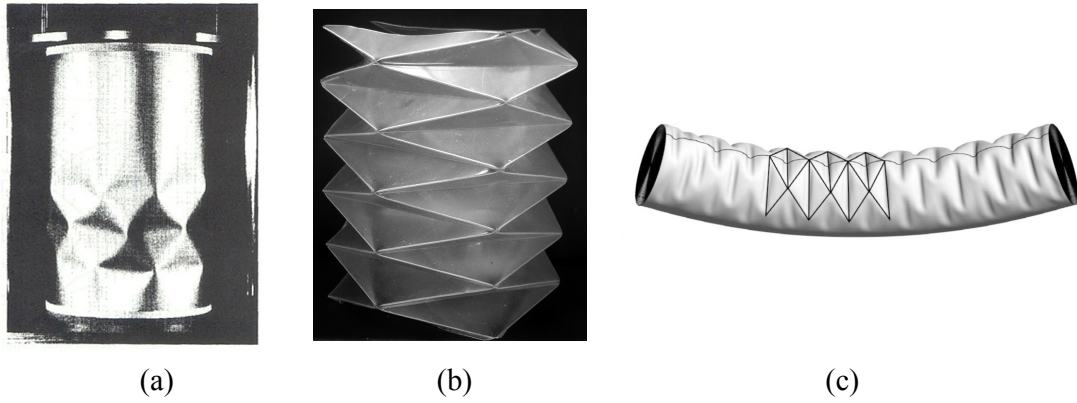


Figure 2.13. Yoshimura pattern; a) a buckling pattern of cylindrical shell under axial loading [2], b) a cylindrical shell folded with Yoshimura pattern, c) Yoshimura folding pattern simulated in MWCN [3].

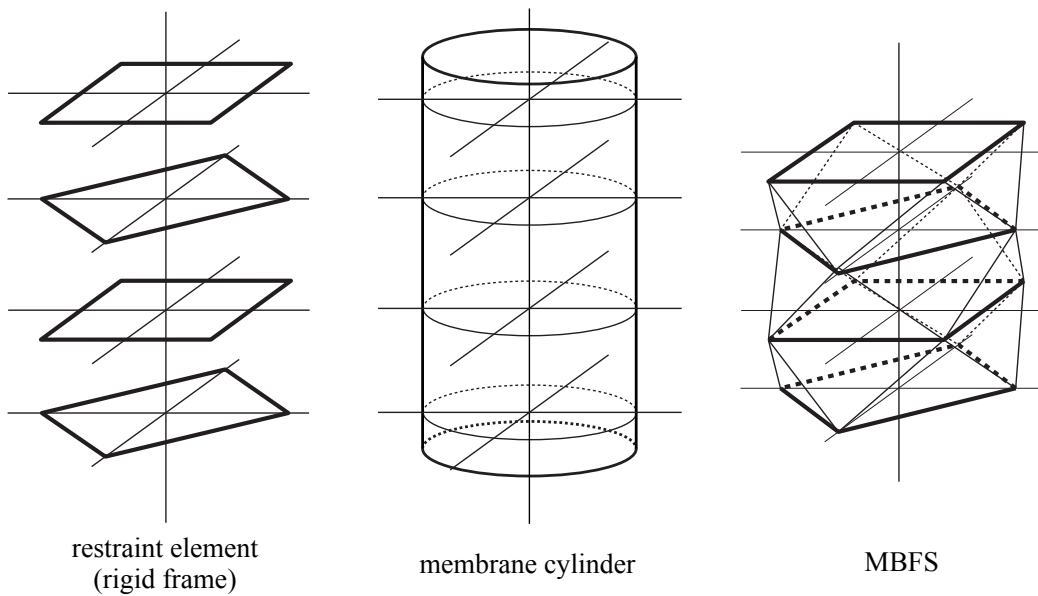


Figure 2.14. Concept of MBFS. Membrane is restrained with rigid frame resulting in folding in axial direction.

2.6 Isotenoid Excess Membrane Structure

The other motivation of the thesis is designing flexible pneumatic cylinder without attaching restraint. Current restraintless group using material or mechanical elasticity is not suitable concept for higher internal pressure because of high bending force and limited deformation range. From the analysis of problems of current restraintless group, it is clear that high tensile stress on axial direction is the essential cause of low bending property. Thus, method to avoid the full tensile stress carried on axial direction due to inner pressure is focused. The amount of tensile stress is highly related with the shape of envelop. To reduce the tensile stress, the shape of envelop should be modified from a plain cylinder. Designing started from a body of revolution which is merged with a plain cylinder. A body of revolution is located in the middle of cylinder along axial direction and take a roll of contraction and extension of membrane performing deformation. The reason why bodies of revolution are candidates of joint shape is because the configuration is able to maintain their volume while side face rotates along their axis.

2.6.1 Shape optimization for axial tensile stress reduction

The first candidate of rotational body is a cylinder. However, a cylinder soon faced difficulty to treat edge between side face and upper/ bottom face. Meanwhile, a spheroid does not have such edge; then, it is feasible to design. One shape that satisfies these conditions is isotenoid. Isotenoid is the oblate spheroid that carries interior pressure purely by meridional stress, i.e. the circumferential stress is zero everywhere. Since circumferential direction corresponds to axial direction of cylinder, it would be highly effective on flexibility of the pneumatic cylinder.

Taylor [4] first derived the form in a study on shape of parachutes as shown in Figure 2.16. The shape is applied in a number of applications. End-closure of general cylindrical pressure vessels are shaped with isotenoid since all filaments are able to carry the same amount of load [44]. A type of pneumatic actuators, which is called pleated pneumatic artificial muscle (PPAM) as shown in Figure 2.16 [5], also has a shape of isotenoid. Super pressure balloons, which are pressurized inside for long-term science observation flight, are based on isotenoid [45].

From now, this section describes why isotenoid results in such stress distribution. Think of meridional curve profile of revolution when it is loaded by inner homogeneous pressure. As shown in Figure 2.15, the principal stresses T_1 and T_2 , act along and perpendicular to the radii. Suppose no shearing stress across radii or concentric circles. The equilibrium of a infinitesimal portion ABCD of the surface bounded by concentric circles in the fabric whose radii are r and $r + dr$, and by radial lines in the fabric, that is by the profile curve in which two axial planes which make a small angle $d\theta$ with one another cut the surface of revolution. ψ is the angle between the tangent plane at any point of surface and the horizontal tangent plane at the vertex. The following is based on the work done by Taylor [4] with some minor modifications.

To begin with, the area of the infinitesimal portion is expressed by $rd\theta(dr/\cos\psi)$. The vertical

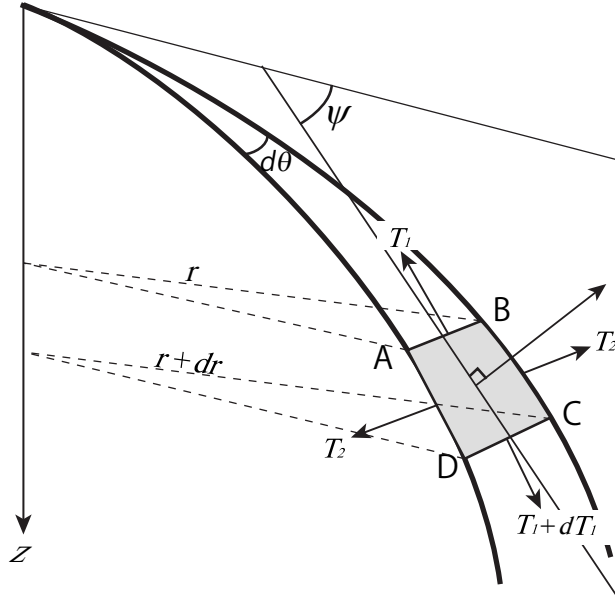


Figure 2.15. Free body diagram of infinitesimal element of pressurized revolution.

component of the pressure force acting on the infinitesimal surface gives

$$pr d\theta \frac{dr}{\cos \psi} \cos \psi, \quad (2.1)$$

and the vertical component of the force due to stresses in the material is

$$\frac{d}{dr} (T_1 r d\theta \sin \psi) dr. \quad (2.2)$$

Thus, the equation derived from the vertical component is

$$\frac{d}{dr} (T_1 r \sin \psi) = pr \quad (2.3)$$

The horizontal component of the pressure force on the small portion acts radially is

$$rp \tan \psi d\theta dr. \quad (2.4)$$

The horizontal component of the radial stress acting outwards in the material is

$$\frac{d}{dr} (T_1 r \cos \psi) d\theta dr. \quad (2.5)$$

The horizontal components of the circumferential stress acting inwards is

$$T_2 d\theta (BC) = T_2 d\theta \frac{dr}{\cos \psi}, \quad (2.6)$$

Hence, the equation derived from the horizontal forces on the small portion is

$$\frac{d}{dr} (T_1 r \cos \psi) = \frac{T_2}{\cos \psi} - pr \tan \psi. \quad (2.7)$$

According to equation 2.3 and 2.7, we get the relationship between r and ψ . We can determine the shape of revolution which is so flat that it is on the point of crinkling by putting $T_2 = 0$ between

equation (2.3) and (2.7).

By making use of elliptic functions it is possible to make this calculation and to obtain the profile of the revolution.

Equation (2.3) may be integrated directly, thus:

$$T_1 r \sin \psi = \frac{1}{2} p r^2. \quad (2.8)$$

Substituting this value of T_1 in equation (2.7),

$$p \frac{d}{dr} \left(\frac{1}{2} r^2 \cot \psi \right) = -p r \tan \psi, \quad (2.9)$$

or writing ζ for $1/2r^2$, equation (2.9) becomes

$$\frac{d}{d\zeta} (\zeta \cot \psi) = -\tan \psi \quad (2.10)$$

Then,

$$\frac{d\zeta}{\zeta} = \cot \psi d\psi \quad (2.11)$$

The integral of equation (2.11) is

$$\log \zeta = \log \sin \psi + \text{constant}. \quad (2.12)$$

Hence,

$$\frac{r^2}{R^2} = \sin \psi \quad (2.13)$$

where R is the greatest horizontal radius.

Equation (2.13) is the equation of the profile of the revolution which is so flat that it is on the verge of crinkling. To be expressed in Cartesian co-ordinate, equation (2.13) is modified by means of elliptic functions. Thus,

$$\tan \psi = \frac{dz}{dr} \quad (2.14)$$

where z is the depth below the vertex.

Equation (2.13) can be expressed that

$$\frac{dz}{dr} = \frac{\sin \psi}{\sqrt{1 - \sin^2 \psi}} = \frac{r^2}{\sqrt{R^4 - r^4}} \quad (2.15)$$

and

$$z = \int_0^r \frac{r^2 dr}{\sqrt{R^4 - r^4}}. \quad (2.16)$$

To integrate this assume

$$r = R \operatorname{cn} u \quad (2.17)$$

cn is elliptic function usually denoted by that symbol. Then,

$$\text{sn}^2 u = 1 - \text{cn}^2 u = 1 - r^2/R^2, \quad (2.18)$$

$$\text{dn}^2 u = 1 - k^2 \text{sn}^2 u = 1 - k^2(1 - r^2/R^2) \quad (2.19)$$

When we take the modulus k to be $1/\sqrt{2}$, $\text{dn}^2 u = 1/2(1 + r^2/R^2)$. Hence,

$$\frac{1}{\sqrt{1 - r^4/R^4}} = \frac{1}{\sqrt{2} \text{sn} u \text{dn} u} \quad (2.20)$$

also, differentiating equation (2.17)

$$dr = -R \text{dn} u \text{sn} u \text{dn} u. \quad (2.21)$$

Hence equation (2.16) becomes:

$$z = R \int_K^u \frac{\text{cn}^2 u \text{dn} u \text{sn} u \text{dn} u}{\sqrt{2} \text{sn} u \text{dn} u} = -\frac{R}{\sqrt{2}} \int_K^u \text{cn}^2 u \text{dn} u. \quad (2.22)$$

The lower limit of integration is K because $\text{cn} u = 0$ when $u = K$, where K is the complete integral of the first kind. Its value for modulus $1/\sqrt{2}$ is given as 1.8541. z may be expressed in terms of the elliptic integral $E(u)$ thus,

$$z = -\frac{R}{\sqrt{2}} \int_K^u (2\text{dn}^2 u - 1)du = \frac{R}{\sqrt{2}} [2E - E(u) - (K - u)], \quad (2.23)$$

where E is the complete elliptic integral of the second kind. Its value for $k = 1/\sqrt{2}$ is $E = 1.3506$. The value of

$$\text{cn} u = \frac{r}{R} \quad \text{and} \quad \frac{1}{\sqrt{2}} [2E - E(u) - (K - u)] = \frac{z}{R}, \quad (2.24)$$

are given for a series of values of u , and the shape of the meridional profile is plotted in Figure 2.17.

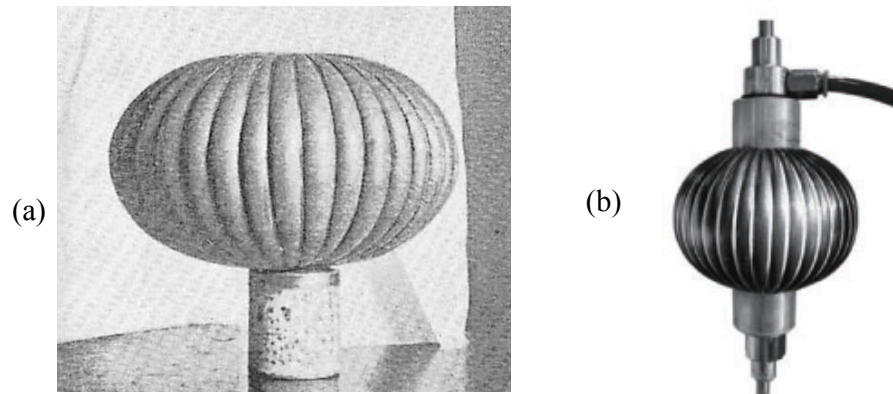


Figure 2.16. Examples of isotenoid shape; (a) the first derivation of isotenoid and proof given by Taylor [4], (b) Pleated pneumatic artificial muscle [5].

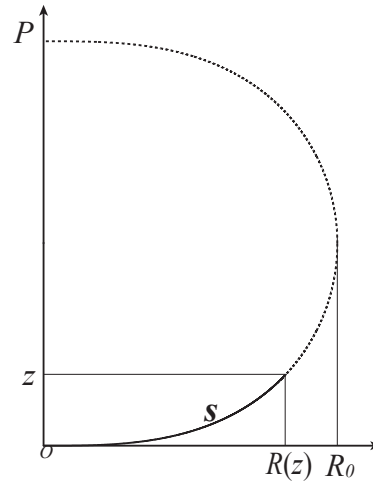


Figure 2.17. Meridian curve of equilibrium pressurized revolution.

2.6.2 A new design concept with optimized shape for restraintless group

Isotensoid carries the load due to inner pressure by only meridian tensile stress nor circumferential stress. Isotensoid is connected with a cylinder by corresponding axial direction of a cylinder and circumferential direction of isotensoid. While reducing tensile stress is effective for large contraction and extension with less applied force, it is not sufficient. Excess area of membrane material should exist along the deforming direction as shown in Figure 2.18.

When excess skin exists, shape has potential to be variable without applied force. To illustrate the function of excess skin, Figure 2.19 explains two-dimensional case. Suppose that a circle-like line are restrained to move only on the trajectory distant from radius r , and point A is fixed. Point B is freely moved to point B'. The movement is realized since outer side has excessive line.

Thereby, isotensoid with excess skin in circumferential direction is designed with adding extra number of gores, which is constituting membrane element of isotensoid part. The structure is another concept design, which we call Isotensoid Excess Membrane Structure.

Selecting isotensoid as initially-designed enclosure brings another important characteristic that is maintaining constant volume during flexion. Decrease of volume can become a predominant cause of higher torque for joint flexion, but the proposed joint is able to maintain the interior volume. At flexion wrinkled membrane on one side (outside) expands along circumferential direction, and at the same time, opposite side of skin contracts. Since isotensoid remains axisymmetric during flex motion, the enclosed body is able to keep the same volume. Thus, there is no extra work necessary to compress inner volume.

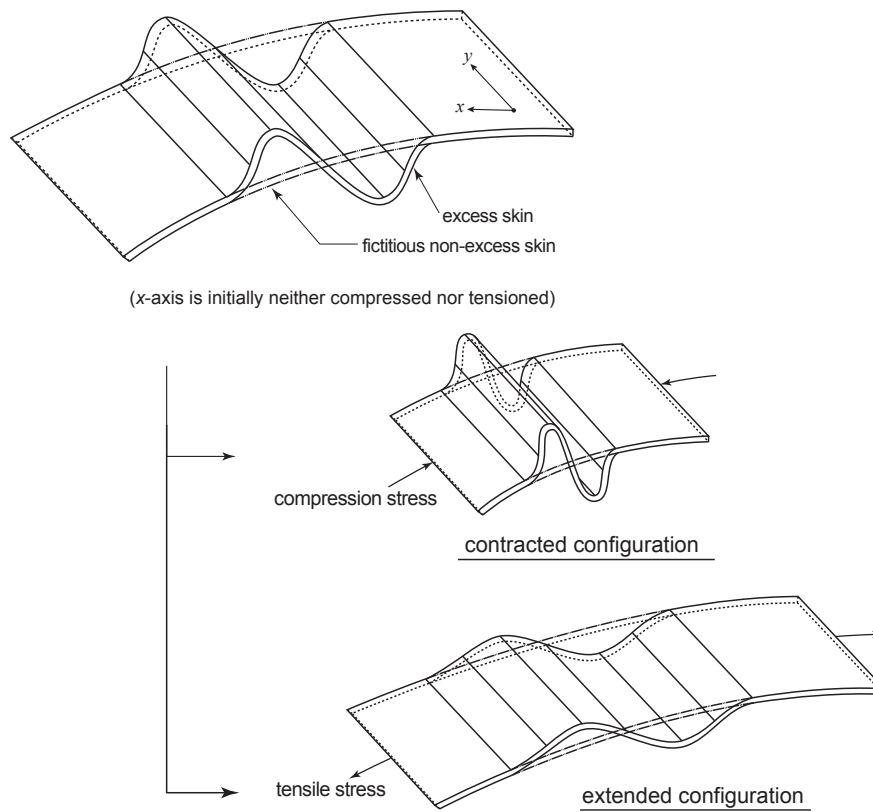


Figure 2.18. Contraction and extension with excess skin.

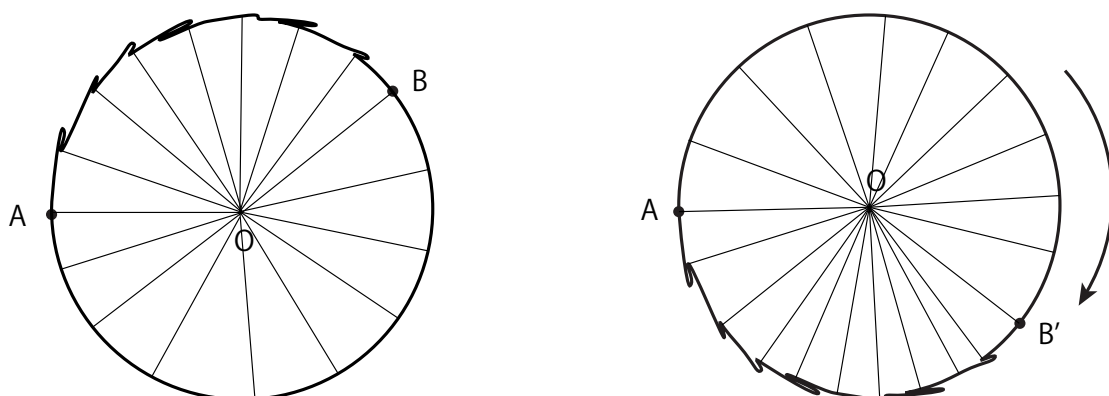


Figure 2.19. Two-dimensional concept of shape variable pneumatic structure.

2.7 Summary

This chapter described the connection between the general motivation and the actual problems in this thesis. Motivation to improve flexibility of current spacesuit joints is reduced to a simplified mechanical problem. The mechanical problem is to investigate a new design of closed pressurized cylindrical membrane structure, which can bend in one plane with a minimal applied force. In this thesis, such structures are called variable shape pneumatic structures. From the overview of existing variable shape pneumatic structure, those are divided into two design categories: restraint group and restraintless group. It is found that both design groups have problems in terms of bending properties. the cause of each problem is focused and eliminating the cause is tried.

In case of restraint group, since axial restraints hinder bending motion and cause instable behavior, the structure which does not have axial restraint is proposed. Instead, transverse restraints inducing folding constraint are designed.

In case of restraintless group, although the structure is simpler than restraint group, bending property is not enough and bending property becomes very poor as inner pressure is high because tensile stress in axial direction is significantly high. The structure which does minimize tensile stress in axial direction by means of optimization of shape is proposed.

The following chapters, chapter 3 and 4, describe design concepts, geometry, manufacturing process, bending test procedure and bending properties of the both proposed structure.

Chapter 3

Fabrication and Characterization of Membrane Beam Folding Structure

3.1 Introduction

This section, to begin with, gives design and fabrication method of the design introduced in Chapter.2 named Membrane Beam Folding Structure (MBFS). This design is aimed to investigate potential bending property of restraint group. Unlike axial restraint type of structure, MBFS restrains axial inflation by transverse restraint applying appropriate folding pattern. To investigate fundamental characteristics of MBFS, two experiments have been mainly performed: (1) the axial elongation test for investigating the inflation of MBFS and (2) the bending test for measuring the bending properties of the pressurized MBFS tube.

3.2 Overview of design

Here, a new structure named Membrane-Beam Folding Structure (MBFS) is proposed. In MBFS, contraction and expansion necessary for bending are realized by folding and unfolding of the tube's fabric. The proposed structure basically represents a fabric cylinder on which polygonal metal frames, consisted of several beams, are regularly attached.

The design is inspired by folding cylindrical shells. As is described in Section, many folding patterns have been developed to fold cylindrical shells, Yoshimura-pattern take advantages in terms of several aspects: simple for manufacturing, loading is perpendicular to axis, radius of cylinder does not change substantially.

If the envelope is pressurized, folding pattern may disappear and material may become fully tensioned, extraction and contraction will not function. Thus, to keep folding state, rigid bars are tied on valley creases of the folding pattern, to keep folding state and guide folding and unfolding. The structure is named Membrane Beam Folding Structure. While there is a variety of configurations of MBFS, this study focuses on the one with Yoshimura folding pattern.

3.3 Experimental

3.3.1 Fabrication of MBFS tubes

A fabric cylinder is 128 mm in diameter and 236 mm in effective length. The cylinder is made with two layers of membranes. An inner layer is a 30 μm -thick polyethylene film to keep airtightness, and the outer layer is 170 μm -thick nylon fabric to maintain strength. Stress-strain diagram is shown in Figure 3.1. Modulus of nylon fabric is 333 MPa, and polyethylene is compliant with outer layer's stretch when is loaded with the stress over 8 MPa. There is no difference between filling direction and warp direction for nylon. The cylinder is closed from both ends with plastic caps binding the film and the fabric.

When a cylinder folded with the Yoshimura pattern is cut along the valley lines, a regular polygon with n sides is found. Polygonal frames are attached on that polygonal valley lines. An angle α between valley line and mountain line on the development of the Yoshimura pattern in Figure 3.2 is defined as $\alpha = \pi/2n$ [25].

In this experiment, beams, stainless steel pipes (external diameter: 3.1 mm, thickness: 0.3 mm, length: 100 mm), are laced by 0.47 mm-diameter stainless steel wire to prepare square frame ($n=4$). Vertexes of the square frames are sewed onto fabric with 21mm between the frames as shown in Figure 3.3 and 3.4.

For the sake of comparison, two other types of tubes are manufactured: plain type and ring restraint type. The plain type of tube is not modified from a nylon-polyethylene cylindrical tube. The ring restraint type of tube is made by the same way as the MBFS, but, instead of frames, wire rings are attached to the fabric with the same distance between them.

3.3.2 Experimental procedure

Instrument

Figure 3.5 shows apparatus of experimental setup. The setup is consists of the tube, tube fixture stand, compressor, valve and pressure gauge. The photograph of the setup is shown in Figure 3.6. The both end of the cylindrical tube are capped with polycarbonate end-caps. Figure 3.7 shows cross section of end structure. The end-caps have shapes of flange with grooves. After polyethylene and nylon material are fitted on the end-cap, O-ring is set on the grooves, then clamped by hose band on the O-ring. The structure is enhances sealing reliability. Stoppers are clamped next to house band to fix house band not to slip. A level is attached on the end-cap to guarantee tube's level. The accuracy of the level is 0.1 °. At the end-cap, a hanger is equipped to hook on tip loads. The pressure gauge AE10-181 manufactured by NAGANO KEIKI has measuring range of 0 to 200 kPa, and accuracy is $\pm 3\text{kPa}$.

