CUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES

MSc THESIS

Tamer KANTARCI

REDESIGN OF HIGH PRESSURE PIPES AND CONNECTIONS FOR WATER-JET CUTTING SYSTEM

DEPARTMENT OF MECHANICAL ENGINEERING

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Tamer KANTARCI

A MASTER OF SCIENCE THESIS

DEPARTMENT OF MECHANICAL ENGINEERING

We certify that the thesis titled above was reviewed and approved for the award of degree of the Master of Science by the board of jury on 03/05/2010

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ABSTRACT

MSc THESIS

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DEPARTMENT OF MECHANICAL ENGINEERING
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Some of the advantages that the water-jet cutting systems provide in industrial area are: wide scale of cutting materials, any form can be cut easily, no tool change or wear, no thermal distortion. For these advantages water-jet cutting technology is getting spread in the industrial usage. A singlelayer prototype was produced in Mechanical Engineering Department Laboratory in the University of Çukurova which has the operating pressure of 180 MPa. Later a double-layer intensifier was produced to increase the operating pressure up to the 300 MPa. In this study, the high pressure connection parts which works under 300 Mpa is redesigned using reverse engineering and redesign methodology, a program is written for conical shrink-fit parts and using this program the conical form parts' optimum parameters are found in the water-jet system. Beside this, basic control parameters and some other parameters relationship was investigated.

Key Words: Water-jet Cutting, Redesign Technique, Shrink-fit

ÖZ

YÜKSEK LİSANS TEZİ

SU JETİ SİSTEMLERİNDE YÜKSEK BASINÇ BORULARIN VE BAĞLANTILARININ YENİDEN TASARIMI

Tamer KANTARCI

ÇUKUROVA ÜNİVERSİTESİ
FEN BİLİMLERİ ENSTİTÜSÜ
MAKİNE MÜHENDİSLİĞİ ANABİLİM DALI

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Prof. Dr. Melih BAYRAMOĞLU
Doç. Dr. Ali KOKANGÜL

Su jeti sistemlerinin, özellikle endüstriyel alanda sağladıkları avantajlardan bazıları söyle sıralanabilir: kesim yelpazesinin çok geniş olması, istenen şeklin kolayca oluşturulabilmesi, takım aşınması probleminin olmaması, ısıl gerilmelere sebebiyet vermemesi. Bu avantajlardan dolayı, su jeti kesme sistemleri endüstriyel alanda giderek yaygınlaşmaktadır. Çukurova Üniversitesi Makine Mühendisliği Bölümü laboratuarında öncelikle tek katlı ve 180 MPa çalışma basıncına sahip bir ilk örnek üretilmiştir. Ardından iki katlı bir ilk örnek üretilerek çalışma basıncın 300 MPa değerine çekilmiştir. Bu çalışmada 300 Mpa çalışma basıncına sahip olan su jeti sisteminin yüksek basınç bağlantı elemanları yeniden tasarlanmış, konik sıkı geçen yapılar için bir program yazılmış ve bu programla sistemdeki konik yapıdaki elemanların optimum değerleri bulunmuştur. Ayrıca, temel kontrol parametrelerinin diğer parametrelerle ilişkileri incelenmiştir.

Anahtar Kelimeler: Su Jeti, Yeniden Tasarım Tekniği, Sıkı Geçme.

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1. INTRODUCTION

Non-traditional machining processes which plasma, laser, electron beam and (abrasive) water-jet machining are some of the most important techniques in modern industry have been developed in recent years.

The popularity of water jet cutting comes from its various distinct advantages over the other cutting technologies such as it generates no heat which means water-jet cutting (WJC) is a unique ‘cold’ cutting process, ability to cut several materials without modifying their mechanical properties, the work-piece does not deform during machining, the process can be initiated anywhere on the work-piece, high machining versatility, high flexibility no pre-machining preparation is needed.

Water jetting is, in its simplest form, concerned with the development, the transmission and the application of power. This power is normally created in a water medium by a pump, pushing a given volume of water into a high pressure feed line and providing it with a certain amount of energy in the process. This water flows down through the line, usually a strong metal tube over at least part of its length, to a nozzle. The nozzle contains one or more exit holes or orifices which are normally of a much smaller size than the feed line. Since a constant volume of water reaches the nozzle, it must accelerate to a higher speed in order to escape through these orifices, which also serve to focus the water into a coherent stream or jet and to direct the streams towards the required point on the target surface or work piece.

This thesis performs to design high pressure pipes, check valves and connection parts for water-jet system based on reverse engineering and redesign methodology. In this study, the optimum dimensions of check valves, pipes and all connectors considering stresses will be determined and the parts will be redesigned for ease of assembly and disassembly, for reducing manufacturing and material costs etc using reverse engineering and redesign methodology. Addition to these, a computer program will be developed to determine shrink fit assemblies for conical structures. This study will be the first study to apply “reverse engineering and redesign methodology” to connectors and ultra-high pressure check-valves.