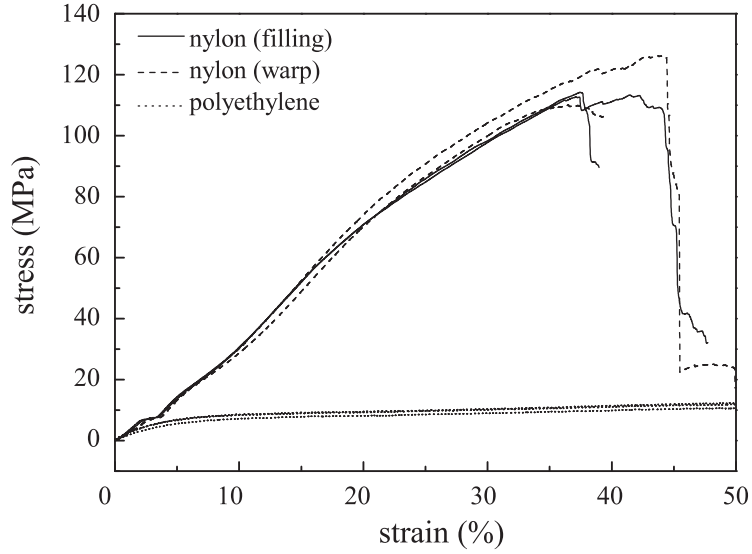


Figure 3.1. Stress-strain diagram of nylon and polyethylene. Nylon is examined in fill and warp direction.

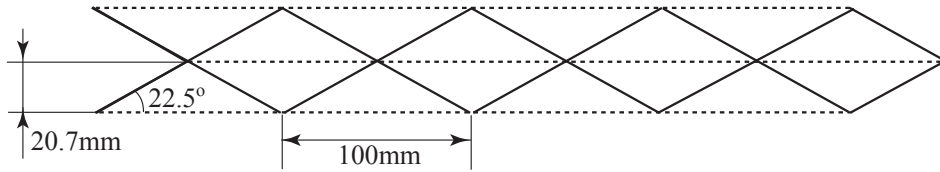


Figure 3.2. The development of the adopted folding pattern.

Procedure

Axial elongation test The base of MBFS is mounted in a horizontal position on a stand so that the tube faces to downward. The tip of the MBFS tube is free. Weight of the MBFS tube and the end cap is 84 g and 372 g, respectively. After discharging air from the tube (Figure 3.9, 0 sec), the tube is gradually pressurized from 0 to 60 kPa (gauge) keeping constant air flux. Development of the tube shape during inflation is recorded by a digital camera with 5 kPa step. Because of low precision of pressure measurement, pressure between 0 kPa and 15 kPa cannot be measured correctly. Instead, pressurizing time is noted.

Bending test A cantilever bending test is set up [1]. Three types of tubes are prepared. Each is pressurized with air up to 15 kPa (gauge), and then a valve of inlet is closed. The tube is initially

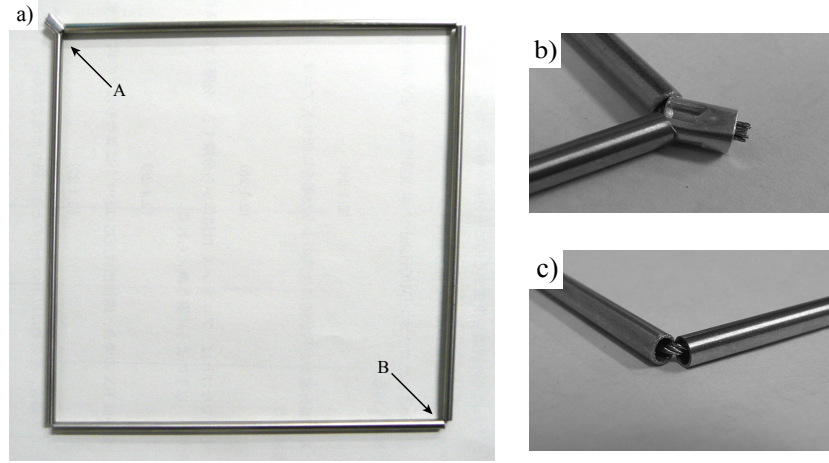


Figure 3.3. Detail of polygonal frame; a) frame, b) enlarged view of A, c) enlarged view of B.



Figure 3.4. Photograph of MBFS.

set parallel to the ground. Then, tip load F is applied from 0 to 98 N with 2.45 N step by hanging weights on the end cap. As the deformation becomes large, bending moment will not be accurate, and part of load may compress tubes unless the load is applied perpendicular to tubes. Thus, to maintain the load perpendicular to tubes, the base of tubes is inclined so that the end cap is always vertical as shown in Figure 3.8. The angle of inclination is defined as a deformation angle ϕ , giving magnitude of deformation. Deflection l is also given to explain magnitude of deformation, which value is used for the following numerical analysis. Photograph of the tube shape is taken to measure ϕ , and internal pressure p is recorded.

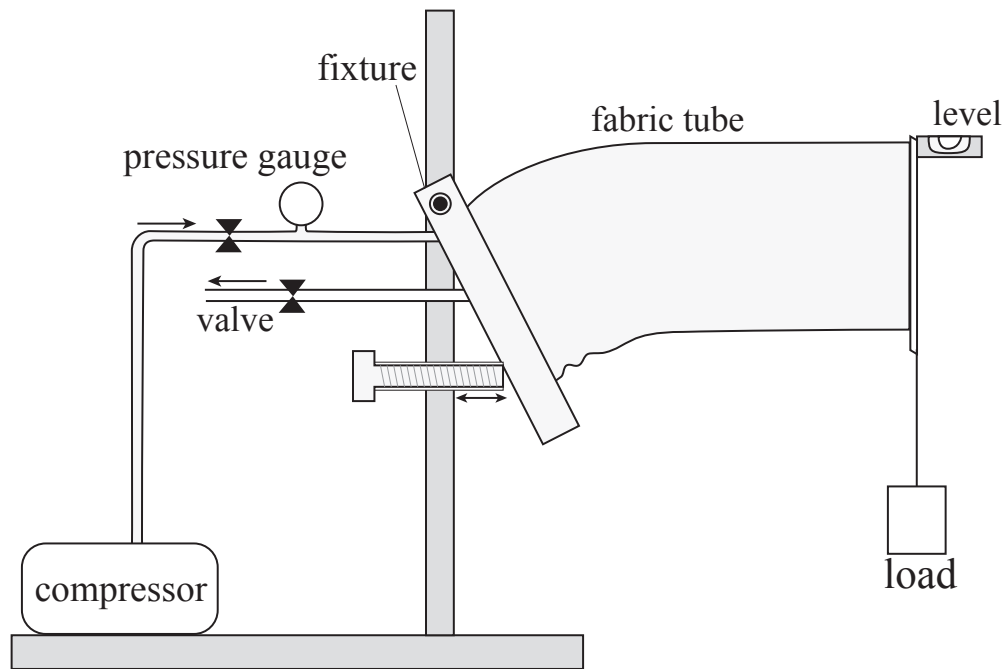


Figure 3.5. Schematic of measuring instrument.



Figure 3.6. Photograph of measuring instrument.

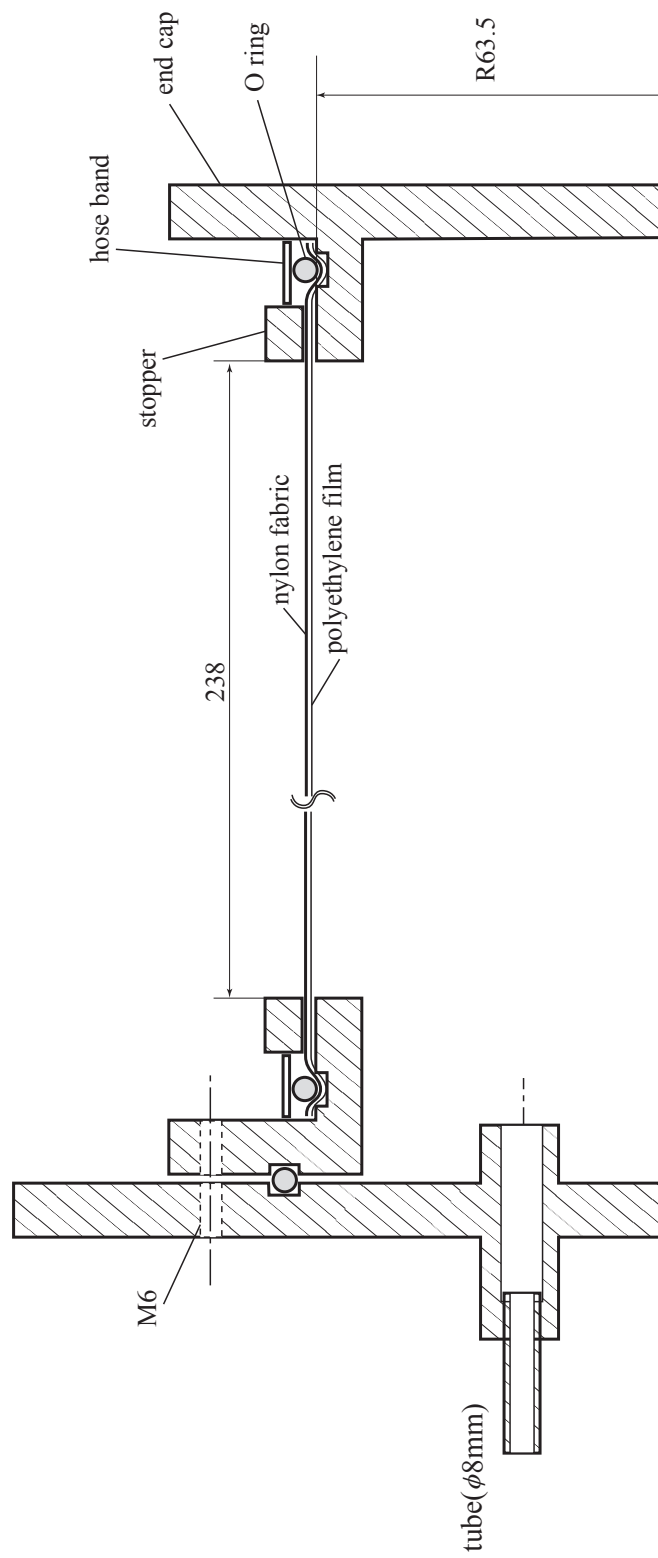


Figure 3.7. Detail of tube end.

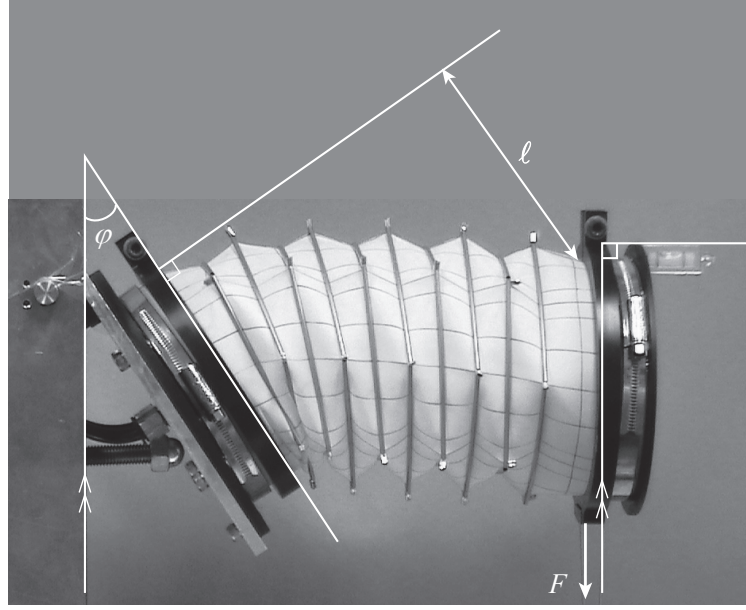


Figure 3.8. Experimental apparatus of bending test and definition of parameter.

3.4 Results and discussion

3.4.1 Effects of internal pressure

The results of the axial elongation test are presented in Figure 3.9. As can be seen from Figure 3.9 (0 kPa, 0 sec) in unpressurized state the MBFS tube is mostly folded. When air is gradually pressurized into the tube, the MBFS tube starts to elongate. At the same time polygonal frames three-dimensionally constraining the fabric start their bending slightly. When pressure reaches values higher than 40 kPa, the tube mostly stops its elongation, and frames gradually lose their constraining function.

This behavior is explained by the beam's bending. When the MBFS is not pressurized, the rigid beams keep valley lines straight, so the MBFS is completely folded. Then, as the pressurization starts, beams three-dimensionally restrain the membrane against inflation caused by pressure. With the increase in pressure, the beams start their bending gradually. Length of the MBFS also gradually becomes longer because the beams' bending releases the restraint function. At the same time, energy of internal pressure becomes stored into beams in the form of elastic strain energy which was supposed to work on fabric as prestress. Finally, beams deform largely then gradually lose their restraining function. Thus, the length of the MBFS depends on pressure, and this property can be referred as a structural elasticity. Figure 3.10 and 3.11 illustrate elastic behavior of MBFS through implication of an equivalent spring model.

Figure 3.10 describes shapes of frames made of four beams before and after the bending deformation caused by pressurization. To understand internal stress which balances isotropic pressure, springs and restraint forces are introduced. Figure 3.11 is the description of equivalent spring model

of the MBFS developed from Figure 3.10. The model can be shown in two-dimensional view because the structure of the MBFS is axially symmetric. It is supposed that a piece of cable altered from membrane is connected with hinges, and a top and bottom of the model are ended with rigid bars. Half of left and right hinges are connected through springs. The springs correspond to the ones in Figure 3.10(a). The other hinges are connected to virtual wall forgiving axial slides with other springs which correspond to the ones in Figure 3.10(b). This spring explains bending elasticity of each beam.

In the unpressurized state the MBFS is completely folded (See Figure 3.9), and that condition is illustrated using spring model in the Figure 3.11(a). The beams work analogous to spring restraints inside and outside the structure. As the MBFS is pressurized, pressure is applied to cable and end bars. Spring restraints resist expansion and control the length of the MBFS as shown in Figure 3.11(b).

The length of the tube is controlled by selecting internal pressure and materials of beams. Also, the structure has capability to be a right-weight actuator inducing large stroke, because the length of tube can be controlled by the value of internal pressure. Once frames are arrayed anisotropically, the prospective actuator can behave in a three-dimensional configuration.

3.4.2 Bending properties of the pressurized MBFS

The results of the bending test shown as relationship between deformation angle ϕ and tip load F can be seen in Figure 3.12. The gradient of the curve in Figure 3.12 means equivalent bending rigidity of tubes. The data are the averaged values of three repeated tests. Figure 3.13 shows photograph of bending behavior of plain, ring restraint and MBFS type of tube from 0 to 7 kgf.

Plain and ring restraint type of tubes start bending slowly until the tip load reaches around 50 N. After that, both begin larger deformation. It is obvious that the work of the load up to 50 N is consumed to counterbalance prestress on fabric of tube. This can be explained by the propagation of wrinkle which indicates that membrane is not tensioned. Wrinkle on fabric is observed when the tip load reaches around 39 N as shown in Figure 3.14.

No difference is confirmed between plain and ring restraint type of tubes. Ring restraint type of tube had wire rings preventing circumferential expansion. The result shows that circumferential restraint does not affect bending property. On the other hand, the MBFS tube starts large bending as soon as the tip load is applied; thus, much less load is required for bending deformation. To bend the MBFS tube from 0° to 10° and 20° , for instance, 61% and 51% less load is required, respectively, in comparison with the plain type of tube. The slope of the plot for MBFS tube is almost linear and parallel to the slope of plain and ring restraint type over 50 N.

This trend can be explained by restoring force effect that is mentioned in the previous section. Since axial prestress induced by internal pressure is reduced, folding of membrane starts immediately resulting in tube's bending deformation.

Polygonal frames and induced folding crease also function as hinges. On the crease, membrane is easily bent resulting in losing rigidity. Therefore, membrane can be contracted with little applied

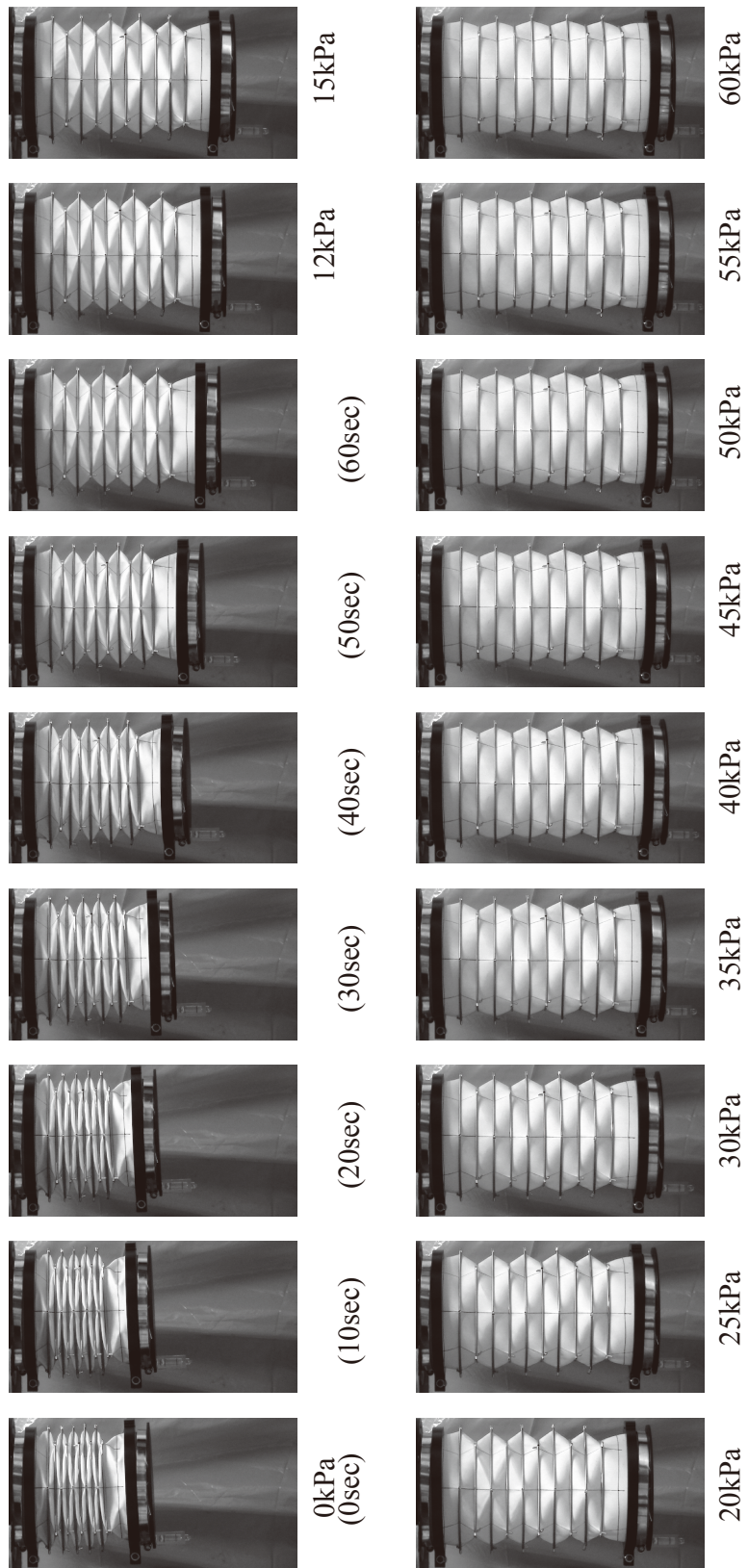


Figure 3.9. Axial elongation of MBFS with pressurization.

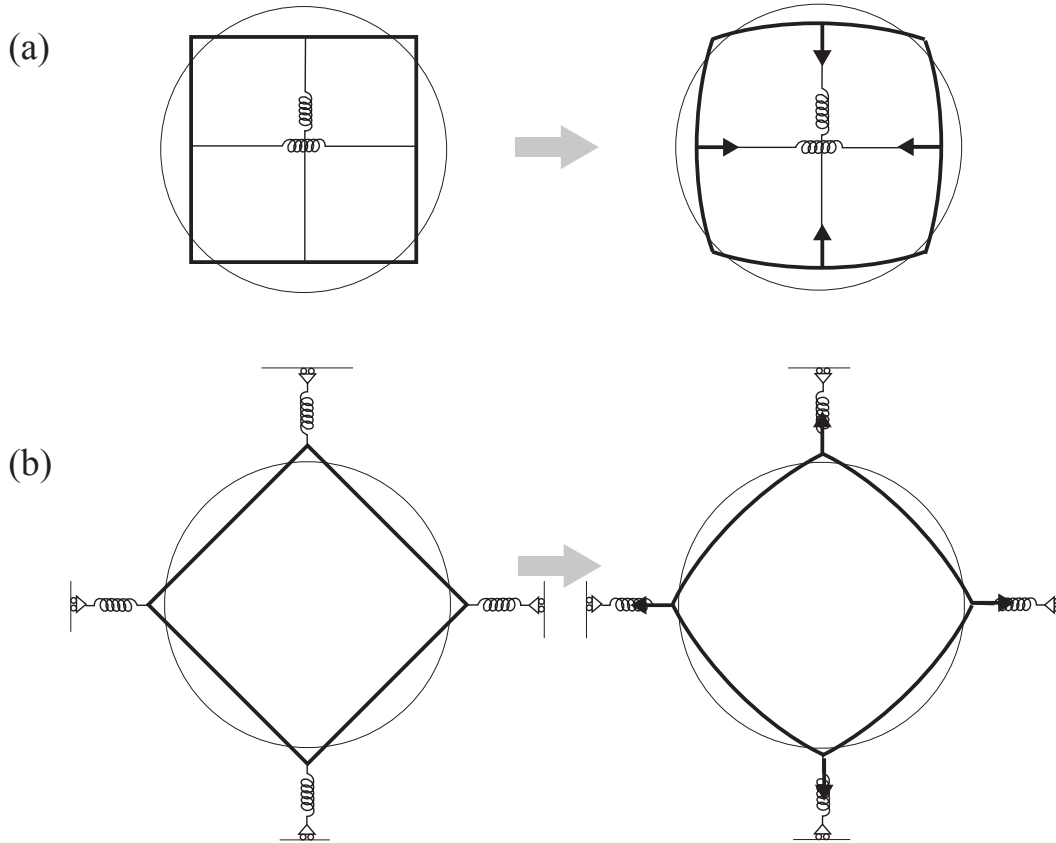


Figure 3.10. Spring model (top view).

force.

On the inner side of MBFS tube for bending, polygonal frames function as restraint along axial direction correctly explained with the restoring force, which helps contraction of skin. It should be noted that, however, to extend skin on the outer side of MBFS, polygonal frames should be bent. It means polygonal frames might hinder extension of skin on the outer side. The measured force may correspond to the force of bending polygonal frames on the outer side and the work for volume change described later.

Hysteresis is a common characteristic of fabric structures. When fabric is loaded, the individual yarns elongate and slide over each other, resulting in load-extension curves characterized by the yarn compliance and the friction between yarns. Frictional effects are particularly notable when fabrics are loaded biaxially and deform in shear, because this loading condition results in the most relative motion of yarns [46].

3.4.3 Growth of inner pressure during bending deformation

From experimental bending property given in Figure 3.12, bending rigidity for all type of tubes still remains above the critical load.

Pressure growth with an increase in deformation angle during the bending test is shown in Figure

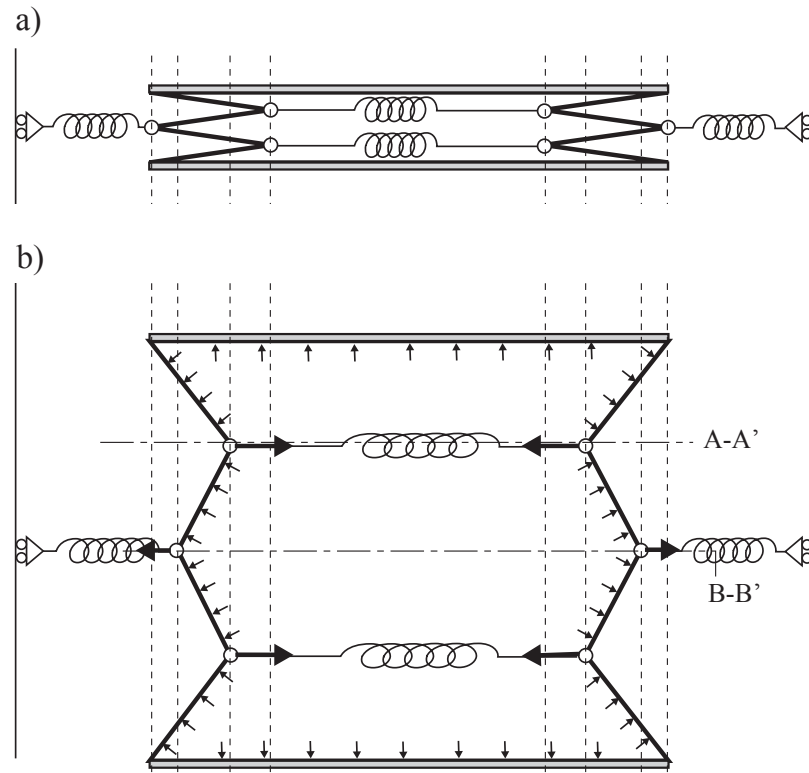


Figure 3.11. Spring model (side view).

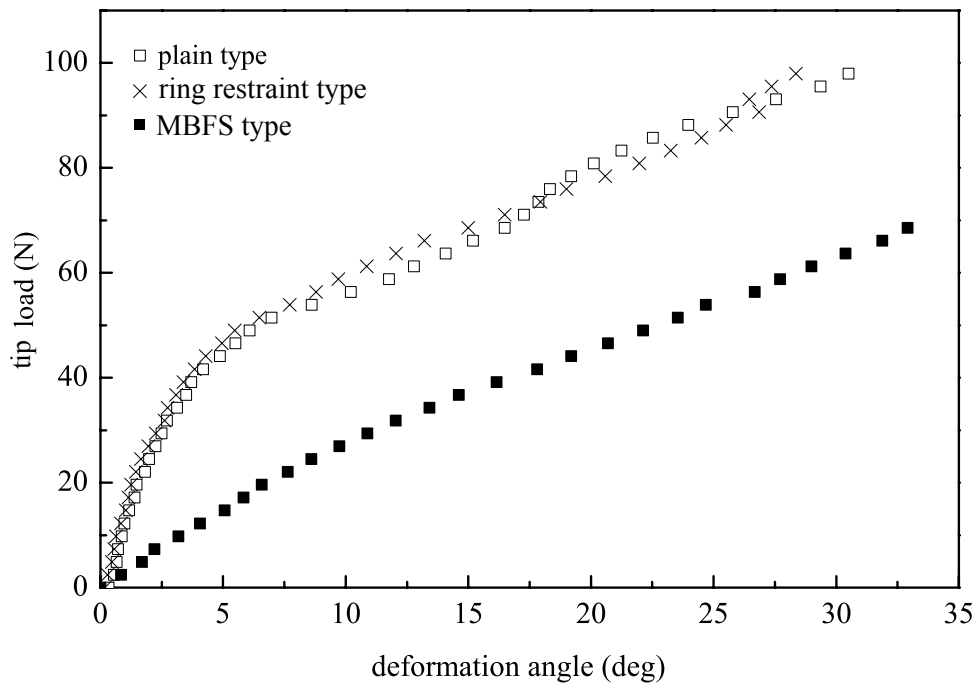


Figure 3.12. Relationship between deformation angle and tip load.

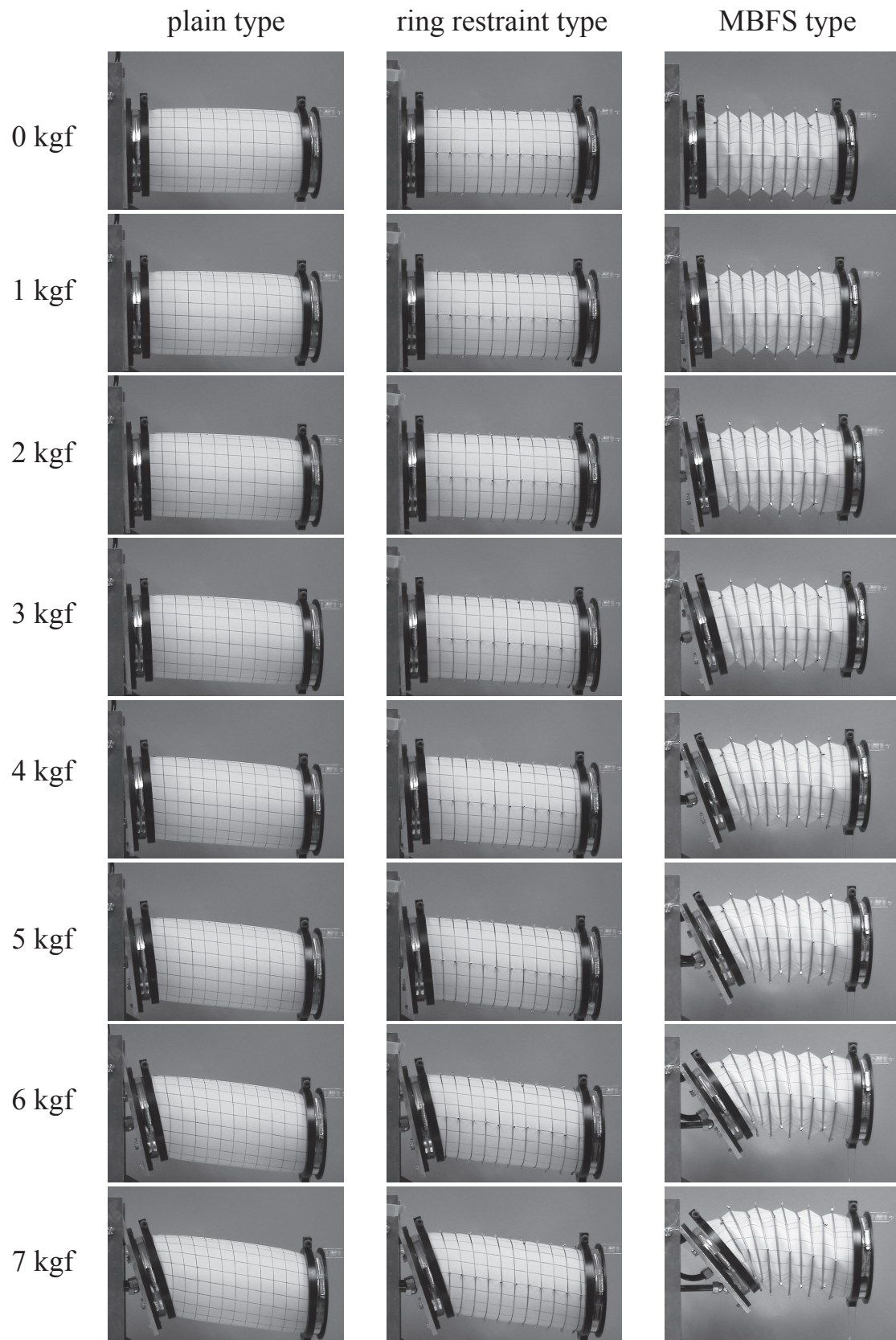


Figure 3.13. Comparison of bending deformation.

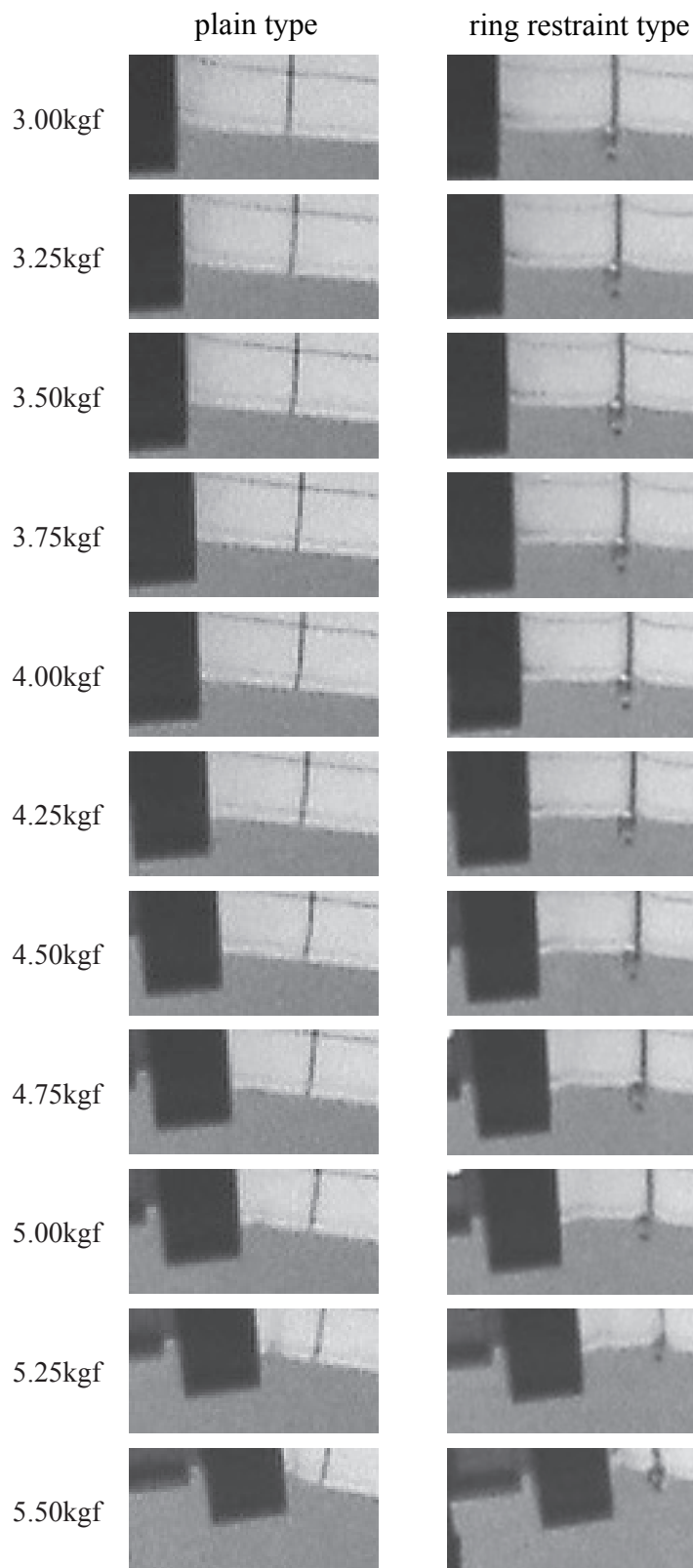


Figure 3.14. Development of wrinkle.

3.15. Though the internal pressure immediately rises for the plain and ring restraint type of tube, it rises more moderately in case of the MBFS tube. This difference indicates that MBFS tube has capability to maintain its volume. However, the capability may not contribute to higher bending property, since the gradient of slope for MBFS and plain type of tube in Figure 3.12 is not different from each other. More detailed work about dependence of bending property of the pressurized MBFS tube on internal pressure growth is necessary.

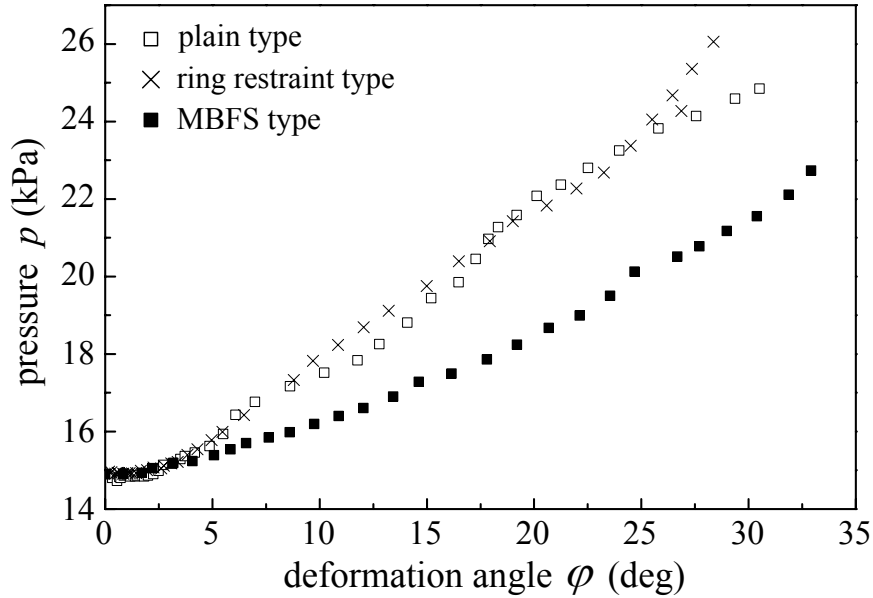


Figure 3.15. Relationship between deformation angle and internal pressure.

3.5 Numerical estimation

A numerical analysis is carried out to estimate how much prestress is counteracted by restoring force. The analysis is based on applied Euler-Bernoulli beam theory developed for pressurized beams [1, 20]. For simplification, three-dimensional geometry induced by frames is neglected, and instead, a simple cylinder shape is assumed as shown in Figure 3.16.

Take the x coordinate of the free end as 0, and y coordinate of the bottom of tube as 0. Axial stress resultant $N(x)$ on membrane at θ , when tip load F is carried, is written as follows. When wrinkle is not induced,

$$N = N_0 \left(\frac{1 + \cos \theta}{2} \right) + N_m \left(\frac{1 - \cos \theta}{2} \right) \quad (3.1)$$

Note that N_0 is the axial stress resultant on the top of the tube, and N_m is that on bottom of the tube. When wrinkle is propagated up to the region at θ_0 ,

$$\begin{aligned} N(x) &= \left(\frac{\cos \theta_0 - \cos \theta}{1 + \cos \theta_0} \right) N_m & \text{for } \pi > \theta > \theta_0 \\ N(x) &= 0 & \text{for } \theta_0 > \theta > 0 \end{aligned} \quad (3.2)$$

Then, restoring stress resultant σ is defined as the compression force divided by the circumference of the tube and is added to the equations of moment and force equilibrium at the free end.

$$M(x) = -2 \int_0^\pi (N + \sigma)r^2 \cos \theta d\theta \quad (3.3)$$

$$p\pi r^2 = 2 \int_0^\pi (N + \sigma)r d\theta \quad (3.4)$$

$M(x)$ is moment about neutral axis, r is radius of tubes, p is internal pressure. The internal pressure is assumed constant. Deflection differential equation is written as

$$\frac{d^2y}{dx^2} = \frac{M(x)}{EI} \quad (3.5)$$

Set of initial conditions are given; $dy/dx = 0$, $dy = 0$ when $x = L$ (the fixed end). Note that moment of inertia I is explained as

$$\begin{aligned} I &= r^3 \left[\pi - \theta_0 + \left(\frac{\sin 2\theta_0}{2} \right) \right] & \text{for } \frac{\pi}{2}(pr^3 + 2r^2\sigma) < M(x) < \pi pr^3 \\ I &= \pi r^3 & \text{for } M(x) < \frac{\pi}{2}(pr^3 + 2r^2\sigma) \end{aligned} \quad (3.6)$$

When wrinkle is exist, there is following relationship between θ and x .

$$\frac{Fx}{pr^3 + 2r^2\sigma} = \frac{\pi(2\pi - 2\theta_0 + \sin 2\theta_0)}{4(\pi - \theta_0) \cos \theta_0 + \sin \theta_0} \quad (3.7)$$

The initial-value problem, Equation 3.5, is numerically solved by the forth-order Runge-Kutta Method. The calculation condition is shown in Table 2.1.

Table 3.1. The calculation condition.

condition	value
dx (mm)	10
range of x (mm)	-238 – -1
step of tip load dF	1
range of tip load F (N)	1–80
modulus E (MPa)	333

The results of numerical calculations are presented in Figure 3.17, where relationship between deflection l (or $y_{x=0}$) and tip load F is shown for restoring stress σ varying from 0 to 471 N/m. Deflection l is defined as mentioned before in Figure 3.8 and regarded as magnitude of bending deformation. Although, deformation angle and defection does not necessarily have linear relationship, it is beneficial to compare experimental and calculated bending property in Figure.3.12 and Figure.3.17. As can be seen from Figure.3.17, when the restoring stress resultant σ is 0 N/m, in the beginning the curve linearly increases, and when tip load reaches around 50 N, then bending rigidity suddenly decreases almost into 0. This trend is consistent with the experimental results of the plain type of tube shown in Figure 3.12 except the gradient when the tip load is over 50 N. This

discrepancy will be discussed later.

As the restoring stress resultant σ increases, the critical tip load necessary to initiate large deformation decreases, and when restoring stress resultant reaches 471 N/m, the critical tip load becomes nearly 0 N. That initial deformation trend is in accord with experimental result of deformation property for the MBFS tube shown in Figure 3.12. This value, 471 N/m, is equal to that of calculated axial prestress on fabric induced by the initial internal pressure. Thus, it indicates that theoretically axial tension on fabric in the MBFS tube is mostly counterbalanced with restoring stress resultant σ , achieving axial tension free condition. In other words, the redistribution from axial tensile prestress on membranes to the bending stress on beams takes place, which gives flexibility for bending deformation to the pressurized MBFS tube.

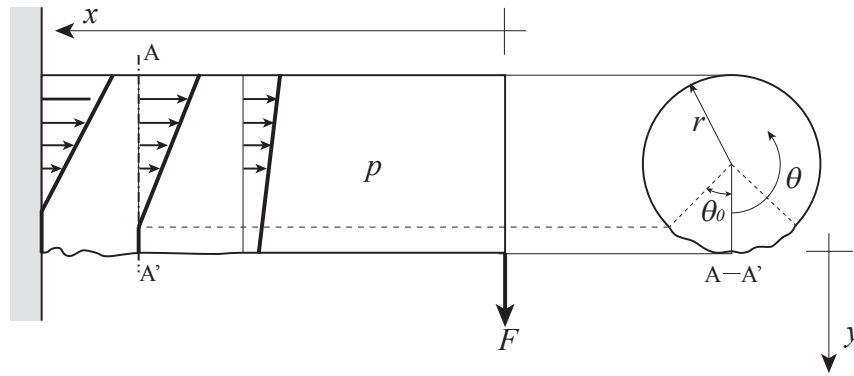


Figure 3.16. Schematic of cantilever pressurized tube. Tip load affects stress distribution on membrane, and slack region is propagated from base of the tube.

3.6 Further development

The present study dealt with the tube on which stainless steel frames are attached. Instead of rigid metal frames, if actuator material such as shape memory alloy (SMA) is attached on valley lines of the Yoshimura pattern, this MBFS structure is going to attain power-assist function.

SMA rods which are memorized to be linear shape are attached on valley lines of the Yoshimura pattern and are bent along tube's curvature when the tube is pressurized. So, the cross section of the tube is nearly circle, and the tube's configuration is straight. When a sensor detects one wants to bend a joint, SMA rods located at inner side of bending are selectively heated resulting in linear shape, then cross-section shape of the tube consists of several sides and arc. Thus, MBFS structure is induced only in inner side of bending, realizing contraction. Consequently, the tube changes to bending configuration without external load but with electrical or thermal energy.

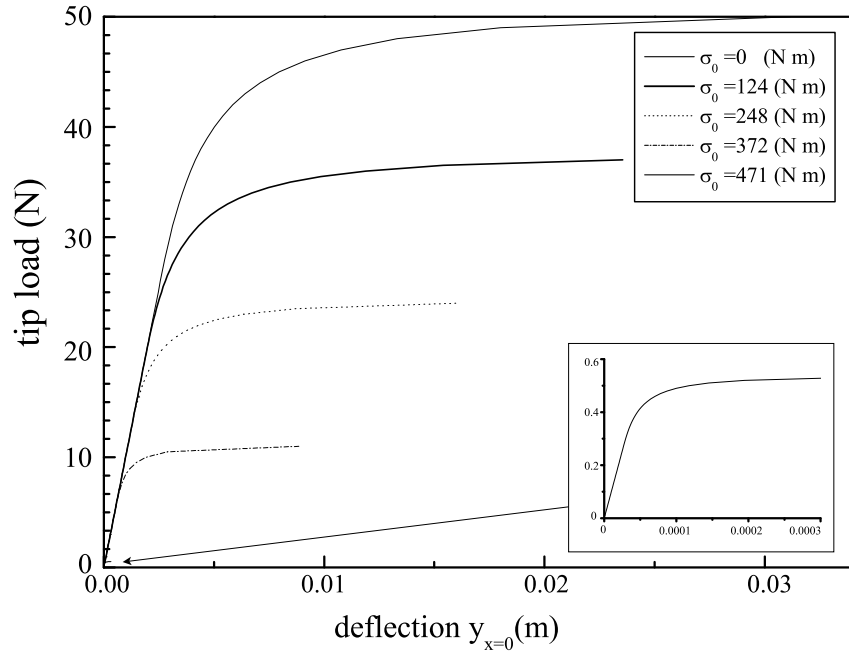


Figure 3.17. Theoretical bending properties in varying restoring stress resultant.

3.7 Summary

A newly designed variable shape pneumatic structure with transverse restraint designed with the aid of origami technology is proposed. The new structure, called Membrane Beam Folding Structure (MBFS), represents cylindrical membrane on which polygonal rigid frames are regularly attached to fold it axially. Axial elongation test and bending test are performed to obtain basic structural characteristics of the new structure. It is shown that application of the new structure provides more flexibility to the pneumatic tubes. Polygonal frames three-dimensionally restrain membrane of tubes, and that restraint effect induces equivalent axial restoring force when the tube becomes inflated. That nearly counterbalances the prestress on membrane due to pressure gap between inside and outside of the tube. The restraint effect help contraction of skin on inner side of MBFS. The mild increase of tip load changing with deformation angle is caused by pressure increase and bending force of outer side of rigid frames.

Chapter 4

Fabrication and Characterization of Isotenoid Excess Membrane Structure

4.1 Introduction

This chapter describes the second proposed design called Isotenoid Excess Membrane Structure. The main part of the joint is isotenoid which is intended to sustain excess material of garment by structural elaboration. The excess skin allows deformation of structure. Moreover, the proposed design has the axisymmetric shape, which takes advantage of maintaining volume. Experiment to investigate flexion property and related properties is conducted.

4.2 Overview of design

Diameter of cylindrical enclosure is arbitrary unless the value is greater than the height of the isotenoid. In this study, the diameter of cylinder is set to be equivalent to the height of the isotenoid. In this configuration, maximum designed flexion angle ϕ_{max} is 106.6° , at which the inner side of isotenoid is totally contracted, and roots of both cylinders contact each other.

The isotenoid enclosure consists of spindle-shaped flat gores as shown in Figure 4.1. Since a finite number of flat gores forms isotenoid, the formed shape is not exactly spherical, and isotenoid is constructed by the following approximation. The meridional curve of the isotenoid corresponds to a centerline of a gore, and width of the gore at distance y from the center can be described as

$$w(s) = \frac{2\pi R(z)}{n} \quad (4.1)$$

Note that n is nominal number of gores at each side (inner and outer side), which can cover surface of isotenoid without excess area. $R(z)$ is radius of isotenoid when height is z . In this paper, n is set to be 5 so that all cutting patterns of gore are designed identically.

The isotenoid enclosure is designed with an excess number of gores more than required for mini-

mum enclosure to form the lateral surface of isotenoid. The actual number of gores N at inner and outer side is the design parameter.

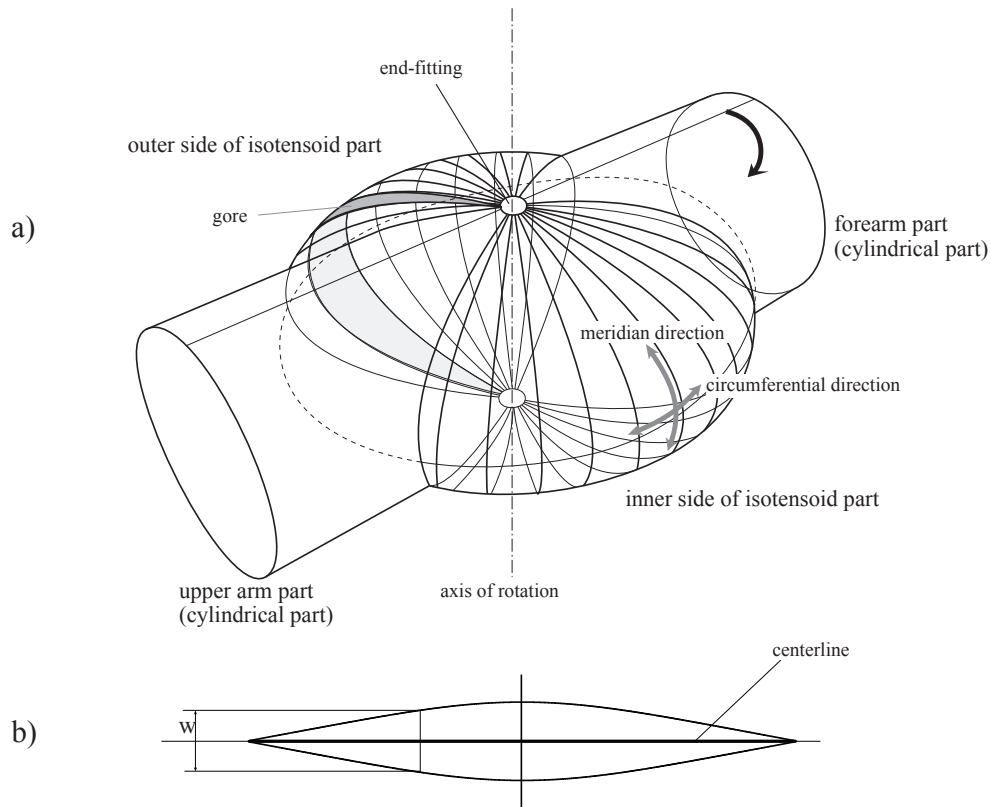


Figure 4.1. Schematic illustration of prototype; a) arm joint, b) geometry of a gore.