1.1 Water-jetting Technology

1.1.1 History

The development about water-jet cutting process was pioneered in the late 1960s by Dr. Norman C. Franz who is regarded as the father of the water-jet at University of Michigan have been examined the cutting of wood with high velocity jets. He got the idea from the way steam leaks were detected on invisible spots. A broom was moved through the locations where the leaks were expected. By the damage to the broom the idea came up that a jet of high velocity water could also cut materials. He was the first person who studied the use of ultrahigh-pressure water as a cutting tool.

But the first commercial application was installed by McCartney Manufacturing Co. (Baxter Springs, KS) a division of Ingersoll-Rand Co. In late 1971, the system was installed at AJton Box Board Co.s Paper tube division. (Catherine 1987).

In 1979, Dr. Mohamed Hashish working at Flow Research began researching methods to increase the cutting power of the water-jet so it could cut metals, and other hard materials. Dr. Hashish invented the process of adding abrasives to the plain water-jet. He used garnet abrasives, a material commonly used on sandpaper. With this method, the water-jet (containing abrasives) could cut virtually any material. The added abrasives increased the range of materials that can be cut with a water-jet drastically. High traverse speed, thicker materials and better edge quality could be achieved. A new technology was born: Abrasive Water-jet Machining (AWJM). In 1980, abrasive water-jets were used for the first time to cut steel, glass, and concrete. In 1983, the world's first commercial abrasive water-jet cutting system was sold for cutting automotive glass. The first adopters of the technology were primarily in the aviation and space industries which found the water-jet a perfect tool for cutting high strength materials such as Inconel, stainless steel, and titanium as well as high strength light-weight composites such as carbon fiber composites used on military aircraft and now used on commercial airplanes. Since then, abrasive water-jets have been introduced into many other industries such as job-shop, stone, tile, glass, jet engine, construction, nuclear, and shipyard, to name a few (FlowCorp).

1.1.2 Water-Jet Cutting (WJM)

Water-jet is a generic term used to describe equipment that uses a high pressure stream of water directed through cutting head toward the material to be cut or clean purposes. Water jet machining (WJM), is a new technology that is quickly gaining wide industrial acceptance (Hoogstrate and Lutttervelt 1997, Kovacevic et al. 1997).

Most water-jet cutting theories explain water-jet cutting as a form of micro erosion. Water-jet cutting works by forcing a large volume of water through a small orifice in the nozzle. The constant volume of water traveling through a reduced cross sectional area causes the particles to rapidly accelerate. This accelerated stream leaving the nozzle impacts the material to be cut. The extreme pressure of the accelerated water particles contacts a small area of the work piece in this small area the work piece develops small cracks due to stream impact. The water-jet washes away the material that "erodes" from the surface of the work piece. The crack caused by the water-jet impact is now exposed to the water-jet. The extreme pressure and impact of particles in the following stream cause the small crack to propagate until the material is cut through. Figure 1.1 shows the schematic representation of the water-jet cutting system.

The inlet water for a pure water-jet is pressurized between 1300 to 6200 bar. This is forced through a tiny hole in the jewel, which is typically 0.18 to 0.4 mm in diameter. This creates a very high-velocity, very thin beam of water traveling as close to the speed of sound 960 km/hr. Pure water-jet use the beam of water exiting the orifice to cut soft material like disposable diapers, tissue paper, candy bars, automotive interiors and thin soft wood, but are not effective for cutting harder materials. In the cases of tissue paper and disposable diapers the water-jet process creates less moisture on the material than touching or breathing on it. Figure 1.2 shows the typical design of a pure water-jet nozzle.

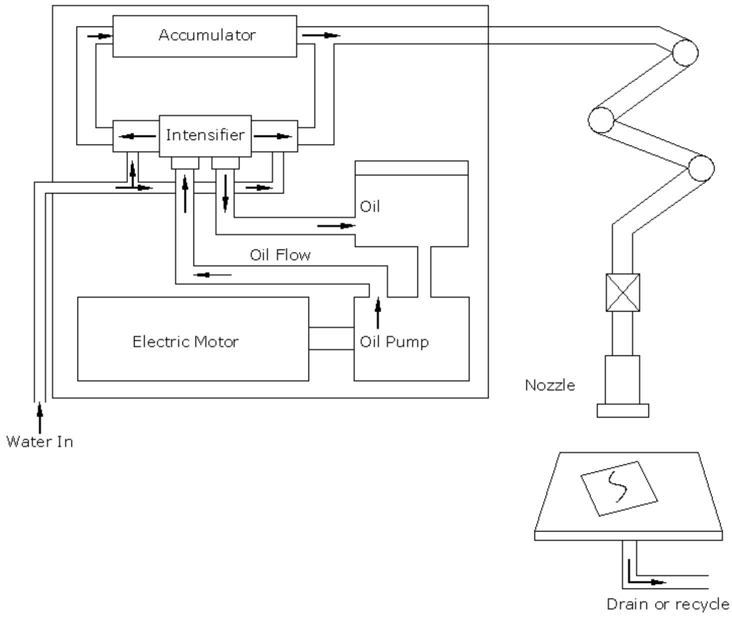

Figure 1.1 Schematic representation of the water-jet cutting

Pure water-jet attributes are as follows:

- Very thin stream (0.004 to 0.010 inch in diameter is the common range)
- Extremely detailed geometry
- Very little material loss due to cutting
- Non-heat cutting
- Cut very thick
- Cut very thin
- Usually cuts very quickly
- Able to cut soft, light materials (e.g., fiberglass insulation up to 24” thick)
- Extremely low cutting forces
- Simple fixturing

1.1.3 Abrasive Water-Jet Cutting (AWJM)

Abrasive water-jet (AWJ) is the improved process of water-jet cutting by means of adding some abrasive particles to the streaming water jet. An abrasive water-jet starts out the same as a pure water-jet. As the thin, high-velocity stream of water leaves the jewel creates a vacuum which pulls abrasive from the abrasive line, however, abrasive is added to the stream and mixes with the water in the mixing tube.

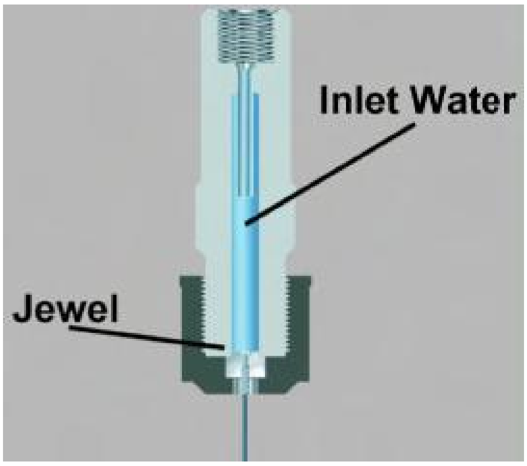

Figure 1.2 Typical design of a pure water-jet nozzle

This increases the cutting ability of the jet. The beam of water accelerates abrasive particles to speeds fast enough to cut through much harder materials.The cutting action of an abrasive-jet is two-fold. The force of the water and abrasive erodes the material, even if the jet is stationary (which is how the material is initially pierced). The cutting action is greatly enhanced if the abrasive-jet stream is moved across the material and the ideal speed of movement depends on a variety of factors, including the material, the shape of the part, the water pressure and the type of abrasive. Controlling the speed of the abrasive-jet nozzle is crucial to efficient and economical machining.

The abrasive water-jet differs from the pure water-jet in just a few ways. In pure water-jet, the supersonic stream erodes the material. In the abrasive water-jet, the water-jet stream accelerates abrasive particles and those particles, not the water, erode the material. The abrasive water-jet is more powerful than a pure water-jet.

Both the water-jet and the abrasive water-jet have their place. The pure water-jet cuts soft materials, the abrasive water-jet cuts hard materials, such as metals, stone, composites and ceramics. Abrasive water-jets using standard parameters can cut materials with hardness up to and slightly beyond aluminum oxide ceramic (often called alumina). Figure 1.3 shows the typical design of abrasive water-jet nozzle.

Abrasive water-jet attributes are provided below:

- Extremely versatile process
- No Heat Affected Zones
- No mechanical stresses
- Easy to program
- Thin stream (0.020to 0.050 inch in diameter)
- Extremely detailed geometry
- Thin material cutting
- 10 inch thick cutting
- Stack cutting
- Little material loss due to cutting
- Simple to fixture
- Low cutting forces (under 1 lb. while cutting)
- One jet setup for nearly all abrasive jet jobs
- Easily switched from single to multi-head use
- Quickly switch from pure water-jet to abrasive water-jet
- Reduced secondary operations
- Little or no burr