4.3 Numerical analysis for understanding isotenoid

4.3.1 Introduction

Numerical analysis is conducted to investigate equilibrium shape and shape stability of proposing design with excess skin. There is difficulty to deal with specific problems concerning the case which has excess skin in terms of several points; 1) handling thin-shell membrane and 2) modeling of initial geometry incorporating excess membrane.

As for 1), when one conducts a finite element analysis of thin-shell structures, employing membrane elements, in which bending stiffness are omitted from shell elements, is generally one of options to be applied. Compared to shell elements less number of degree of freedom allows efficient calculation for membrane elements. However, in case of three-dimensional large deformation analysis, the analyses often result in numerical instability, thus a little number of analyses has been reported. The instability is mainly caused by wrinkling phenomena. Membrane has little resistance against compression and thus easy to wrinkle. Wrinkling significantly affects both static and dynamic characteristics, and wrinkling gives stiffness matrix complicated singularity. Membrane elements are very sensitive against wrinkling, so conducting numerically stable analysis becomes difficult.

Conducting finite element analysis of thin-shell structures using shell elements also has been studied, which is called buckling analysis. The method allows to calculate detailed properties of out-of-plane deformation such as wavelength of wrinkling. However, the method inevitably required very fine meshing, thus calculation cost becomes significantly high.

As for 2), to incorporate excess membrane with initial geometry of three-dimensional model, expressing geometry of wrinkling or folding is very complicated and difficult.

4.3.2 Finite element method for wrinkled membrane

FEM based on Tension-field theory

To estimate stress field of partly wrinkled membrane, the tension-field theory have been developed [47]. In the tension-field theory, a thin membrane is idealized as a membrane with zero bending stiffness. Theoretically, such an idealized membrane does not sustain any compressive stresses. When compressive stresses are subject to be applied in the membrane, they are immediately released by out-of-plane deformations, that is, the membrane wrinkles. Wrinkled regions of the membrane, therefore, are supposed to be in uniaxial tension state, where stresses perpendicular to wrinkle lines equal to zero. While the tension-field theory cannot predict the details of wrinkles themselves, such as the amplitude and wavelength of the wrinkles, the theory is an efficient way to estimate macroscopic stress field of partly wrinkled membrane.

There are several methods to deal with partly wrinkled membrane in finite element analysis based on tension-field theory, and the most of method simulate wrinkling phenomenon by modifying stress strain relations. Miller et al. [48] modified elasticity matrix of wrinkled region to take into account for the tension-field responses. Liu et al. [49] proposed a penalty parameter modified material model, in which the stress-strain tensor of wrinkled membranes is modified via a penalty parameter. Although these modification schemes are easily integrated into finite element codes, the physical meaning of the modified stress-strain relations is somewhat obscure because the stress-strain relations of membranes do not change when wrinkling occurs. Meantime, Roddeman presented another model to modify deformation gradient tensor to account for the tension-field responses of membranes. Though other researchers took similar approach to improve, calculation procedures to obtain tangent stiffness matrix is complicated. Then, Nakashino proposed modification of the deformation gradient tensor in Roddeman model [50, 51]. The modification leads to the consistent tangent stiffness matrix and calculation of the consistent tangent stiffness matrix, simplifying the virtual work equation of wrinkled membrane, is much simpler compared to the previous approaches. Our numerical analyses as shown in the following sections employed the method developed by Nakashino [52]

Finite element methods for partly wrinkled membranes based on tension-field theory

The following is based on the work done by Nakashino [52]. First of all, the principal strain and principal stress are derived from Green-Lagrange strain and the second Piola-Kirchhoff stress obtained by conventional membrane theory. Then, according to the following criteria, determine

which stress field is correspond at objected region.

Stress field of partly wrinkled membrane can be divided into three regions: taut, slack and wrinkled. In taut region, minimum principal stresses are greater than zero. In wrinkled region, the minimum principal stress is equal to or lower than zero and the maximum principal strain is greater than zero. In slack region, the maximum principal strain is equal to or lower than zero

In case that the targeted stress field is taut, generate tangent stiffness matrix K and inner force vector Q . The procedure is totally same as conventional membrane theory.

In case that the targeted stress field is wrinkled, generate the tension-field equation, then solve θ . After stress-strain matrix C is modified, generate matrix K and vector Q as well as conventional theory.

In case that stress state is slack, membrane does not affect on stiffness of structure, thereby, tangent stiffness matrix K and inner force vector Q are set to be zero.

Conventional finite element methods incorporating geometrical nonlinearity is detailed in the literature written by Hisada and Noguchi [53].

4.3.3 Procedure of analysis

Procedure of analysis

This section outlines the procedure of analyses. The motivation is gained to make clear the equilibrium shape when the following condition is provided. The finite elements incorporating excess material help simplify modeling. Providing excessive material to curved surface requires several precursors modeling step which is not efficient and difficult to automatically apply every models when model changes. The initial shape has excessive skin although out-of-plane deformation such as wrinkling is not expressed.

- Length of meridian is fixed
- Length of diameter at any height can be freely changed

Investigation begins with the shape of a prolate spheroid, a spheroid in which the polar axis is greater than the equatorial diameter. By means of the special finite element, excess material at certain excess ratio in circumferential direction is given in order to provide freedom of diameter. Then homogeneous pressure is applied inside and equilibrium shape is solved. Excess material in circumferential direction is given to the spheroids by means of increasing excess ratio. The equilibrium shape is assumed to be isotenoid when sufficient excess material is given. The solver generates not only values of displacement of nodes but also values of principal stresses and wrinkles. These dataset enables to analyze the stress distribution in membrane.

By using a pre-post processor, three-dimensional models are designed at first. Then, mesh is generated to discretize the material into finite elements. After that, material properties, such as Young modulus, Poisson ratio and thickness, and constraint conditions and loading conditions are provided, all the data, coordinate values of nodes, relations between nodes and elements, and the

condition data are provided to the solver. The solver solves static analysis problem, and generate data output such as displacements at all nodes, principal stresses, Mises stresses and wrinkles at all elements. To see the results, the pre-post processor is used again to visualize distributions of results.

Software GiD version 8 for the pre-post processor is employed and the solver developed by Nakashino [52] was run with TSUBAME2, the latest supercomputer operated in Tokyo Institute of Technology. The detail about special finite element model in which material excess is embedded are described in Appendix II.

Models

Several configurations are modeled: Model 1 to 5. All models are revolutions of an ellipse. Equations of ellipse of each model are as follows.

$$\text{model 1 : } \frac{x^2}{0.0100^2} + \frac{z^2}{0.0635^2} = 1 \quad (4.2)$$

$$\text{model 2 : } \frac{x^2}{0.0300^2} + \frac{z^2}{0.0635^2} = 1 \quad (4.3)$$

$$\text{model 3 : } \frac{x^2}{0.0635^2} + \frac{z^2}{0.0635^2} = 1 \quad (4.4)$$

$$\text{model 4 : } \frac{x^2}{0.0635^2} + \frac{z^2}{0.0100^2} = 1 \quad (4.5)$$

Figure 4.2 shows the meridian profile of each model. Taking account for symmetry, 1/4 part (90 °) of a revolution is the targeted object. Excluding model 4, all models have the same height, which is identical to the design of experimental model in Chapter 3. Z-axis is rotational axis. Rotation is -45 ° from x-axis to +45 °.

Calculation conditions

The same constraint condition is applied to all models: the displacement of node at the bottom vertex is set to be zero, and the x- and y-component displacements of the top vertex are zero, i.e. the top vertex is free along z-direction. The both side edges are constrained to only allow displacement in the plane rotated by -45 and 45 ° from xz-plane.

Modeling fabric as isotropic membrane assumes that the Young modulus and the Poisson ratio are constant for a given material. Young modulus E is 7.2 GPa, Poisson ratio ν is 0.3, thickness is 190 μm . These property is identical to the material used for the experiment conducted in Chapter 3.

Here, The value of differential pressure along the envelope is another parameter as a loading condition. Excess ratio in circumferential direction B is defined, which is an important parameter for the analysis. The excess ratio B is defined by the following equation. The state at $B = 1.0$ is the nominal state which does not have excess material on membrane.

$$B = \frac{L + L_0}{L_0} \quad (4.6)$$

where L_0 is the length of fictitious non-excess skin and L is the length of the skin including excess skin.

Linear triangle element is adopted for discretization. Model 1 to 5 are separated to 4076, 3334,

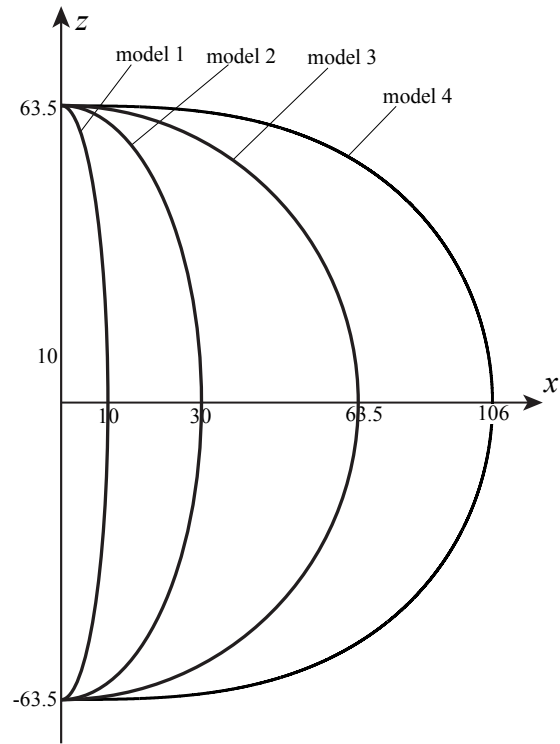


Figure 4.2. Initial meridian profiles of each model.

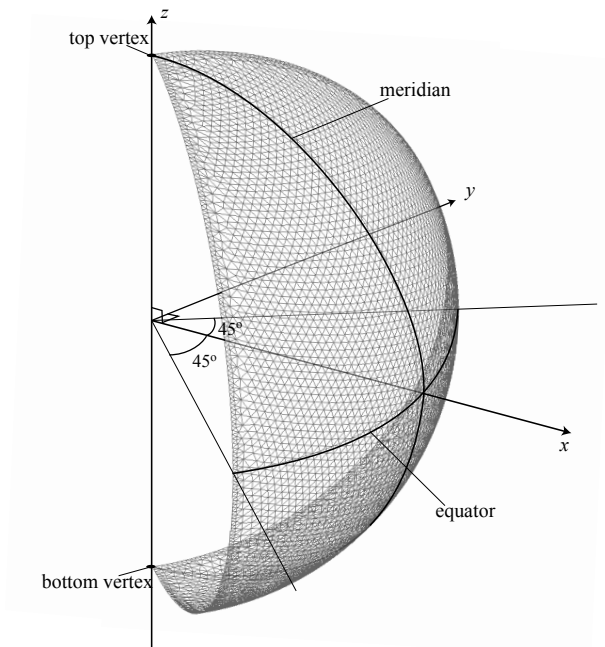


Figure 4.3. Schematic of three-dimensional model (the model shown is Model 2).

9040, and 6208 elements, respectively.

Convergence criteria are defined by the value of residual error during iteration of Newton-Raphson method. When the value of residual error falls below the specified value within maximum number of iteration, the solution is assumed to be converged.

4.3.4 Dependency of equilibrium shape of revolutions on excess skin ratio

Numerical calculation was carried out to estimate equilibrium shape of each model when circumference is extended by increasing excess ratio B under differential pressure.

Figure 4.4 shows sequential view of cross section on x-z plane with increasing excess ratio B at 100 kPa of the constant differential pressure P . The meridional profiles are generated by cutting the deformed shape of each model. It is clear that as excess ratio B increases, the length of equator becomes larger. Thereby, the diameter at each height becomes larger accompanying meridional profile becomes flat gradually. The shape maintains symmetry and equator is able to be regarded as an arc with constant curvature. Then, when the excess ratio reaches the value at around 4.9, meridional profile mostly identical to Euler's elastica mentioned in section 2.6.1. When the excess ratio is set over around 4.9, symmetry of the entire shape was broken and distortion occurred. Figure 4.5 shows transition of deformed shape from stable to unstable geometry at 100 kPa for model 1. Membrane at top vertex becomes twisted, and the phenomenon develops more and more as excess ratio increases. Model 2, 3 performed mostly same characteristics as shown in Figure 4.4.

Those results suggest that right after the meridian profile becomes similar to Euler's elastica the shape becomes unstable. The critical excess ratio is called the maximum excess ratio B_{max} . Table 3.1 shows comparison a is x-component of meridian profile of each model when $z=0$, R is the one of Euler's elastica of which length of meridian profile S is identical to each model. Assuming equator is arc, length of Euler's elastica should be $\pi R/2$, length of equator of each model should be $\pi a/2$. Thus, a/R is the amount of required extension rate of equator from each model to Euler's elastica. By comparing the value with B_{max} , B_{max} is mostly agree with a/R for all models.

Table 4.1. Comparison of critical excess ratio B_{max} and extension ratio of equatorial radius a/R .

	model 1	model2	model3
a	0.0100	0.0300	0.0635
R	0.0501	0.0578	0.0761
a/R	5.01	1.93	1.17
B_{max}	4.90–5.00	1.80–1.90	1.20–1.30

The characteristic is not totally true depending on the differential pressure. In case of model 1, when the pressure is under approximately 80kPa, as excess ratio increases, the convergency becomes worse and is not achieved before excess ratio achieve the stable limit. The Figures 4.6, 4.7 and 4.8 show the maps in which whether the solution is converged or unconverged and stable or unstable at each excess ratio and pressure for model 1,2 and 3 .

4.3.5 Shape stability of isotenoid with excess skin

To investigate the reason why Euler's elastica becomes unstable, stress distribution is analyzed. Figure 4.9 shows sequential view from y-z plane by excess ratio showing taut region where wrinkle is vanished and biaxial stress is induced by blue color. When excess material in circumferential direction is not provided ($B = 1.0$), whole region is taut. When excess material is provided, taut region start vanished from both polar region, then taut region forms a band around equator region. As excess ratio increases, wrinkle develops from polar region gradually. In other words, the band becomes narrower, and when excess ratio is 5.0, the band of taut region vanishes completely. Simultaneously, the shape becomes unstable. As for model 2 it is confirmed similar phenomena. The result indicates that vanishing taut region around equator results in unstable unsymmetrical shape formation. Namely, pure uniaxial meridional stress field is not sufficient to maintain axisymmetric shape.

That discussion suggests that to achieve more flexible isotenoid excess membrane Structure, attaching restraint on equator is very effective improvement plan to maintain stability of shape.

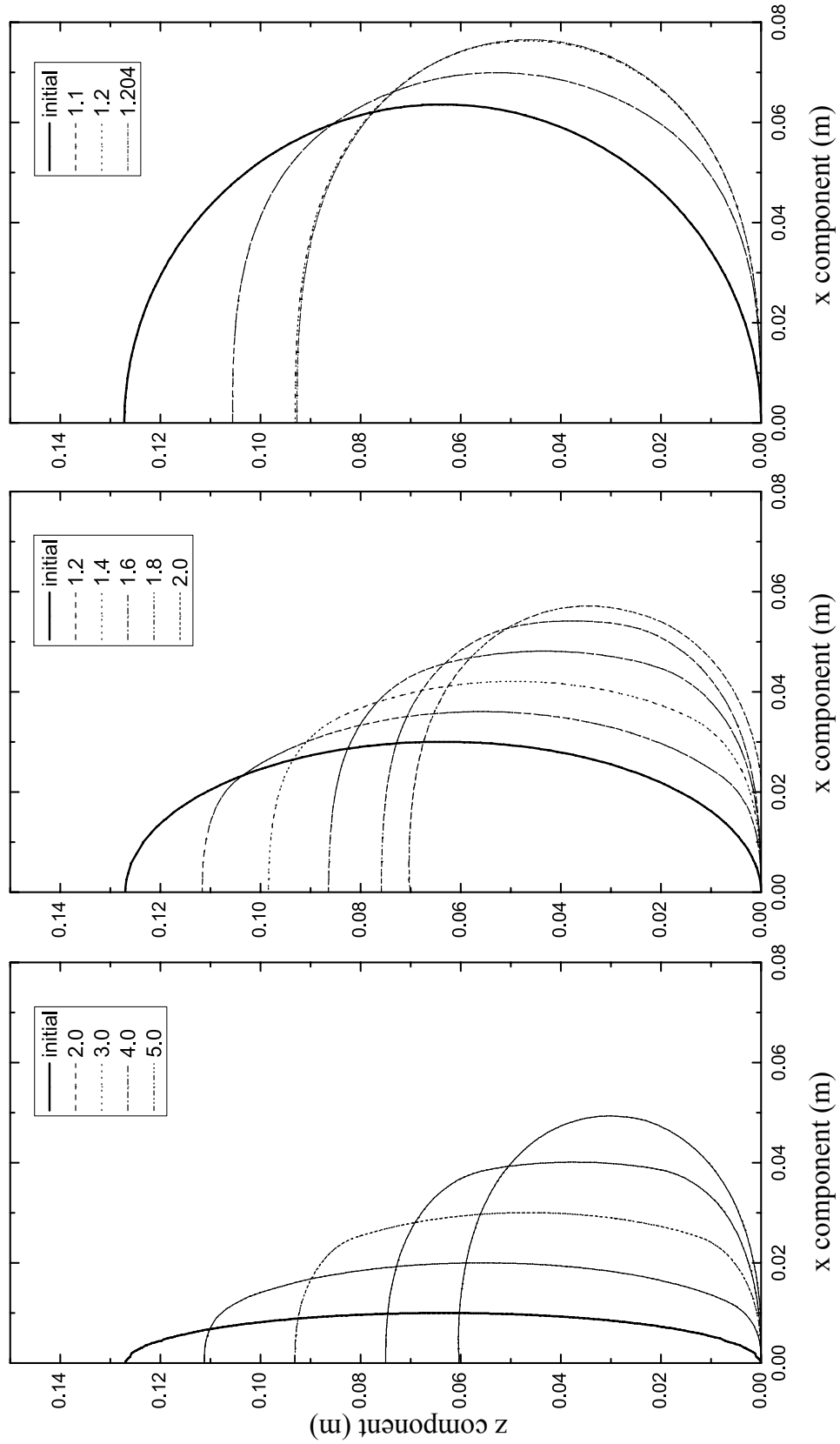


Figure 4.4. Equilibrium meridian profile, Model 1, Model 2, Model 3 (z-component of nadir is set to be zero).

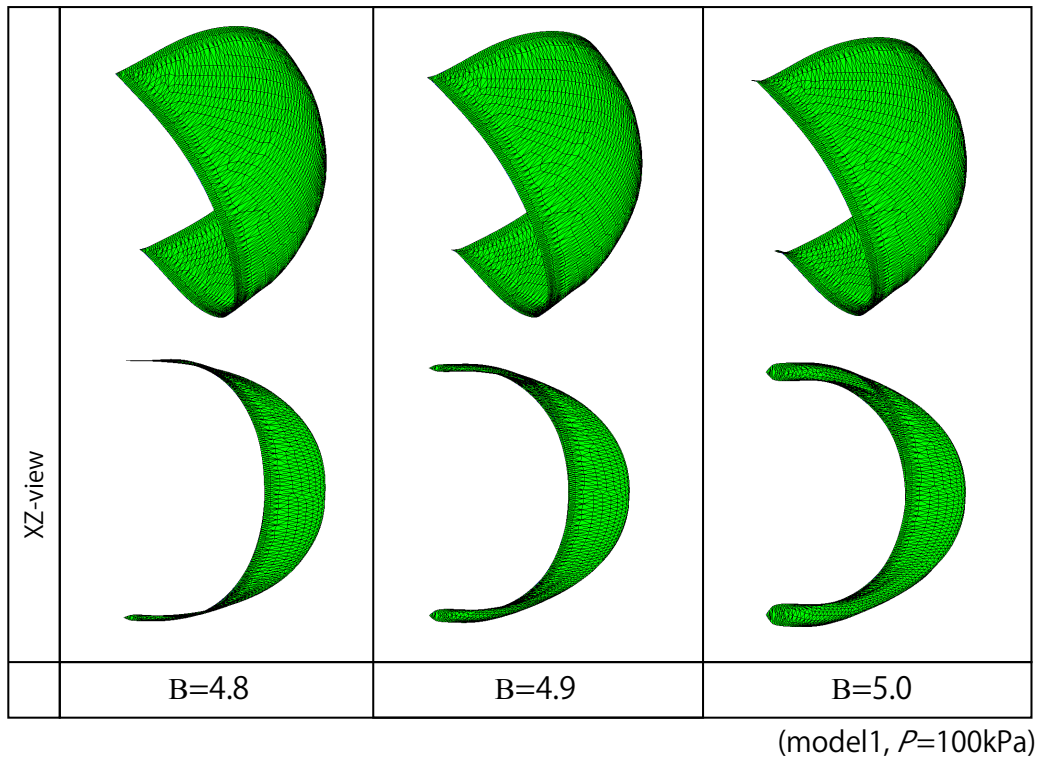


Figure 4.5. Sequential view of equilibrium shape for Model 1 at 100 kPa of pressurization. Transition from stable deformation to unstable deformation.

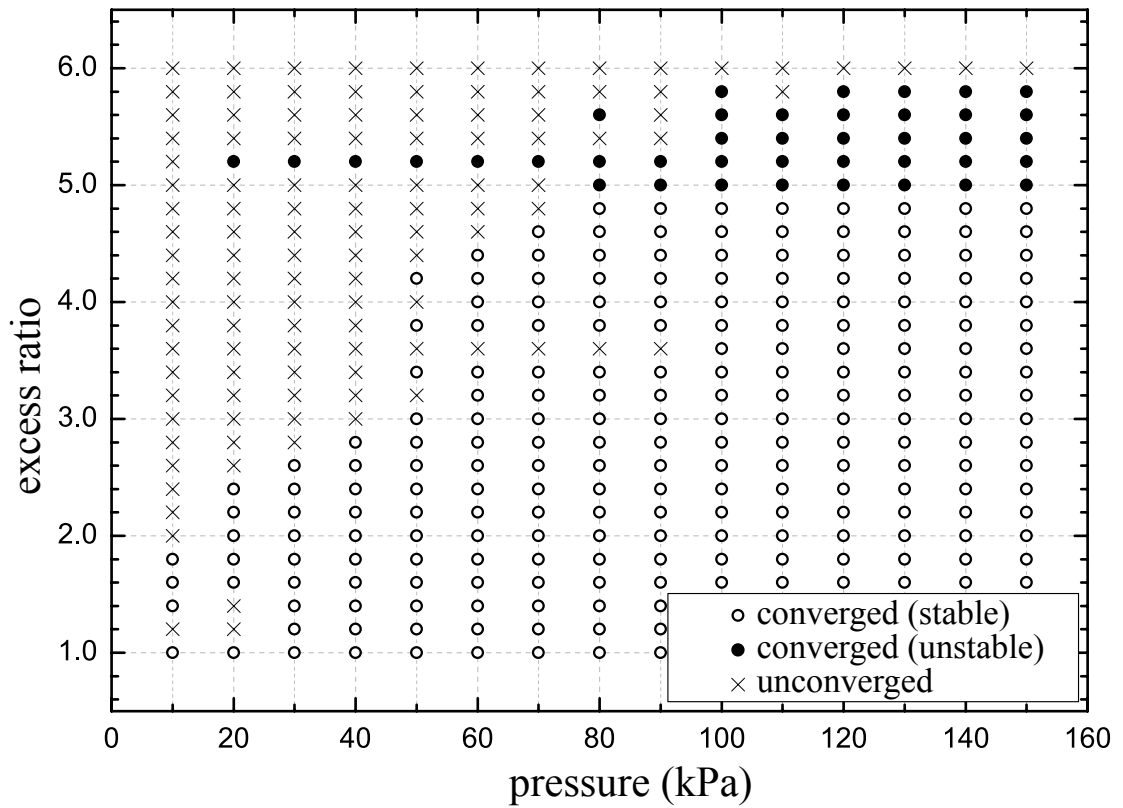


Figure 4.6. Map of convergency and stability at each excess ratio and pressure for Model 1.

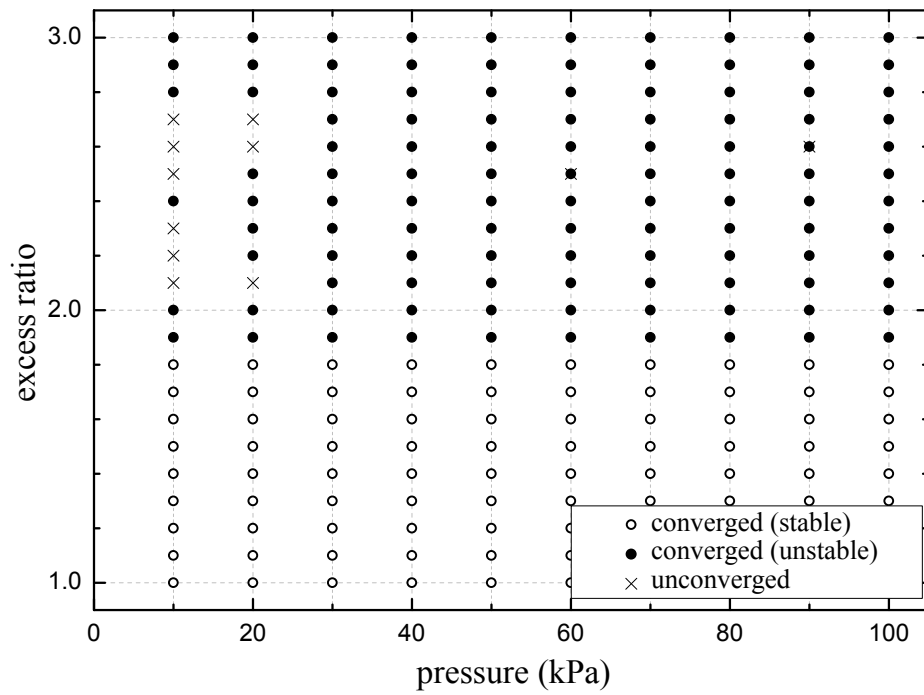


Figure 4.7. Map of convergency and stability at each excess ratio and pressure for Model 2.

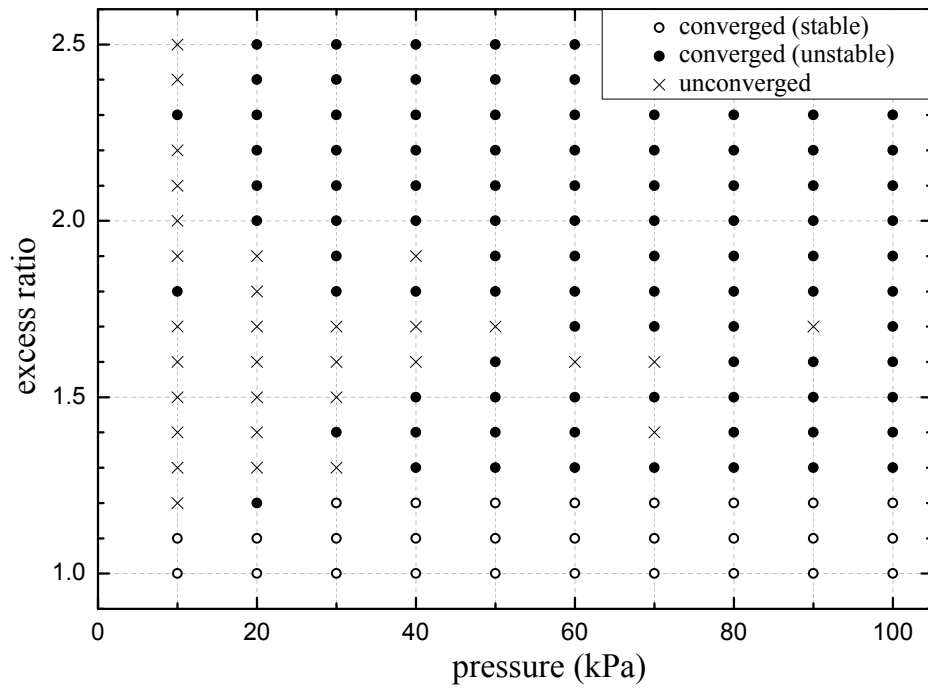


Figure 4.8. Map of convergency and stability at each excess ratio and pressure for Model 3.

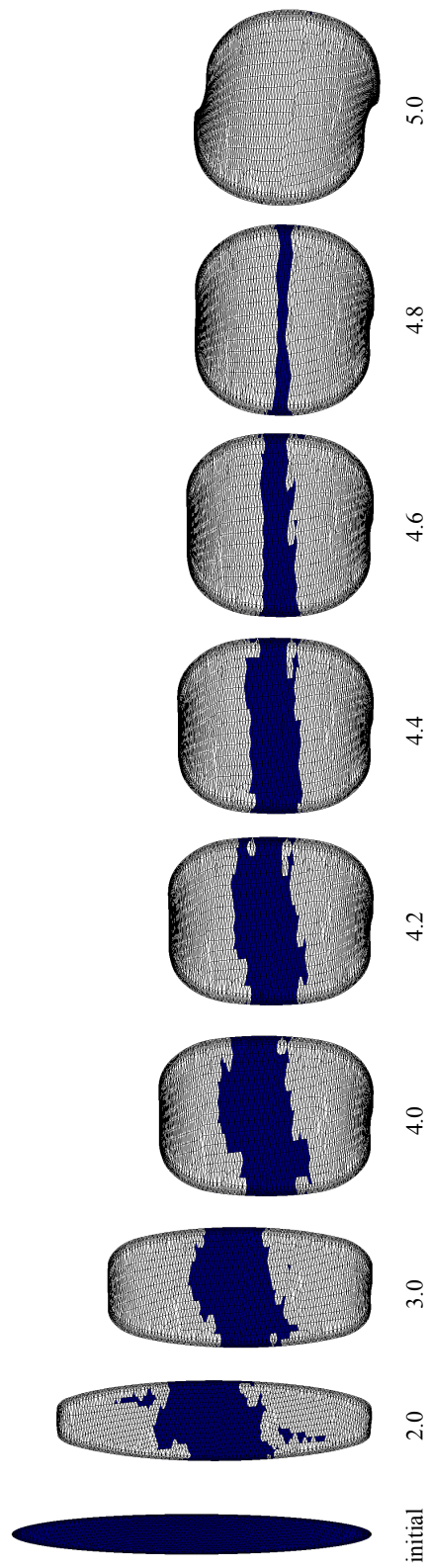


Figure 4.9. Taut region dependency on excess ratio B for Model 1.

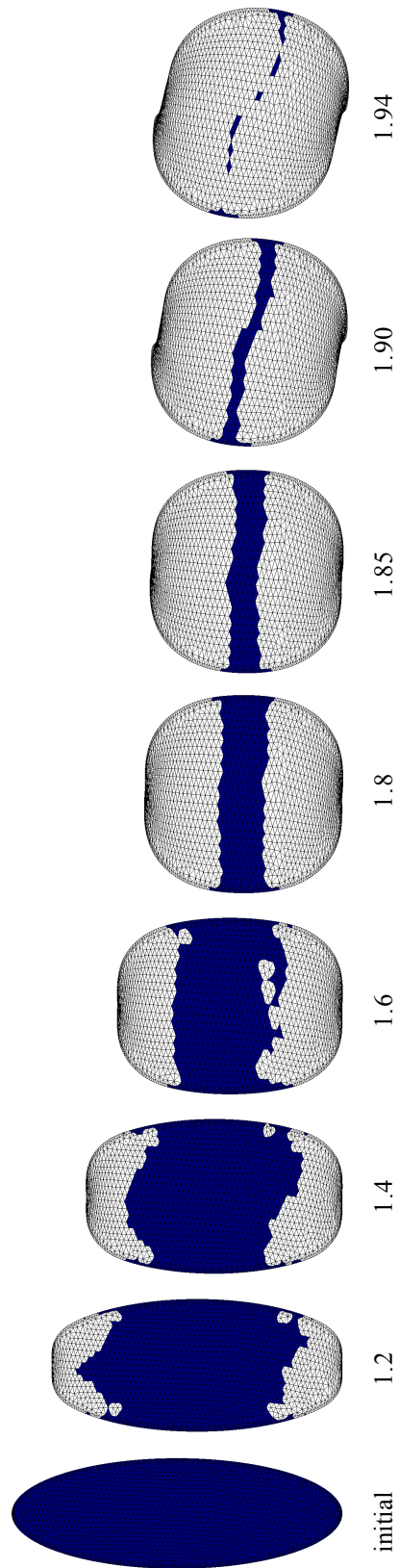


Figure 4.10. Taut region dependency on excess ratio B for Model 2.

4.4 Experimental

4.4.1 Fabrication of joint prototype

Geometry

Geometry of prototype of joint is shown in Figure 4.11. Diameter of the cylinder as well as height of isotenoid is 127 mm, and the distance from axis of rotation to the end of cylinder is 247 mm. The radius of the isotenoid at equator is 106 mm. Two types of prototype depending on number of gores were prepared: N9 and N11. In case of N9, nine gores are connected together at each side, and isotenoid enclosure at outer side is fully extracted at 85.3° of flexion angle, which is smaller than maximum designed flexion angle ϕ_{max} . In case of N11, eleven gores are connected and fully extracted at 127.8° which is larger than ϕ_{max} .

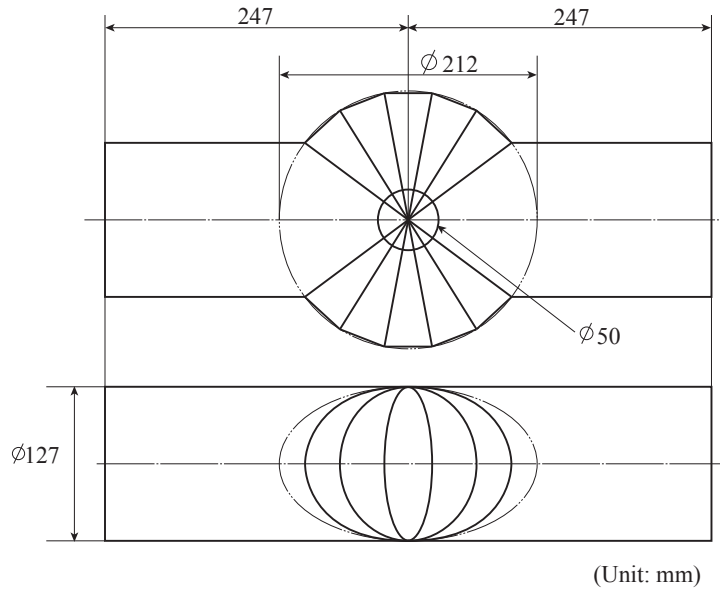


Figure 4.11. Geometry of the arm joint.

Materials

Prototypes of arm joint were manufactured with single laminated material manufactured by Kuraray Co.; a base layer represents woven fabric of liquid crystal polymer fiber, Vectran HT-0255, and a coating layer is silicone rubber. Vectran HT-0255 base fabric is 55×55 plain weave and 0.19 mm thick. It offers 653 N/cm of resultant strength which yields required strength. Vectran has excellent abrasion resistance and has found application in space inflatable structures, such as Mars Pathfinder's landing airbags. Laminate layer of silicone rubber shows high gas-retention properties at a low thickness, it was readily applied to fabrics and lends itself well to the assembly process. Any leak was not confirmed at leakage test up to 30 kPa.

Silicone rubber compound, KE-3479 manufactured by Shin-Etsu Chemical Co., was selected for sealing stitch lines. The silicone rubber demonstrated excellent adhesive property to silicone laminate

layer and showed high elasticity after curing. Any other material was not used for the fabrication.

Mechanical property of materials

To measure mechanical property of Vectran fabric, tension test was conducted prior to the experiment. Two types of specimen from Vectran fabric were prepared: one with 3 threads and the other with 10 threads. Each type has three or four specimen prepared by totally same conditions to prove repeatability. Clamp distance was 30 mm for 3 threads, 60 mm for 10 threads. Since the fabric performs high strength, it was observed that specimen with 10 threads was slipped with jigs when certain tensile force was applied in preliminary experiment. Thereby, the tests are separated to two types: whole observation from start to break and elastic modulus measurement with higher precision. The test pieces with 3 threads are to observe elongation until break point, the other piece is for measure elastic modulus precisely. The stress machine used was EZ-S 500N manufactured by SHIMAZU, and all tests were conducted with 20 mm/min of elongation velocity.

Figures 4.12 and 4.13 show force-strain diagrams of 3-thread specimens and 10-thread specimens, respectively. It is noticed that the gradient of curves change drastically when strain reach 2–3%. It is assumed that sagging of fibers disappeared at the strain, then material itself start elongation from the strain. Modulus, according to 10-thread specimen, is derived by linear regression between 100 N and 200 N and is taken average from three repeated tests. The value is shown in Table 3.2. The fabric has 55 threads per inch and thread has 0.19 mm of averaged thickness. Equivalent modulus is derived regarding as a homogeneous film.

Tensile strength and tensile elongation are derived by 3-thread specimen tests.

Table 4.2. Mechanical property of Vectran.

	3 threads		10 threads	
	warp	fill	warp	fill
Modulus [N/thread]	N/A	N/A	648	621
Modulus [GPa]	N/A	N/A	7.39	7.08
Tensile strength [N]	152	124	N/A	N/A
Tensile elongation	0.076	0.089	N/A	N/A

Fabrication process

Details of the prototype were cut according to the cutting pattern and connected by sewing. Two cylindrical parts corresponding to forearm and upper arm were made of one flat fabric sheet each. The overlapping area of the sheets with isotenoid part was cut off. Then, rest of sheet was rolled up, and both edges were sewn together. Gores were sewn each other for inner and outer side, then each edge of connected gores was sewn with the edge of cylindrical part. Then, surface around stitch line was degreased by a small amount of ethanol, and KE-3479 was put along stitch lines and placed for 3 days at room temperature for curing.

It is important to note that end fittings capped on vertices were specially designed to allow rota-

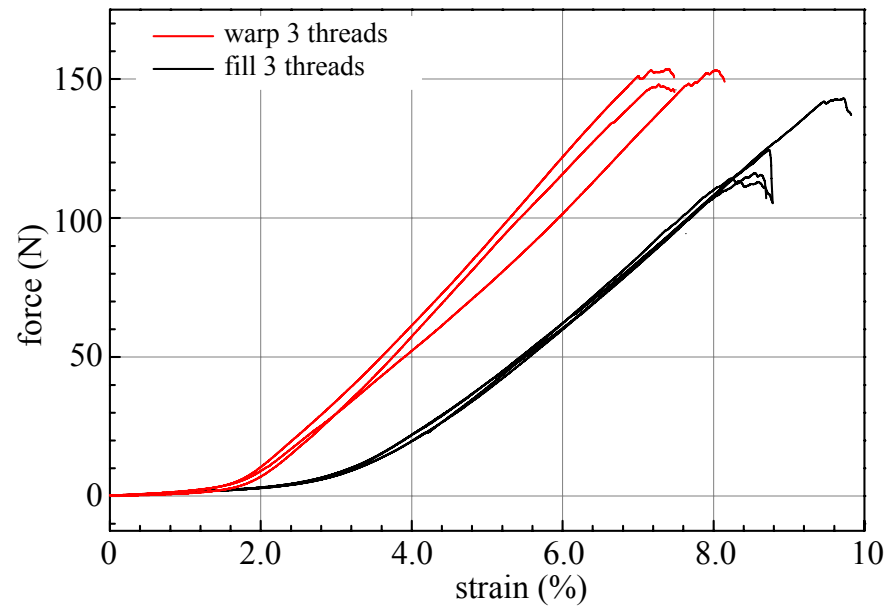


Figure 4.12. Force-Strain diagram of Vectran warp and fill direction with 3 threads

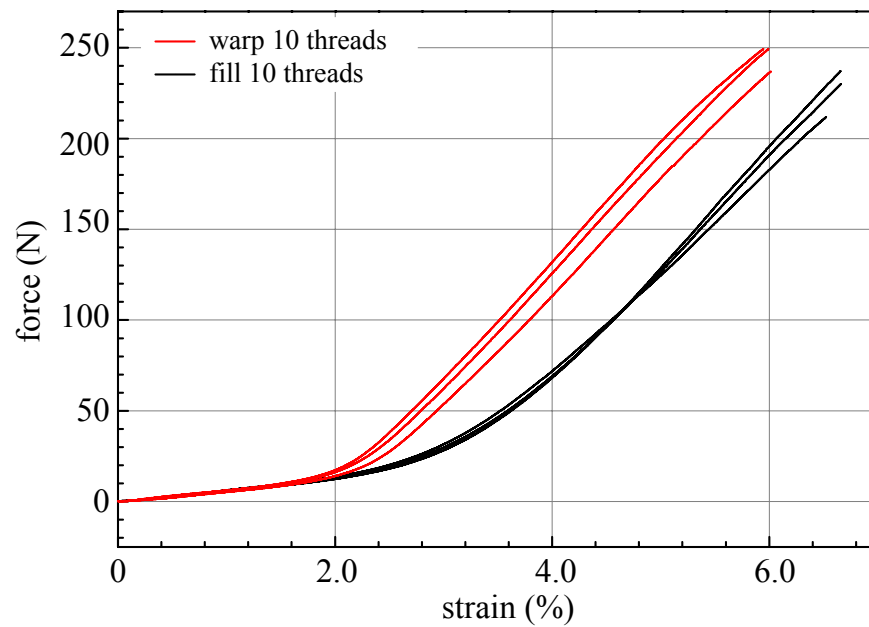


Figure 4.13. Force-Strain diagram of Vectran warp and fill direction with 10 threads

tional degree of freedom maintaining air-sealing. If the geometry of end-fitting is just like a disk, it would fix rotational movement of the joint. Thus, the end-fitting was made from two major circular sectors, which were connected to each other to make circle with excess skin that corresponded to excess skin of gores.

4.4.2 Experimental procedure

Instrument

To measure static torque (resistance to rotation about the joint's axis) trajectory during flexion and to investigate possible causes of torque change, such as interior pressure variations, the custom-made single-axis robotic arm was designed and built. The measuring system is aimed at testing bending property of single-axis joints, such as elbow, knee and finger. It acquires reproducible and quantified data. Using the system the torque required to flex the joint from the inside of suit can be measured, so approximation how the suit subject flexes the joint, can be obtained. The inside of prototype was pressurized up to defined pressure, while the instrument was located under atmospheric pressure, which simulated pressure difference between spacesuit and outer space.

Schematic illustration of measuring instrument is shown in Figure 4.14. Robotic arm's motion is accomplished through transferring linear movement powered by hydraulic linear actuator to rotational movement using slider-crank mechanism composed of forearm assembly (2), connecting rod (7) and link (8). Flexion angle ϕ is measured by rotary encoder (5), MA-20 manufactured by Microtech Laboratory Inc., which has 1024 per rotation of resolution and is installed on rotational axis. Interior pressure P_{in} is measured by the pressure gauge (9) installed at the inlet of compression air from compressor (14). Resistance torque of the joint T is acquired by transferring force data which is measured by loadcell (4), LUR-A-SA1 manufactured by KYOWA Electronic Instruments Co., installed at the connection part between linear actuator and slider-crank component. Note that since direction of gravitational force is aligned to the axis of arm rotation, and the plane passing through arm flexion is horizontal, measured torque does not include gravity effect.

Schematic illustration of the slider-crank mechanism is described in Figure 4.16. Relation between force induced by actuator F_{in} and generated torque T is depicted as follows.

$$T = \frac{\sin(\alpha + \beta)}{\cos \alpha} \cdot F_{in} \cdot \overline{OC} \quad (4.7)$$

$$= l_L \cdot \left(\sin(\phi + 15) + \frac{l_L}{2} \cdot \frac{\sin 2(\phi + 15)}{\sqrt{l_C^2 - l_L^2 \sin^2(\phi + 15)}} \right) \cdot F_{in} \quad (4.8)$$

α and β are the angles between connecting rod and horizontal axis, link and horizontal axis, respectively, l_L and l_C , are the length of the link and length of the connecting rod. Longitudinal axes of link and forearm assembly are tilted at 15° relative to each other to shift the dead center from 0° of flexion angle. Range of flexion angle ϕ is from -10° to 125° . The maximum force F_{in} induced by actuator is 2814 N. Calibration is performed appropriately.

It should be noted that this method does not simulate human body's arm surface profile and accu-

rate joint rotation. For instance, human elbow joint is not exactly located at the center of arm width. Therefore, this measurement does not estimate internal contact and friction between joint garment and skin. In addition, since an actual spacesuit is not separated at the end of arm, the interior volume change in this measurement tend to be exaggerated. Further investigation is anticipated to estimate how such factors affect joint mobility.

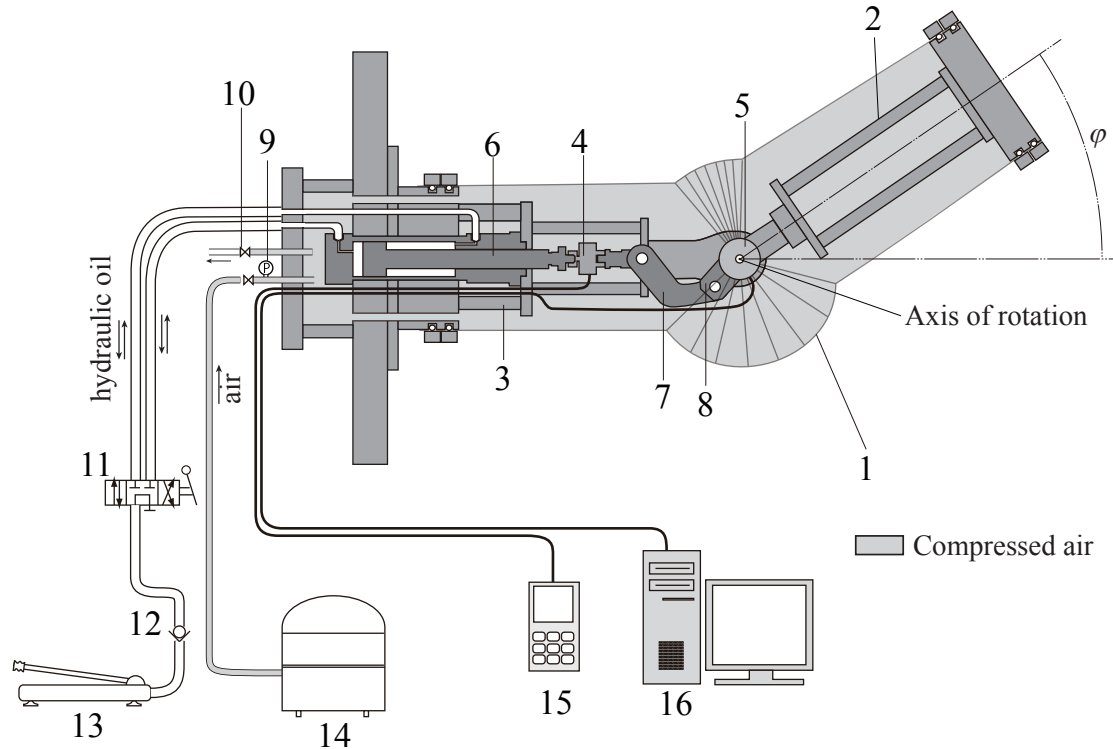


Figure 4.14. Schematic illustration of measuring instrument. 1: Prototype of arm joint, 2: Forearm assembly, 3: Upper arm assembly 4: Loadcell, 5: Rotary encoder, 6: Fluidic actuator, 7: Connecting rod, 8: Link, 9: Pressure gauge, 10: Stop valve, 11: Control valve, 12: Leak valve, 13: Pump, 14: Compressor, 15: Data logger, 16: PC.

Procedure

Measuring procedure was as follows. A joint prototype was mounted on the robotic arm. The center of isotenoid and rotation axis of robotic arm was aligned. Then, both ends of joint prototype were fixed and sealed by liquid gasket and O-ring. After that, initial flexion angle of robotic arm was set at 0° . The robotic arm was subjected to quasi static rotational motion from 0° to 120° and then back to 0° . During the motion joint angles and torques were recorded with 5° step. Measurement was also conducted without mounting the prototype of arm joint. The difference between torques measured by the robotic arm in suited and unsuited states represents the torque required to flex the joint prototype.

Two modes were defined: mass-constant mode and pressure-constant mode. In case of mass-constant mode, valves at inlet and outlet of compressed air were closed during the following mea-

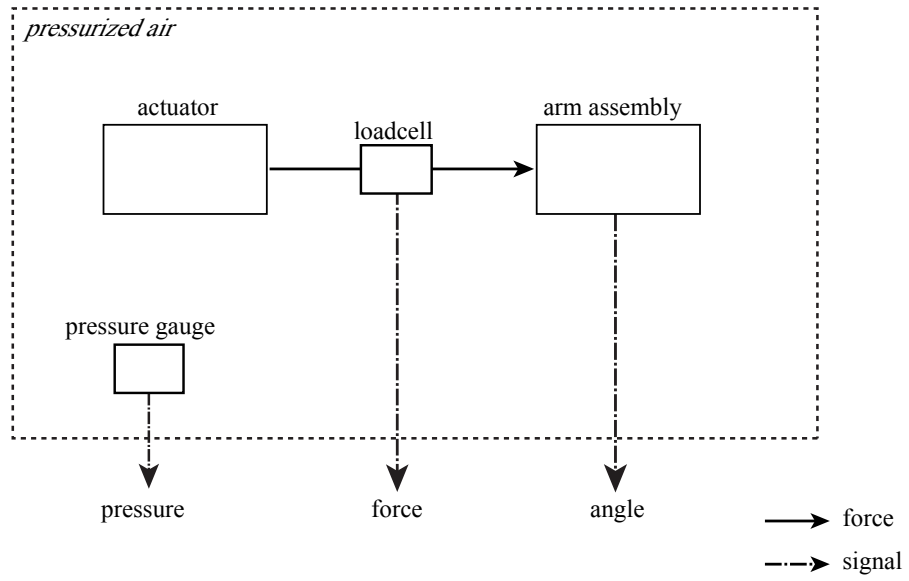


Figure 4.15. Diagram of data acquisition process of the measuring instrument.

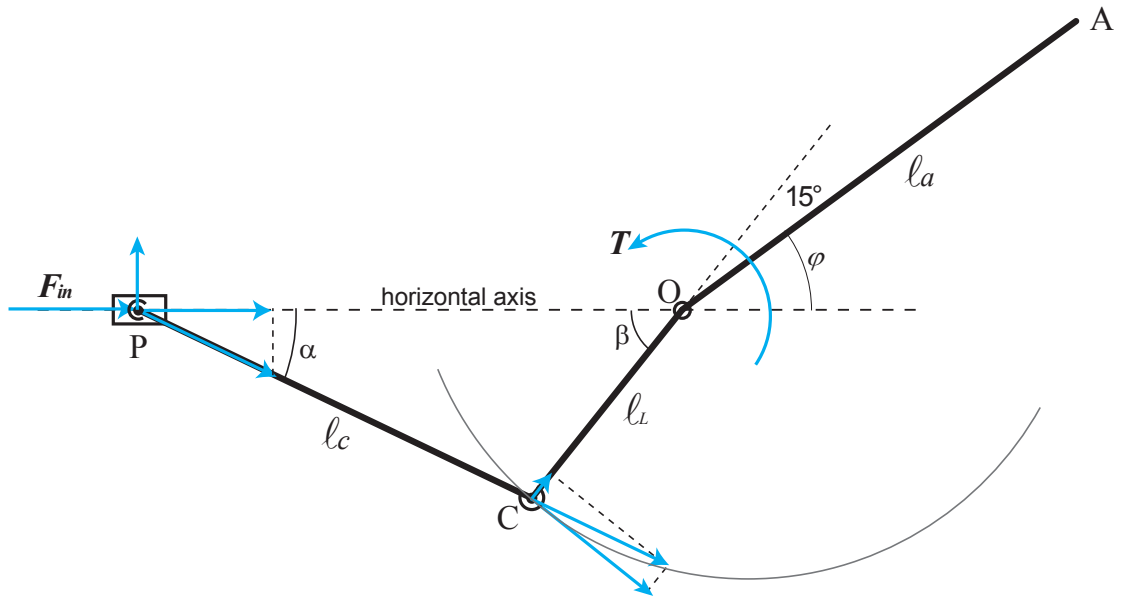


Figure 4.16. Slider-crank mechanism.

surement process to keep inner air mass. In case of pressure-constant mode, to maintain internal pressure constant, compressed air was periodically pumped or exhausted during measurement.

Fixed-angle test was carried out to measure dependence of flexion torque on pressure. Flexion angle was fixed at various angles first, and then the torque was measured as inner pressure was increased from 1 kPa to 7 kPa by pumping compressed air.

To investigate the shape deformation of the prototype, photographs were taken from the top at various flexion angles.

4.5 Results and discussion

4.5.1 Shape observation

Top views of N11-type at 0°, 40°, 90° and 120° of flexion angle are shown in Figure 4.17. The observation was done in mass-constant mode under 2 kPa initial interior pressure. Solid lines and dotted lines indicate assumed edge lines of cylindrical parts and isotenoid part, respectively. At 0°, many wrinkles can be observed on spherical surface indicating the presence of excess skin, which gives rotational freedom. As flexion angle increases, extraction of wrinkles on outer side of joint is confirmed.

As can be seen from the figure, places of connection between cylindrical and isotenoid parts look squeezed by spheroidal sections. Top view shows that at 90°, inner side of isotenoid part is not fully compacted within illustrated sector area, but keeps bulged pushing cylindrical parts. When flexion angle reaches maximum designed flexion angle ϕ_{max} , 106.6°, as is mentioned in section 3, both cylindrical parts overlap each other geometrically. The photograph at 120° shows that both cylinders push each other and become slightly bent.

The other important finding is that the actual inflated shape of isotenoid part did not perfectly match with isotenoid; wrinkles were merged together making cleft in meridian direction, and bulge in meridian curve was not uniform. Similar to buckling phenomena observed for super pressure balloon [45], stress distribution for such non-symmetric shape is more complex, and value of circumferential tensile stress cannot be neglected. It means that remaining in isotenoid part tensile stress may inhibit contraction of the inner side of joint.

Inner side views of N9-type at 0°, 40°, 60°, 80° and 110° of flexion angle under 2 kPa of initial inner pressure are shown in Figure 4.18. At 0°, isotenoid part already has excess amount of skin, which is mainly placed in middle and connection part between isotenoid and cylindrical part. In the middle, big cleft is confirmed and skin is twisted and deepen into inside. Separating isotenoid part with the cleft, as flexion angle increases, two lumps seem crossing and twisting each other. In addition, it was confirmed that connection parts close to end-fittings were raised, that indicates the initially round cross section of cylindrical parts acquired shape of vertical ellipse.

On the other hand, Figure 4.19 shows outer side views of N11-type at 60°, 80°, 110°, 120° and 125° under 2 kPa of initial inner pressure. As flexion angle increases, skin of isotenoid part is extended and cleft and wrinkle become sparse. Interestingly, two clefts are observed; one starts from top and the other starts from bottom.

Comparison between N9 and N11 are shown in Figure 4.20. The photographs show top views at 0 to 100° of flexion angles in every 20° step.

Figure 4.21 shows comparison of center-line traces between N9- and N11-type of the prototypes at 0°, 60°, 100° of flexion angle in mass-constant mode under 2 kPa initial interior pressure. On N11-type and N9-type except 100°, the center line is bent at near the point corresponding to the axis of isotenoid. The observation agrees that extension of the outer side and contraction of the inner side of isotenoid part are performed correctly. On the other hand, at 100° of flexion angle of N9

type, the center line is bent at connection part between cylindrical and isotenoid part, and flexed point is translated from the axis of isotenoid. This is explained that excess skin on outer side of isotenoid part disappears at 100° for N9 type, and bending deformation of cylindrical part started, which is not preferable for maintaining volume constant.

4.5.2 Pressure profile

Figure 4.22 shows pressure profile of N11-type's flexion test in mass-constant mode at 2 kPa and 5 kPa of initial inner pressure. In this experiment acquired pressure values were corrected by adding air leakage rate, which was preliminarily recorded.

As can be seen from the figure, with changing flexion angle from 0° to $35\text{--}45^\circ$ inner pressure smoothly decreases from 2 kPa (5 kPa) to 1.7 kPa (4.7 kPa), and after that increases and reaches values of 3.4 kPa (6.5 kPa) at 120° . Since pressure and volume are in inverse relation, these data mean that the inner volume reaches maximum at around $35\text{--}45^\circ$.

Concerning area of high flexion angles (from 80° to 120°) it should be mentioned that according to the joint design cylindrical parts touch each other at 106.6° and start to deform their shape after, thus decreasing total volume of the joint and causing pressure increase. In actual situation inner sides of cylindrical parts connected to the isotenoid begin to deform at much smaller angles, as can be confirmed from Figure 4.17, because of presence of compacted folds of inner isotenoid part. It is considered that deformation of cylindrical parts is the main reason of pressure increase. Decrease of pressure in low flexion angles area (from 0° to 50°) may be connected with structural stability problem, similar to buckling of a lobed inflatable structure, where the total volume in a deformed mode appears to be greater than the initial volume [54].

4.5.3 Torque trajectory

Figure 4.23 shows the relationship between flexion angle and torque for N9- and N11-types, which have full extracted angle at 85.3° and 127.8° respectively, at 5 kPa initial inner pressure. For better clarity only flexion direction (from 0 to 120°) curves are displayed. It can be seen from the Figure that for N9-type the flexion torque starts to increase rapidly after 85° , while curve of N11-type keeps increase with continuous incline rate. These data clearly show that until some excess skin is still available, the torque value remains low, and after all excess skin is completely extracted, as in case of N9 at 85° , the flexion torque starts to rise rapidly.

The full (flexion-extension) torque trajectory for N11-type at 2 kPa and 5 kPa of initial inner pressure is shown in Figure 4.24. While the gradient of curve at 2 kPa differs from that at 5 kPa, general trend of both curves is mostly identical. For both initial pressure conditions, torque hysteresis is observed, that means that for the same flexion angle torque measured during flexion is higher than measured at extension. Such hysteresis curve is typical for fabric structures. When fabric is loaded, the individual yarns both elongate and slide over each other, resulting in load-extension curves characterized by the yarn compliance and the friction between yarns. Torque-angle measurements made on EMU in previous studies [6, 15] also indicate that the torque-angle relationship for a fabric space-

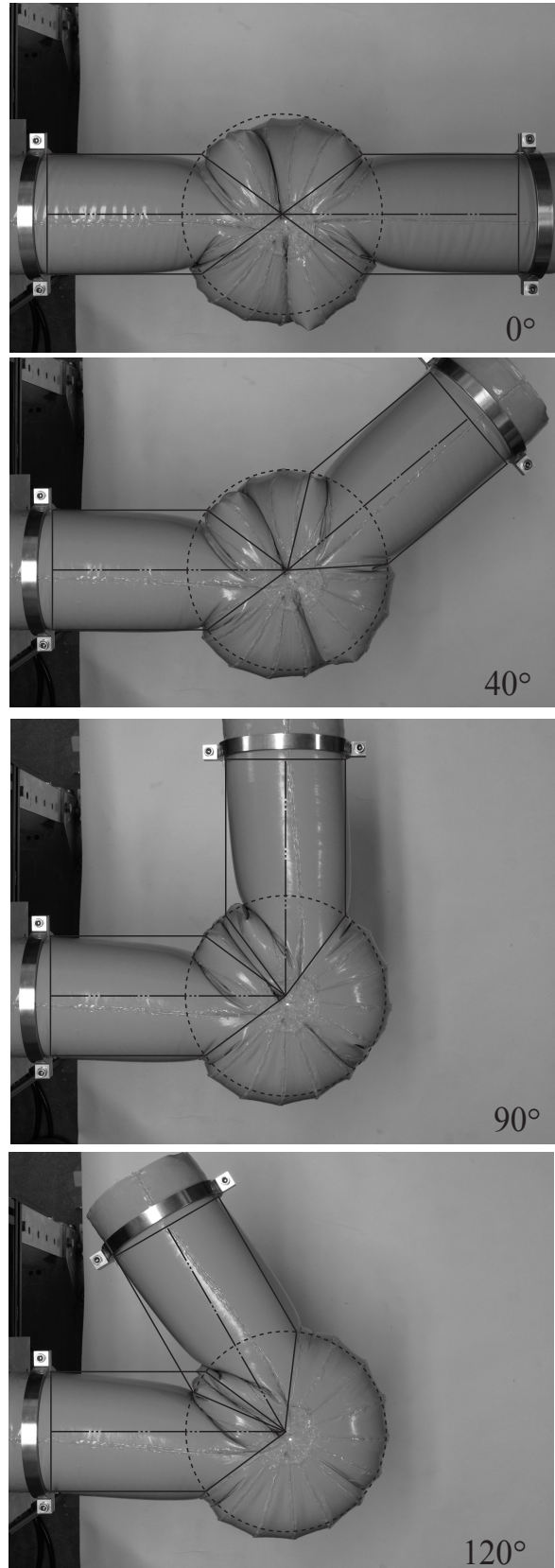


Figure 4.17. Photographs of arm joint flexion at 0°, 40°, 90° and 120° of flexion angle (top view).

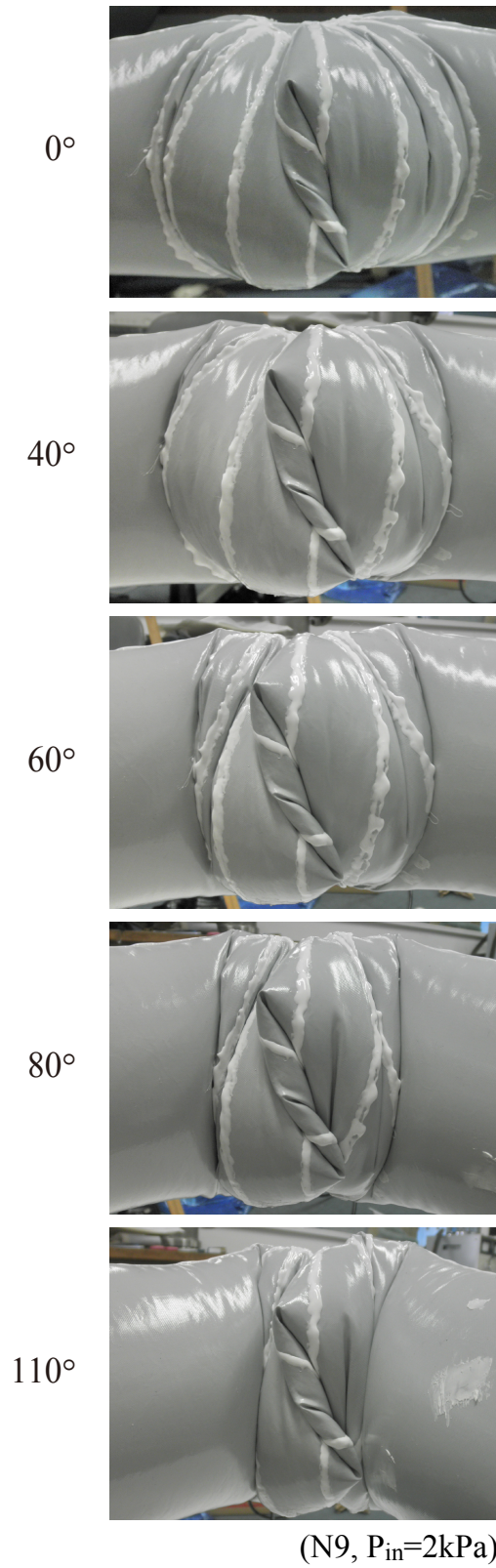


Figure 4.18. Photographs of arm joint flexion at 0°, 40°, 60°, 80° and 110° of flexion angle(inner-side view).

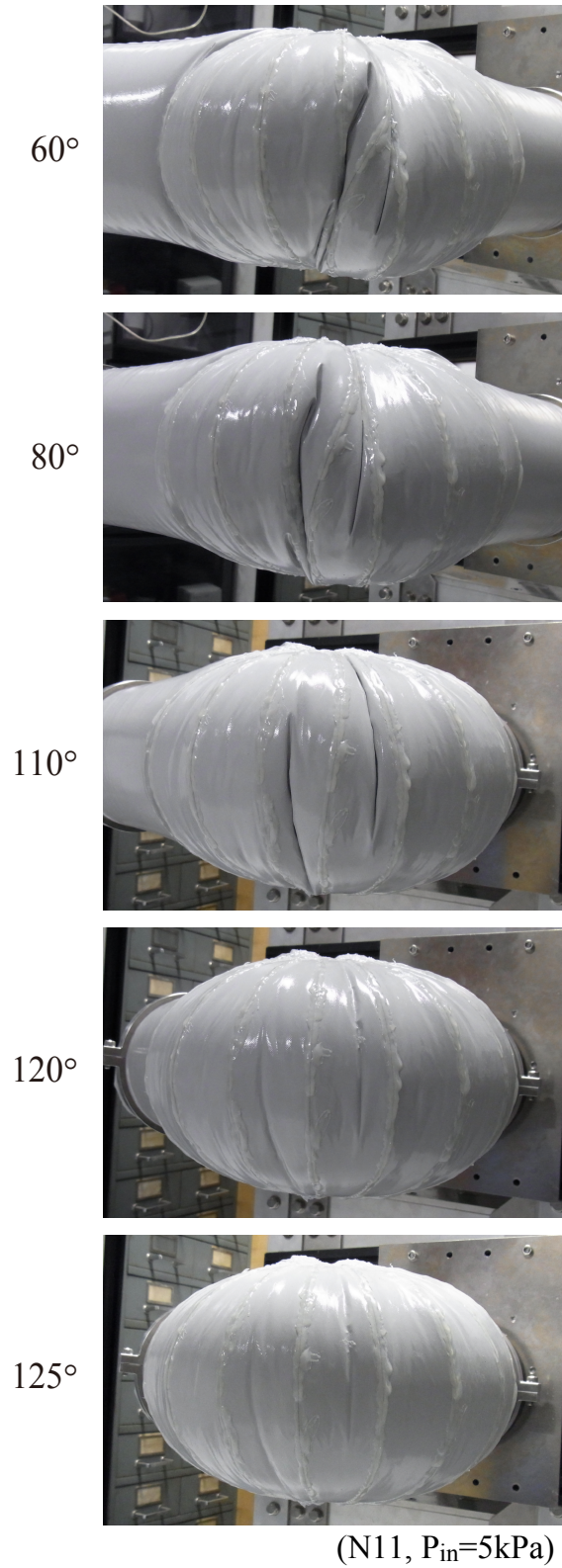


Figure 4.19. Photographs of arm joint flexion at 60°, 80°, 110°, 120° and 125° of flexion angle(outer-side view).

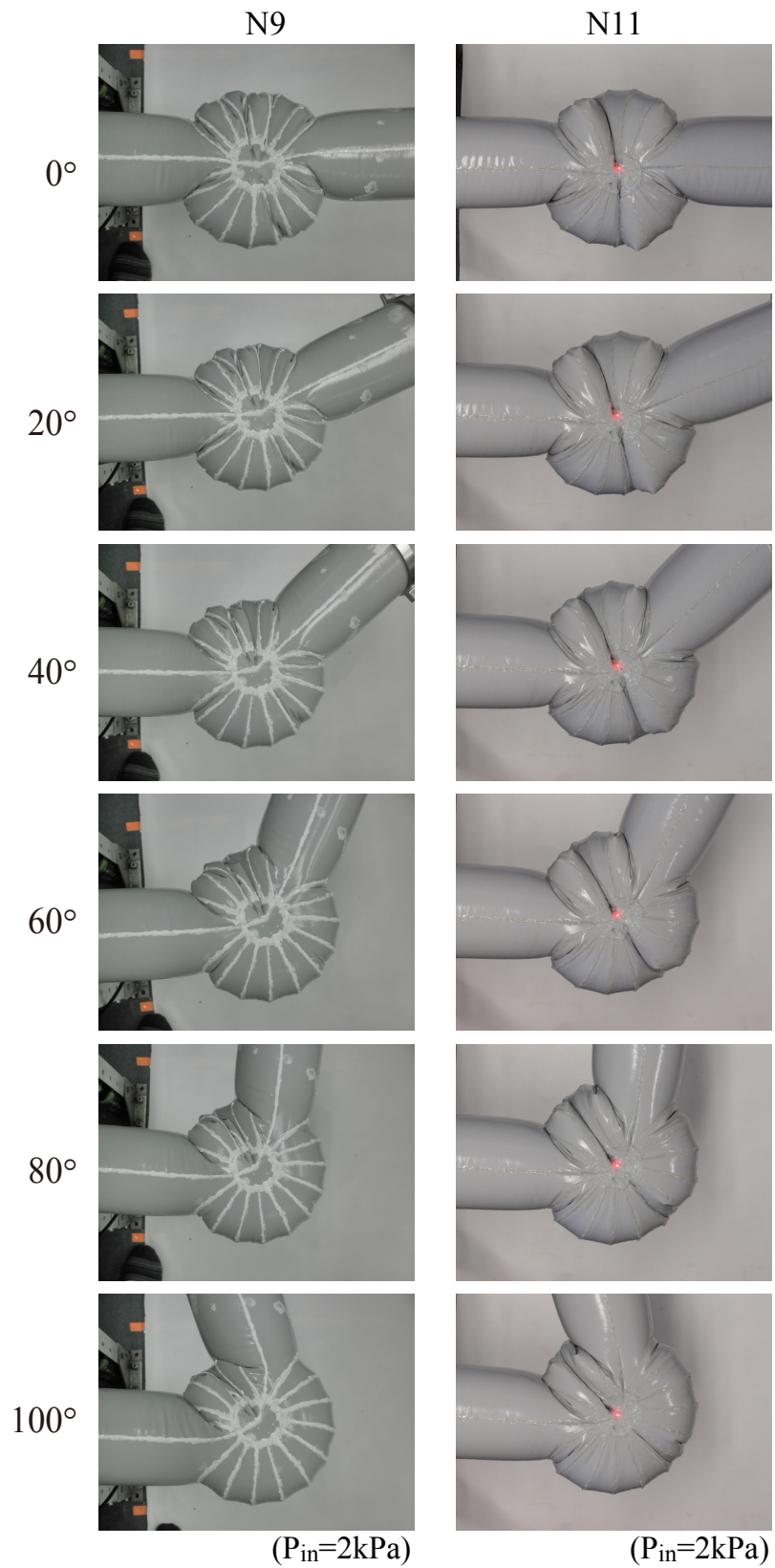


Figure 4.20. Photographs of arm joint of N9 and N11 type of prototypes from 0° to 100° of flexion angle (top view).

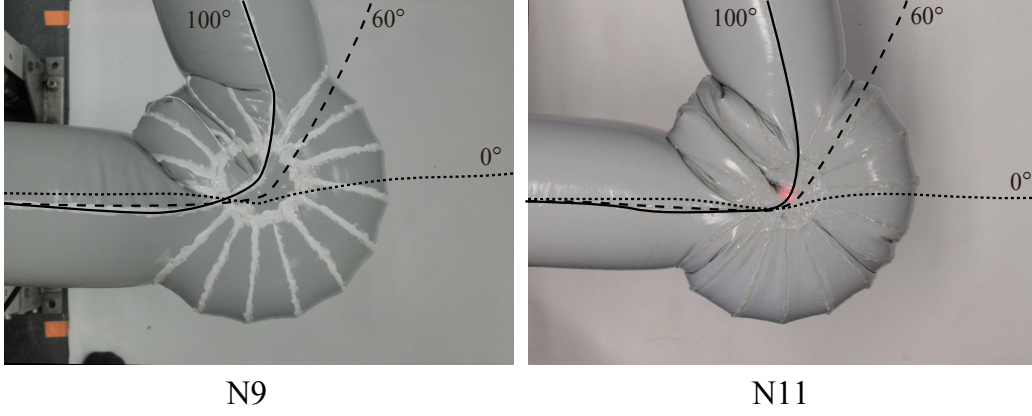


Figure 4.21. Center-line traces at 0, 60 and 100° of flexion angle in N9 and N11 types of prototypes under 2 kPa of initial inner pressure.

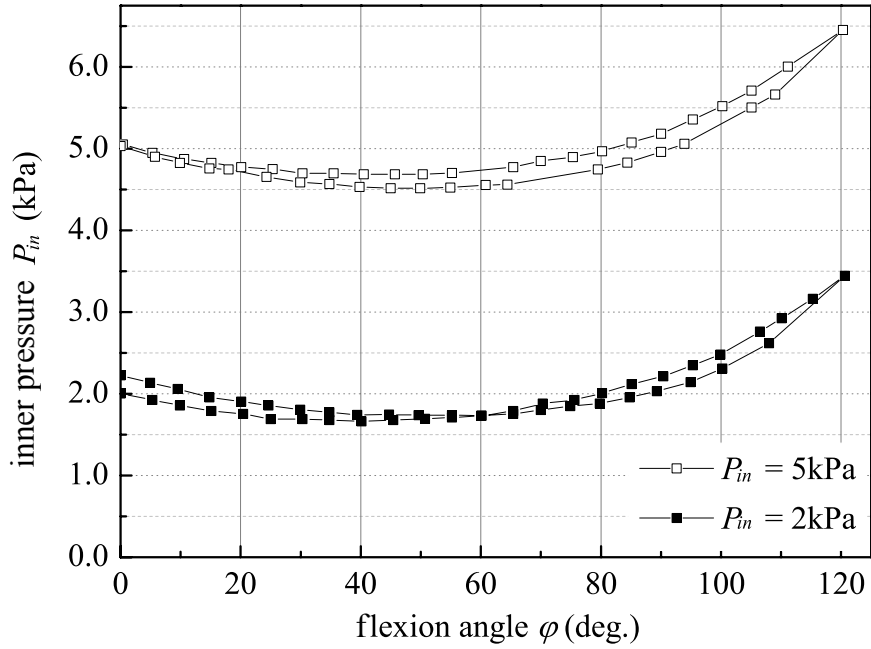


Figure 4.22. Pressure profile at 2 kPa and 5 kPa of initial inner pressure.

suit joint is hysteretic.

Neutral position, at which torque vanishes during flexion, is around 30° for both pressure values. Torque at 0° has negative value which means the joint prototype is forced to flex up to the neutral position. The shift of neutral position from 0 to 30° can be explained by the fiber reorientation, and it can be related with the changes of enclosed volume of the joint prototype as well. As can be seen from Figure 4.22, the prototype encloses the maximum volume around 35-45°. Also from the results of static torque dependency on inner pressure acquired by fixed-angle tests (see Figure 4.26) it was found that at approximately 30° of flexion angle, static torque keeps constant value though inner

pressure changes; above 30° torque increases with pressure, and below 30° decreases. In case of all flexion angles, torque shows linear dependence on inner pressure. The proximity of the values of flexion angle where volume is the maximum and torque is independent on pressure to the neutral angle may indicate their connection, and deeper investigation to clear this point is required.

As shown in Figure 4.24 torque linearly increases from 20° to 80° , and then rises sharply. This trend become clear by describing derivative of torque-angle curve as shown in Figure 4.25. As mentioned in section 4.1, some amount of tension is remained on spheroidal surface along circumferential direction. This tension force may inhibit contraction of the inner side of joint and may be a primary cause of linear torque increase. Rapid torque increase from 80° to 120° can be related with cylindrical parts squashing, which causes pressure and elastic strain energy increase in cylindrical parts' skin material.

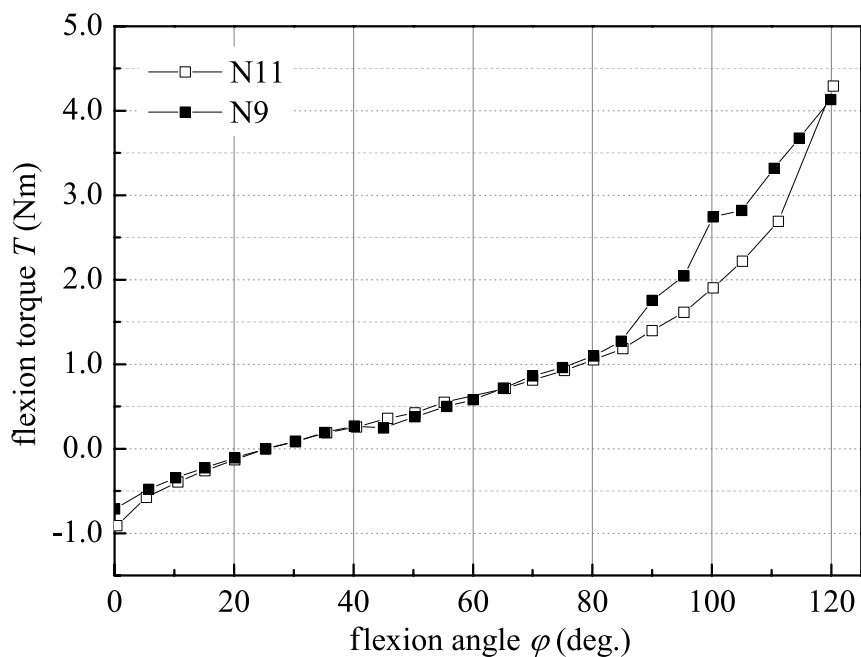


Figure 4.23. Comparison of torque trajectory for types N9 and N11.

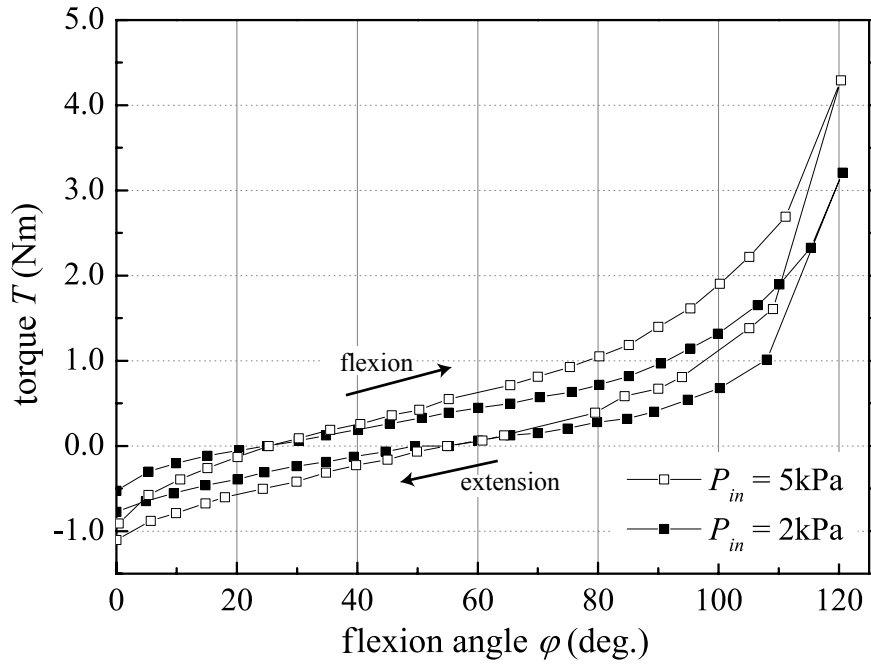


Figure 4.24. Relationship between flexion angle and torque. Initial pressure is set as 2 kPa and 5 kPa.

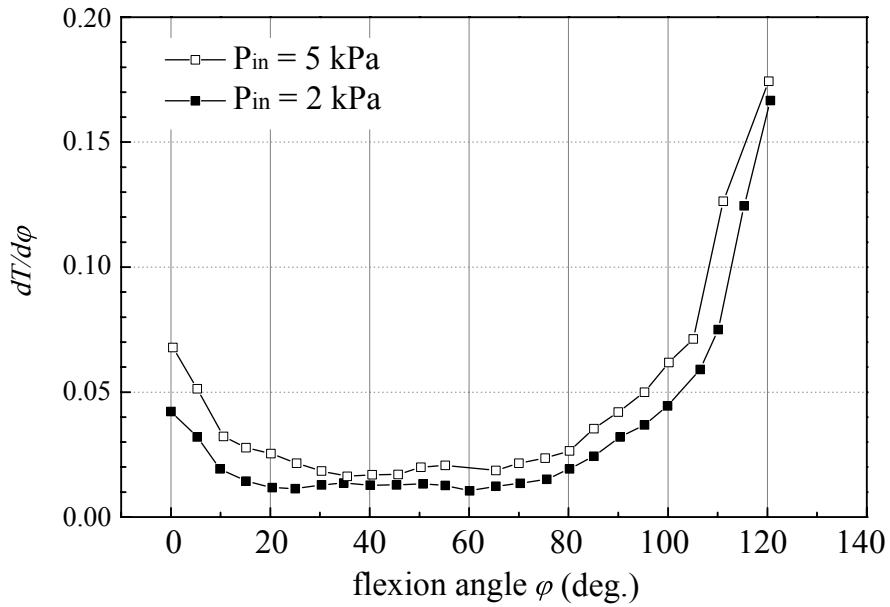


Figure 4.25. Relationship between derivative of torque and flexion angle.

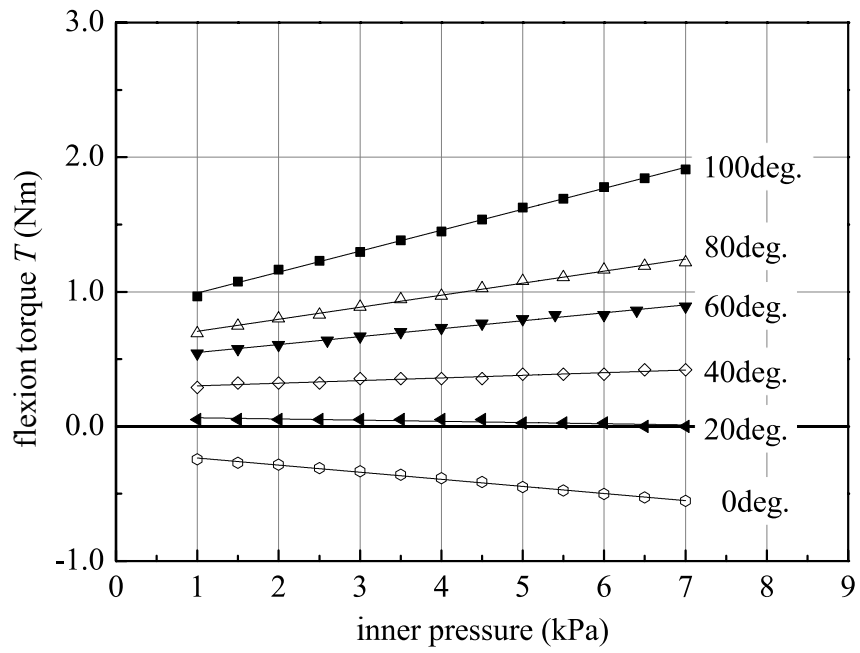


Figure 4.26. Torque dependency on inner pressure at various flexion angles.

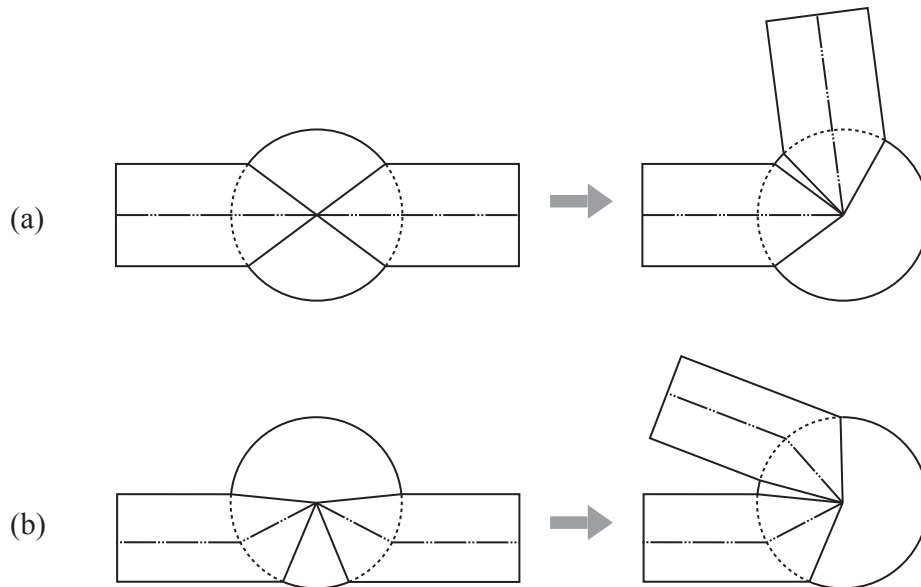


Figure 4.27. Schematic top-view; a) current design, b) an improved design.

4.6 Summary

Single-axis joint made of soft material with potentially high flexibility for knee, elbow and finger mobility elements of spacesuits is proposed on the basis of employment of isotenoid shape for providing excess skin to the joint parts subjected to contraction and extension.

From the numerical analysis, any revolution with sufficient excess skin along circumferential direction surely becomes the shape of isotenoid. It is found that membrane of isotenoid lose its circumferential stress mostly when the shape reaches isotenoid. The results indicate isotenoid with excess skin may allows its skin to transit along circumferential direction without applied force.

Prototypes of the elbow joint of two types were fabricated and tested on their flexibility under pressurization from inside the prototypes using originally developed measuring instrument based on robotic arm.

The results showed that the torque maintains low level and mildly increased in the range of 20° to 80° . As long as excess skin remains on the outer side of isotenoid part, extension of the skin does not require mechanical work. Linear increase of the torque in the range of 20° to 80° was caused by remaining tensile stress on lateral surface of isotenoid part, which was found to have the shape deviated from ideal isotenoid.

Inner volume change with flexion angle also keeps low level up to 80° since extension on the outer side and contraction on the inner side are carried out at the same rate. Fiber reorientation and buckling phenomena are considered to be primary reason of the volume change over 80° . Moreover, it was found, that the neutral position of the joint is stabilized at around 30° , and static torque becomes independent on pressure at 30° . The proximity of these values to maximum volume angle suggests their mutual connection, that requires further investigation. Deformation of cylindrical parts (squeezing and bending) at flexion angles higher than 80° is responsible for decreasing of volume inside joint and sharp increase of torque.

Future improvements can be connected to optimization of shape and constructive element design to reach better constant volume capability and higher flexion angle capability and to enhance structural stability of isotenoid part without occurring buckling phenomena.

Chapter 5

General Conclusion

The primary aim of this thesis is to advance understanding of variable shape pneumatic structures in order to improve flexibility of spacesuit joints. This thesis defined variable shape pneumatic structures as closed pressurized cylindrical membrane structures, which can bend in one plane with minimum applied force.

To begin with, Chapter 2 described fundamental requirements for variable shape pneumatic tubes, then provided an overview of existing mechanical concepts of those structures. After analyzing existing concepts, problems of these concepts are described. Then, the new design concept by eliminating the problems was proposed.

The existing concepts could be divided into two groups: restraint group and restraintless group. Most of concepts in restraint group has axial restraint cables, which becomes stiffened since strong tensile load is applied due to pressurization and hinders bending deformation of the tube. In addition, because axial cable is locally attached, shape stability has been another issue. A structure which does not have axial restraint is proposed, but it has transverse restraint performing axial restraint, called Membrane Beam Folding Structure (MBFS).

On the other hand, restraintless group has intrinsic problem because bending property relies on its material or mechanical elasticity. As inner pressure is high, elasticity must be designed higher to prevent significant inflation consequently bending property becomes degraded. One solution is to modify shape in the middle of a membrane cylinder to avoid tensile stress in axial direction. The proposed structure is named Isotenoid Excess Membrane Structure (IEMS).

Chapter 3 presented a design of MBFS. Polygonal frames attached on cylindrical membrane regularly constrain the structure to reduce axial tensile stress on membrane. That allows extension of outer part and contraction of inner part with smaller external force. With measurement of bending property of MBFS, compared to a plain pneumatic tube, the MBFS showed large deformation soon after loading started. This result supports axial tensile stress on membrane is mitigated with the polygonal frames. The stress mitigation is also indicated from the numerical study conducted with the model of inflatable tube bending. The MBFS also showed less volume change compared from plain pneumatic tube, which also suggest feasibility of MBFS.

Chapter 4 presented another design of variable shape pneumatic structure: IEMS. The proposed

joint is intended to sustain excess material which gives rotational freedom since isotenoid does not carry circumferential stress analytically. The structure has the axi-symmetric geometry, which has advantage for maintaining constant volume. From the numerical analysis, it is found that membrane of isotenoid shape loses its circumferential stress mostly which agrees with analytical study.

Prototype of IEMS maintained its torque at low level and mildly increased its torque in the range of 20° to 80° . As long as excess skin remains on the outer side of isotenoid part, extension of the skin requires little mechanical work. Linear increase of the torque in the range of 20° to 80° was caused by remaining tensile stress on lateral surface of isotenoid part, which was found to have the shape deviated from ideal isotenoid. Inner volume change with flexion angle also keeps low level up to 80° since extension on the outer side and contraction on the inner side are carried out at the same rate. In the range of higher angle over 80° , large deformation of cylindrical parts occurs resulting in degrading flexibility.

Table 5.1. shows qualitative comparison of important characteristics between MBFS and IEMS. Flexibility of joint mainly consists of the three factors: work for contraction of skin on the inner side, work for extension of skin on the outer side and volume change. In case of MBFS, the contraction needs lower applied force since transverse frames restrain tensile stress on skin. On the other hand, the extension needs small amount of mechanical work since frames hinders the extension. In case of IEMS, the extension is carried out with little mechanical work with the aid of excess skin. The contraction requires mechanical work since circumferential tensile stress remains. Volume change during flexion is derived from difference between decrease in contraction of skin on the inner side and increase in extension of skin on the outer side. Since IEMS does not have large difference in applied force between inner contraction and outer extension, volume change is much smaller than that of MBFS. Symmetric geometry also contributed volume maintenance.

Holding excess material by structural elaboration has great potential approach for variable shape pneumatic structure. MBFS holds excess material by constraint with attachment of polygonal frames. IEMS holds excess material by its unique geometry. It is suggested that stability should be incorporated with pneumatic structures to avoid unexpected shape change. Through those discussion, the investigations of both MBFS and IEMS helped understand designing variable shape pneumatic structures.

Table 5.1. Comparison of characteristics regarding flexibility between MBFS and IEMS.

	Work for extension on outer side	Work for contraction on inner side	Volume change
MBFS	very low (by the aid of transverse frame restraint)	high (due to hindrance by frames)	low
IEMS	low (with remained tensile stress)	very low (by the aid of excess skin)	very low

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Appendix I Spacesuit development

This appendix describes functions of present spacesuit garment and the environment for future missions.

Significance of spacesuit development

Spacesuits have served as one of the indispensable tools for space exploration since the very beginning of human space flight programs. During launch, landing and in harsh space environment spacesuits perform the fundamental role of supporting astronauts' survival, which is especially true in case of extravehicular activity (EVA), where spacesuits become a single interface between astronauts and outer space. Spacesuits generally can be divided into several categories depending on their purposes: Launch, Entry, and Abort (LEA) suit, Intravehicular activity (IVA) suit and Extravehicular activity (EVA) suit. Here, this study focuses on improvement for capability of extravehicular activity suit.

To protect from such harsh environment, life-support function is the most fundamental and critical function of EVA suit. Safety and reliability is most important criteria on spacesuit development and operations. Many factors including atmosphere composition and pressure control, thermal control, radiation protection, human performance, and other areas must be addressed. Thus, a spacesuit is often referred to as a smallest spacecraft.

In addition to the fundamental life-support function, spacesuits should provide capabilities to do useful work outside the spacecraft. Spacesuits are required to not restrict human abilities in perceiving external environment, estimating the situation and taking physical actions, and moreover to enhance these abilities where it is possible. Compared to robotics and automation, future EVA missions should offer many advantages for accomplishing space missions. The following capabilities during EVA are key factors to enhance mission levels: flexibility, dexterous manipulation, human visual interpretation, human cognitive ability, and real-time approaches to problems.

According to NASA's vision of future mission opportunities, their potential mission applications include EVAs for missions to near Earth objects (NEO), Phobos, or future surface missions. Surface missions could include either exploration of the moon or Mars [55]. Then, NASA reported that missions to planetary surfaces will require technology advancement in areas such as increased component life, protection from radiation events, mobility in partial gravity environment, dust mitigation, interfaces with rovers and habitations, high-data-rate-network-based communication, innovative life support systems that minimize consumables or take advantage of in-situ resources, smart caution and

warning capabilities, and advanced information systems that enable crew autonomy.

Japan Aerospace Exploration Agency, JAXA, documented future plans about human space activities in JAXA Vision, in which the agency's proposal on its own initiative and the ideal situation of aerospace areas in the next 20 years are described [56]. It was stated, "JAXA will accumulate necessary technologies through the operation and utilization of the International Space Station. This will be done with a view to participating in future human exploration of the moon through international cooperation. JAXA will establish technologies required for human presence in space and human space activities, so that it would be ready for its own space activities in about twenty years." it should make necessary preparation for enabling its own human space activities that would contribute to various space utilization. Development of spacesuit, which is a key component of extravehicular activity system, should be included in the preparation.

Lately, several organizations have been involved in projects to develop spacesuits [57, 58]

Environments on the lunar and Martian surfaces

Compared to operations on the Earth orbit, spacesuit system for planetary surface missions will especially take more significant roles for astronaut survival and mission accomplishment than other previous programs. This section presents the difference of environment between an Earth orbit and lunar and Martian surfaces.

■**Gravity** Equatorial gravity on earth, the moon and Mars is 9.83, 1.62 and 3.71 m/s², respectively. Gravity determines carry weight, body posture and reaction methods for doing useful work of spacesuit. On planetary surfaces, people must be able to walk in their suit. The mechanics of walking are deceptively complex, involving the ankles, knees, and hip as well as controlling center of gravity, tactility and visibility. Gravity strongly influences our posture and the way we do work.

■**Radiation** Except for passes through the South Atlantic Anomaly, the Earth's geomagnetic field protects EVA in Low-Earth orbit. The risk changes for EVA outside the Earth's magnetic field where solar-particle events represent the greatest threat of ionizing radiation. The shuttle's spacesuit cannot reasonably provide protection against a solar particle event, therefore, the best protection is to limit the time for EVA and employ an early warning scheme that allows time to get to a storm shelter. Radiation is still a concern for EVA on Mars, but the atmosphere offers an equivalent protection of 27 g/cm² [Brown and Shentrup, 1990].

■**Mircometeoroid** Micrometeoroids are a real hazard to the EVA crew, and because of the added spacecraft debris, LEO represents the worst conditions.

■**Dust** Lunar dust is actually a coarse grit that abrades suit coverings, scratches helmet visors, covers external displays, degrades outer garment absorptivity and evissivity, and contaminates seals and bearings. The dust's electrostatic and mechanical properties allow it to cling to the suit, making it difficult to remove before entering the airlock. The dust on Mars is like talcum powder, so it should be less abrasive than the lunar grit but still will contaminate bearings and seals. Mars has

global dust storms that can obscure visibility and should influence planning for excursions and sizing contingency supplies.

■ **Thermal environment** A typical 90-minute orbit in LEO means about 55 minutes in hot Sun and 35 minutes in the Earth's cold shadow. A planned 6-hour EVA would experience 4-5 of these cycles. The thermal environment on the Moon changes slowly, remaining virtually constant over a planned EVA period. A lunar day is 29.53 Earth days; the surface temperature can vary from 114 °C at lunar noon to -183 °C during lunar night. Note that for Mars, because of its atmosphere, heat rejection through sublimation does not work, and multi-layer insulation works less well because the atmosphere conducts heat and the gravity compress the layers of materials.

Functions of present U.S. spacesuit garment

This section overview how the design of present spacesuit tackles with hazardous environment mentioned in the previous section. This section especially focuses on functionality of the garment, which is composed of multiple fabric layers, and clarifies which layer this thesis focuses on. The section is abstracted by the literatures written by Larson[59] and Newman [60]

Present U.S. spacesuit system, known as the extravehicular mobility unit (EMU), consists of a space suit assembly (SSA), an integrated life support system (LSS), and the EMU support equipment as shown in Figure 5.1. The SSA is a 29.6 kPa, 100 % oxygen spacesuit made of multiple fabric layers. The liquid cooling and ventilation garment (LCVG) is the innermost layer and provides thermal control by circulating air and water (cooled by sublimator) over the crew member's body. The LCVG is made of nylon/spandex which is lined with tricot. Ethylene-vinyl-acetate plastic tubing is woven throughout the spandex to route the water close to the crewmember's skin for body cooling.

Next layer is pressure garment modules to retain pressure over the arms, legs and feet. These pressure garment modules are made of urethane coated nylon and are covered by a woven Dacron restraint layer. Sizing strips are used to adjust the length of the restraint layer. This study focus on replacement of design of this layer.

Then, the thermal meteoroid protection garment (TMG) comprises the final layers of the EMU's fabric components. The TMG liner is neoprene-coated rip-stop nylon and it provides puncture, abrasion, and tear protection. The next five layers are aluminized Mylar thermal insulation which prevent radiant heat transfer [61].

Radiation shielding is of the utmost importance for crew safety during EVA. The space shuttle EMU provides shielding with an approximate aluminum equivalent of 0.5 g/cm² to the upper torso. It has been noted by Thompson et al. that 1.62 g/cm² would be required for polar and geostationary earth orbit (GEO), where less deflection of solar particles is afforded by geomagnetic fields.

The usable lifetime of a spacesuit is important issue, especially when considering permanent human presence on space stations and planetary bases. Many shuttle EMU spacesuit components had an 8-year lifetime and are currently inspected after every EVA and certified for 20 hours of EVA time, after which a major overhaul is undertaken.

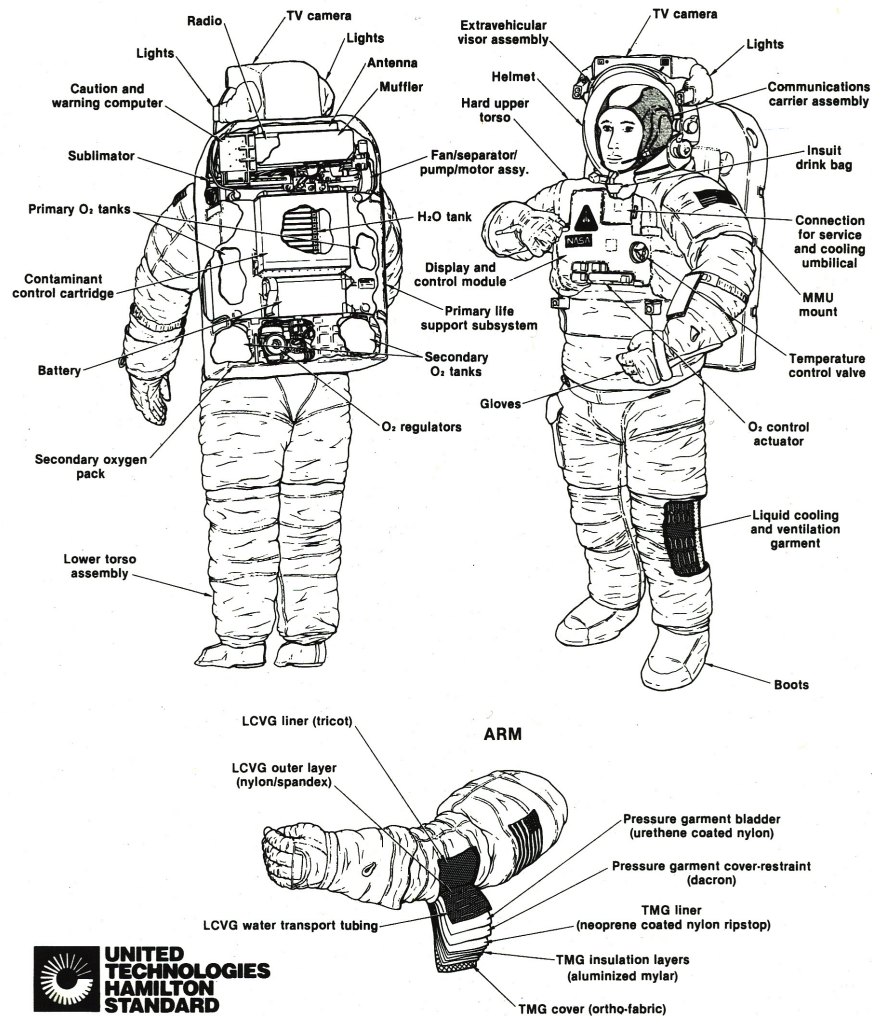


Figure 5.1. Schematic of US spacesuit: Extravehicular Mobility Unit [6].

Operating pressure and physiological issues

The suit operating pressure selection calls for a trade-off of physiological tolerances of the crewmember versus technical feasibility, such as mobility, amount of oxygen available to compensate for leakage, mass and strength. A high pressure suit of 52.2 kPa to 57.3 kPa reduces the need for prebreathing prior for EVA to prevent decompression sickness and was established with the risks of decompression sickness and physiological considerations as guiding parameters [60]. On the other hand, improved mobility of the pressurized suit and enhanced performance characteristics of the LSS can be offered with the minimized positive pressure. Setting lower operating pressure such as USA EMU, 4.3 psi (29.6 kPa) while cabin pressure of international space station is 101.3 kPa, is most preferable for the point of view of the astronaut mobility and enhancement; however, this pressure requires the prolonged denitrogenation, which makes the preparation to EVA more complicated [16].

When the human body is exposed to a sudden decrease in atmospheric pressure (for instance, from

a 70.3 kPa cabin pressure to the 29.6 kPa of the suit) nitrogen traces in the bloodstream will expand during this decompression and can create tiny bubbles and the potential for decompression sickness. In order to lessen this effect, the astronaut must "prebreathe" pure oxygen to help purge nitrogen from his or her bloodstream before entering the low pressure pure oxygen environment of a spacesuit [60].

Prebreathing for scheduled and unscheduled EVAs is normally accomplished from an orbiter cabin pressure which has been reduced to 70.3 kPa (10.2 psia) using protocol options identified in Table 5.2. The initial prebreathe must be an unbroken prebreathe with over 95 % oxygen, at least 45 minutes which must be accomplished prior to cabin depressurization below 86.2 kPa (12.5 psia). The final prebreathe necessary determined by the time at 70.3 kPa (10.2 psia). With flight surgeon consultation, the final in-suit prebreathe time may be gradually reduced for 70.3 kPa (10.2 psia) exposures longer than 12 hours but less than 24 hours. Figure 5.2 shows protocol for prebreathing using staged decompression at 70.3 kPa [19].

To conduct prebreathe efficiently with shorter time, the protocol of exercise prebreathing was proposed. Exercise increases ventilation and tissue perfusion. By increasing perfusion, exercise allows more nitrogen to diffuse into blood returning to the heart per unit of time and increases efficiency of denitrogenation. The protocol takes 2 hours 20 minutes [62, 63]. If the exercise prebreathing is proved to decrease the risk of decompression sickness as much as conventional prebreathing protocol, the prebreathing option will be adapted for implementation and prebreathing time will be shorten. However, in case of contingency which requires evacuation from pressurized spacecraft or habitant module as soon as possible, prebreathing still remains complexity of EVA operations. Risk control of decompression sickness remains, too.

To control the risk of decompression sickness, multiple regulations are brought. It is necessary to have optimal interval between EVAs interval since recent prior exposure to low relative pressure is also risk factor for the development of decompression sickness. Reexposure to altitude within a few hours of a previous flight is associated with an increased incidence of decompression sickness. Repeated EVA exposure after 17 hours did not appreciably increase the risk of decompression sickness. The effect of physical exercise during and after exposure increasing the incidence of decompression sickness is well recognized [60, 64]. Duration of exposure to the lower pressure also contributes to the risk of decompression sickness. It is well known that the greater the duration of exposure to lower relative pressure, the greater the likelihood of decompression sickness. Increasing age, obesity, female gender, and decreased ambient temperature are also thought to increase the risk of decompression sickness

In summary, lower operating pressure brought two issues; 1) duty of prebreathing, and 2) control risk of decompression sickness. Consuming time for prebreathe shortens crew time and it directly reduces operation time for other missions such as microgravity experiments in ISS [65]. Risk control of decompression sickness significantly limits EVA schedule and severely limit astronauts who can take EVA.

Table 5.2. Scheduled and unscheduled EVA prebreathe options from a 70.3 kPa cabin pressure.

Initial prebreathe time (min)	Time at 70.3 kPa (h)	Final in-suit prebreathe time(min)
60	24	40
60	20	50
60	16	60
60	12	75

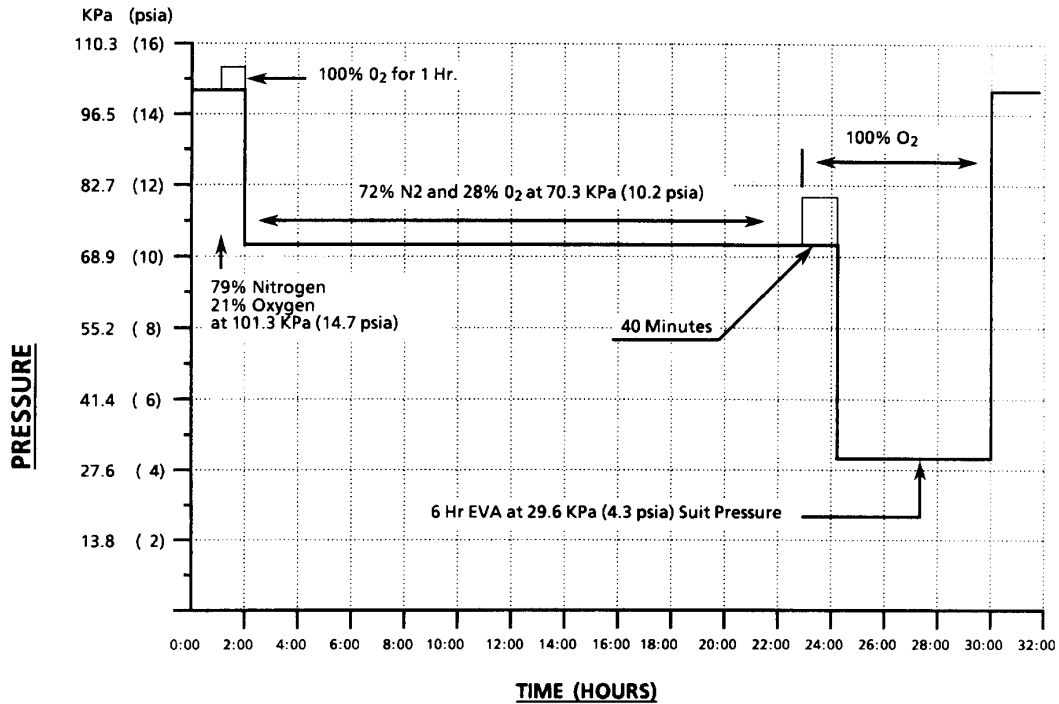


Figure 5.2. Protocol for prebreathing using staged decompression at 70.3 kPa [19].

Joint flexibility: guideline, harm and cause

As is described in the previous section, joint mobility is highly affected by operating pressure. This section presents joint flexibility in detail. First of all, this section mentions design guideline regarding joint flexibility established by NASA, then current joint mobility taken by a number of analyses. After that,

Specification for spacesuit designing

NASA's requirement for designing spacesuits can be referred from the NASA Space Flight Human Systems Standards ,NASA-STD-3000, and its superseded version, NASA-STD-3001 including the Human Integration Design Handbook (HIDH) [8]. The documents specify how to design systems

to support human health, safety, and productivity during space flight, intended to apply to all NASA vehicles and systems with a human crew.

In the documents, considerations and design requirements data regarding joint range of motion are provided. For example, the range of motion of elbow joint is 0 to 141°.

The documents mention that body strength of human, ability to generate force, is influenced by a number of factors. For instance, pressurizing the suit reduces body strength. Gonzalez et al. [66] indicated that joint torque production capabilities of subjects while they were wearing pressurized suit. Joint torque production capabilities of subjects decreased 15 % and 21% for elbow extension. Strength is also affected by longer missions because of the deconditioning of muscles. The result of a study by Adams et al. [67] indicate that spaceflight without exercise may cause greater muscle atrophy than bed rest. Exercise countermeasures have been used to counter these deficits, but to date have been only partially effective.

Here, Table 1.2 shows the strength data of elbow flexion and extension movement: subjects move forearm in a sagittal plane around the elbow joint. The table compares strength between unsuited, unpressurized suited and pressurized suited. Maximum forces are intended to be operated by unsuited crewmembers withstanding the forces exerted by the strongest crewmembers without sustaining damage. Criticality 1 load limits are to be used for crew safety situations and the design of items where a single failure could result in loss of life or spacecraft. Criticality 2 load limits are to be used for the design of items of which a single failure could result in a loss of mission.

Table 5.3. Strength Data Tables (edited from [8])

Elbow flexion	Minimum Crew Operation Loads [N]			Maximum Crew Operational Loads [N].
	Critical 1 Ops.	Critical 2 Ops.	Other Ops.	
Unsuited ^a	13	18	36	347
Unpressurized suited ^b	9	12	25	243
Pressurized suited ^c	7	9	18	174

NASA recognizes the need to improve mobility for future spacesuit development: Chullen [55] stated that to satisfy future development need, "the suit must also be able to operate at higher pressures 57.2 kPa (8.3 psid) as opposed to 29.6 kPa (4.3 psid) and provide for improvements in sizing and mobility of upper torso and lower torso." And, "as far as previous work is concerned, no significant lower torso development work has been performed in the past decade." "The lower torso will need to incorporate advances in materials, sizing methods, or improved fabrication techniques to improve mobility, increase life, or decrease mass."

Elbow extension	Minimum Crew Operation Loads [N]			Maximum Crew Operational Loads [N]
	Critical 1 Ops.	Critical 2 Ops.	Other Ops.	
Unsuited ^a	27	36	67	249
Unpressurized suited ^b	19	25	47	174
Pressurized suited ^c	14	18	34	125

^aPost-space flight estimated strength decrement. Range is 0%-26%. Average estimated is 20%. Based on max EDOMP Data. Not all motions were measured on EDOMP.

^b Post space flight estimated strength decrement. Range is 0%-47%. Average estimated is 33%. Based on CRV Requirements Document. Suit decrement not measured directly, but estimated based on functional strength testing of other movements

^c Post space flight estimated strength decrement. Range is 0%-47%. Average estimated is 33%. Based on CRV Requirements Document. Suit decrement not measured directly, but estimated based on functional strength testing of other movements

Flexion torques of spacesuit joints and torque analysis methods

A number of investigations have been reported on torque and angle measurements taken on space suit fabric bending joints.

Since it is impossible to measure torques directly in a human test subject's joints while he or she is wearing a suit without using invasive instrumentation or restricting the subjects' motion, measurements of the torque have to be accomplished indirectly. Several types of measurement methods are reported, and torque data are affected by the measurement methods.

A few study collected torque data on spacesuit joint from tests of empty pressurized suits [14, 16]. Morgan [17] performed strength measurements by comparing the subject's EMU-suited strength with that subject's unsuited strength using an isokinetic dynamometer. Schmidt [7] investigated spacesuit mobility by using both human subjects and an instrumented robot to obtain torque data. The EMU installed on the robot was pressurized to 30 kPa and maintained at the pressure. An joint torque-angle data for elbow flexion motion is shown in Figure.

Lately, Matty [18] has tested several kinds of suits to to define mobility requirement for the Constellation Space Suit Element (CSSE) : Extravehicular Mobility Unit(EMU), Advanced Crew Escape Suit (ACES), I-Suit, D-Suit, Enhanced Mobility (EM)-ACES, and Mark III(MK-III). The detailed results are described in another paper [68], but is not open to public yet.

Table 1.3 shows comparison of torque data for knee and elbow flexion of various torque analyses. The data taken by Schmidt shows much higher value on elbow torque. The reason is possibly due to contact effects between the spacesuit and the robot.

Issues caused by low joint flexibility

To begin with, poor suit mobility described in the above section results in astronaut's general and finger fatigue. The Apollo crews often felt they were fighting the resistance in the suit, the most fatiguing part of surface EVA tasks was repetitive gripping. One crew stated that "efficiency was no more than 10 % of the use of the hand" [69].

Table 5.4. Comparison of torque analyses of spacesuit joints.

	Dionne [14]	Menendez et.al [15]	Abramov et.al [16]	Morgan et.al [17]	Schmidt [7]
Method	EMU empty	Prototype joint segments	Orlan- DMA, empty	EMU, human subjects	EMU, human sub- jects and robot
Knee, 72deg.	3.2Nm	NA	6.0Nm	8.1Nm	3.7Nm
Elbow, 80deg.	2.0Nm	2Nm	2.2Nm	3.4Nm	14.6Nm

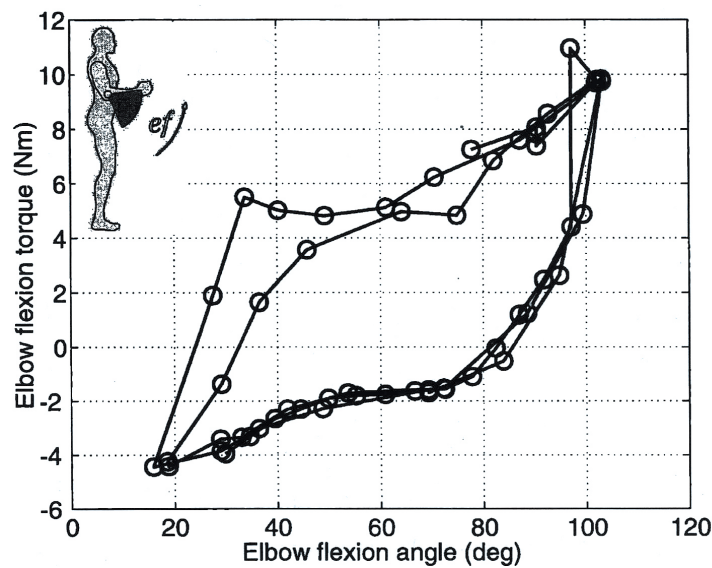


Figure 5.3. Torque trajectory of elbow joint of EMU measured by Schmidt [7].

Experience with EVA shows that a fulfillment of the crucial, difficult and long EVA operations can result in a development of regional and general fatigue accompanied by pains in the muscles of hands and arms during the first 1-2 days after the end of space walk. In other report [70], crewmembers have reported that at the end of the EVA's they were "tuckered out". Cardiac arrhythmias observed on the electrocardiograms are further evidence of the strenuous nature of EVA assignments. Occurrences of fatigue were repeatedly marked in US astronauts during EVAs in the Gemini, Apollo and Shuttle Space Programs [13].

Causes of flexion resistant torque

Resistant torque of spacesuit joint against flexion motion is caused by multiple factors.

Holschuh [71] investigated the cause of required torque for spacesuit flexion motion and character-

ized the factors into three: structural effects, volume effect and pressure effect. Structural effects are associated with flexure resistance of the suit materials and construction, such as the type, thickness and attachments of the base fabrics, bladder and axial restraint system. The structural effect is most significant at low flexion angles, corresponding to the nearly constant-volume regime. At higher flexion angles, constant-volume behavior breaks down, and magnitude of structural effect decreases and magnitude of both pressure effects and volume effects increase. Pressure effects is the work required to compress in the closed operating volume of the suit, and volume effect is the work required to displace gas due to internal volume change during movement

About structural effect, Harris [72] described in detail. The flat pattern joint adopted for single axis joint of EMU, which formed a “U” shape when deflected past 90 degrees. This tendency means the joint does not follow normal human anthropometry, and thus can exhibit high bending moments.

Another feature that results in higher torques is that the joint has two redundant 12.5 mm-thick longitudinal axis restraint members on each respective side. When the joint is pressurized, these axis restraint straps become quite stiff, then become harder to be bent.

Design parameters that affect spacesuit mobility are the internal pressure, radius of the spacesuit segment, and pleat design that allows the joint to bent without reducing the internal volume [7]. Schmidt mentioned friction between bladder layer and TMG (Thermal-Micrometeoroid Garment) layer cause structural effect

Appendix II Finite element model for excessive membrane

To enable undeformed geometry of excess, special finite element model in which excess material is embedded, is employed for FEM analyses.

This section explain how to modify a finite element model to embed excess material by using a parameter named “excess ratio”. The following explanation are based on the work done by Nakashino[73].

As shown in Figure 5.4, suppose that the point expressed by vector $\mathbf{X}$ in undeformed state moves to the point expressed by vector $\mathbf{x}$ in deformed state. The convected coordinate system r^1, r^2 is employed to locate a point on the membrane mid-surface is consistent with deformation of membrane. The covariant base vectors $\mathbf{G}_\alpha, \mathbf{g}_\alpha$ in the undeformed configuration $\mathcal{J}$ and deformed configurations $\mathcal{D}$ of the membranes are defined, respectively, as

$$\mathbf{G}_\alpha = \frac{\partial \mathbf{X}}{\partial r^\alpha} \quad \mathbf{g}_\alpha = \frac{\partial \mathbf{x}}{\partial r^\alpha} \quad (5.1)$$

If excessive membrane does not exist on the surface in undeformed configuration,

$$d\mathbf{x} = \mathbf{F} \cdot d\mathbf{X} \quad (5.2)$$

$$\mathbf{F} = \mathbf{g}_\alpha \otimes \mathbf{G}^\alpha : \text{the deformation gradient tensor} \quad (5.3)$$

where $d\mathbf{X}$ and $d\mathbf{x}$ is infinitesimal linear element in the configuration $\mathcal{J}$ and $\mathcal{D}$, respectively. When no excess membrane exists, Green-Lagrange strain tensor $\mathbf{E}$ in configuration $\mathcal{D}$ is described as,

$$\frac{1}{2}(d\mathbf{x} \cdot d\mathbf{x} - d\mathbf{X} \cdot d\mathbf{X}) = d\mathbf{X} \cdot \mathbf{E} \cdot d\mathbf{X} \quad (5.4)$$

Then, suppose excess skin is included on membrane surface in configuration $\mathcal{J}$. The case when membrane has $(L_0 - L)$ of excess membrane in configuration $\mathcal{J}$ along direction $\mathbf{W}$. Then $d\mathbf{X}$ becomes $d\mathbf{X}_{ex}$, both elements can be described that

$$d\mathbf{X} = (\mathbf{I} + B\mathbf{W} \otimes \mathbf{W}) \cdot d\mathbf{X}_{ex} \quad (5.5)$$

note that,

$$B = \frac{L - L_0}{L_0} \quad (5.6)$$

the B is defined as “excess ratio”.

When excess membrane exist, $d\mathbf{X}_{ex}$ corresponds to the infinitesimal linear element, the Green-

Lagrange strain tensor $\mathbf{E}_{ex}$ will satisfy the following equation:

$$\frac{1}{2} (d\mathbf{x} \cdot d\mathbf{x} - d\mathbf{X}_{ex} \cdot d\mathbf{X}_{ex}) = d\mathbf{X}_{ex} \cdot \mathbf{E}_{ex} \cdot \mathbf{X}_{ex} \quad (5.7)$$

The Green-Lagrange strain tensor taking account for an excess ratio is expressed that

$$\mathbf{E}_{ex} = \mathbf{E} + \frac{1}{2} [\mathbf{B}(\mathbf{F}^T \mathbf{F} \mathbf{W}) \otimes \mathbf{W} + \mathbf{B} \mathbf{W} \otimes (\mathbf{F}^T \mathbf{F} \mathbf{W})] + \frac{1}{2} [\mathbf{B}^2 (\mathbf{F} \mathbf{W})^2 \mathbf{W} \otimes \mathbf{W}] \quad (5.8)$$

It is clear that $\mathbf{E}_{ex}$ is a symmetric tensor according to equation (5.8).

Green-Lagrange strain tensor $\mathbf{E}$, and the one taking account for $\mathbf{E}_{ex}$ are expressed with the contravariant base vectors $\mathbf{G}^\alpha$ that

$$\mathbf{E} = E_{\alpha\beta} \mathbf{G}^\alpha \otimes \mathbf{G}^\beta \quad (5.9)$$

$$\mathbf{E}_{ex} = E_{ex, \alpha\beta} \mathbf{G}^\alpha \otimes \mathbf{G}^\beta \quad (5.10)$$

Here, taking account for symmetric property of both tensors, each covariant component is described that

$$\hat{\mathbf{E}} = \begin{bmatrix} E_{11} \\ E_{22} \\ 2E_{12} \end{bmatrix}, \quad \hat{\mathbf{E}}_{ex} = \begin{bmatrix} E_{ex11} \\ E_{ex22} \\ 2E_{ex12} \end{bmatrix} \quad (5.11)$$

$$\hat{\mathbf{g}} = \begin{bmatrix} g_{11}/2 \\ g_{22}/2 \\ g_{12} \end{bmatrix} \quad (5.12)$$

Therefore, the modified Green-Lagrange strain is written as

$$\hat{\mathbf{E}}_{ex} = \hat{\mathbf{E}} + (\mathbf{M}_I + \mathbf{M}_{II}) \cdot \hat{\mathbf{g}} \quad (5.13)$$

where

$$\mathbf{M}_I = \mathbf{B} \begin{bmatrix} 2W_1 W^1 & 0 & W_1 W^2 \\ 0 & 2W_2 W^2 & W_2 W^1 \\ 2W_2 W^1 & 2W_1 W^2 & W_1 W^1 + W_2 W^2 \end{bmatrix}, \quad (5.14)$$

$$\mathbf{M}_{II} = \mathbf{B}^2 \begin{bmatrix} W_1 W_1 (W^1 W^1) & W_1 W_1 (W^2 W^2) & W_1 W_1 (W^1 W^2) \\ W_2 W_2 (W^1 W^1) & W_2 W_2 (W^2 W^2) & W_2 W_2 (W^1 W^2) \\ 2W_1 W_2 (W^1 W^1) & 2W_1 W_2 (W^2 W^2) & 2W_1 W_2 (W^1 W^2) \end{bmatrix} \quad (5.15)$$

note that W_a and W^a are the covariant and contravariant components of vector $\mathbf{W}$, respectively.

From the above explanation, the tangent stiffness matrix will be derived from modified Green-Lagrange strain tensor according to equation (4.13). Non-linear finite element analysis can be conducted by the derived tangent stiffness matrix.

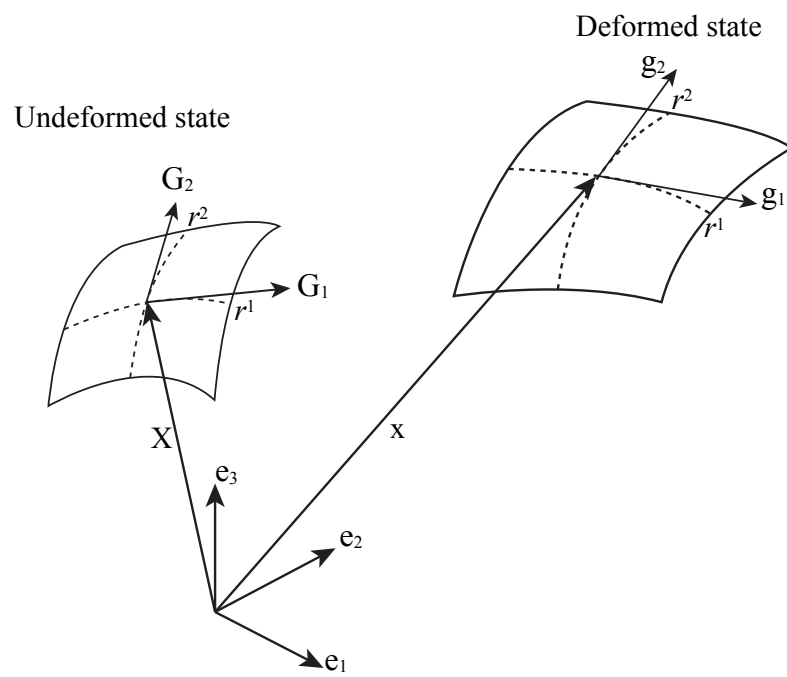


Figure 5.4. Deformation of membrane.

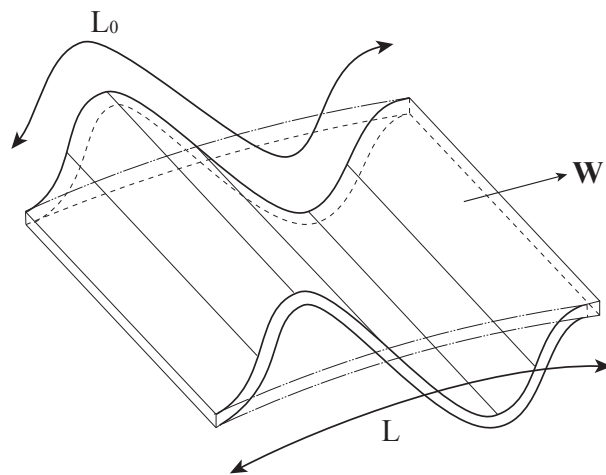


Figure 5.5. Membrane with excess skin along direction W .