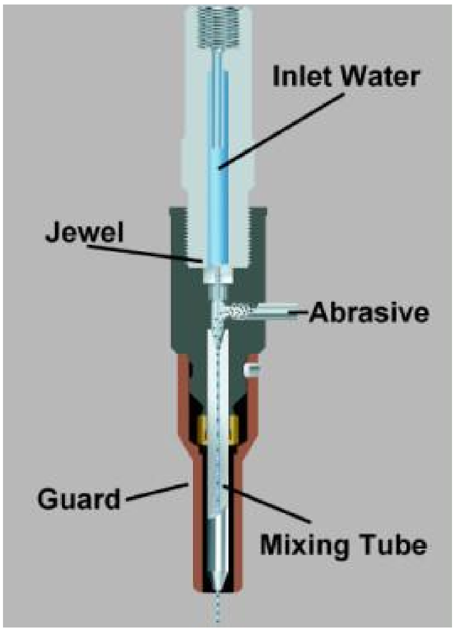

Figure 1.3 A diagram of abrasive water-jet nozzle

1.1.4 Advantages of Water-Jet Cutting

- **Fast Turnaround**: No waiting on dies or fixtures or even programming, since the machine works directly from your CAD files.
- **High Precision:** Accurate to +/- 0.003" for parts under 1' and +/- 0.005" for parts over 1' long.
- **Smooth Finish:** Cutting with water-jet leaves a smooth, sandblasted edge, with virtually no burr, eliminating finishing steps.
- **Material Integrity:** Since no heat is used there is no hardening, no burning, and no property changes.
- **Less Waste:** Kerf width of only 0.030" and tight part nesting allows maximum usage of material, especially important for expensive materials, such as titanium.
- **Environmentally Friendly:** Uses only water, garnet and electricity. No cutting fluids, oils, fumes, or burning.
- **Low Cost:** No costly dies or fixtures, and fast machining. The water-jet is programmed directly from your CAD files, greatly reducing CNC programming costs.
- **Single Price Minimum:** The fast setup, easy programming and machine speed allows for economical one-off production...great for prototyping.
- **Cuts Virtually Any Materials:** Water-jet can machine virtually any material, any thickness, tool steel to aluminum, glass to titanium seen in Table 1.1.

1.1.5 Disadvantages of Water-Jet Cutting

High pressure water-jetting systems have high investment cost, are still relatively expensive and therefore, for single small operations they are not a practical tool and the time to cut a part can be very long.

Very thick parts cannot be cut with water-jet cutting. If the part is too thick, the jet may dissipate and depth control is difficult. This phenomena cause to cut on diagonal or wider cut at the bottom of the part. It can also cause a ruff wave pattern on the cut surface.

Table 1.1 The cutting material with water-jet cutting systems

o Mild Steel	o Cork
o Stainless Steel	o Spring Steel
o Tool Steel	o Lexan
o Aluminum	o Polycarbonate
o Brass	o Acrylic
o Bronze	o Reflective Metals
o Alloys	o Glass
o Wood	o Rubber
o Derlin	o Ceramic
o Phenolics	o Composites
o Laminates	o Any Other Materials

Tapered cutting is also a problem in water-jet cutting for very thick materials. Taper is occurred when the water-jet exits the part from different angle. This means that limited cutting speed and quality. Figure 1.4 shows the cone generation in very thick materials instead of desires cylindrical shape during cutting.

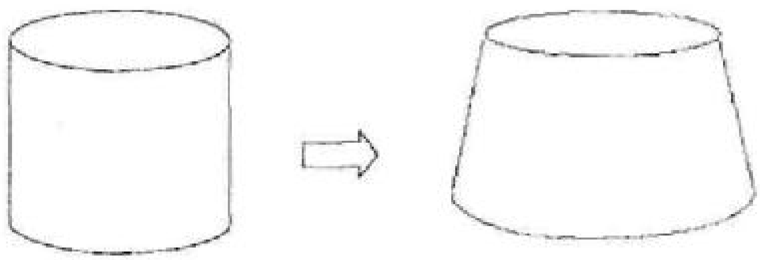

Figure 1.4 Schmeatic view of cone generation instead of cylindrical shape

Another error caused by the jet is the creation of a "V" shaped taper, sometimes called a draft angle. The jet loses power as it travels through the material. Maximum cutting speed the greater the taper will be. Figure 1.5 shows the taper generation.

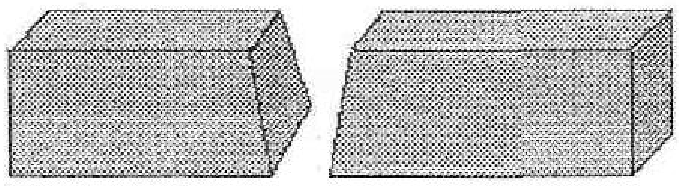

Figure 1.5 "V" shaped taper

Another disadvantages are noise pollutionand limited nozzle life because of high-pressure.

1.1.6 Application of Water-Jet Cutting

- Automotive Industry
- Aerospace Industry
- Food Industry
- Electronic Industry
- Textiles Industry
- Mining Applications
- Safety, Health and Medical Applications
- Stone, Glass and Metal Artwork
- Metals, Exotic and Nontraditional Materials Cutting
- Foam Product Cutting
- Fiberglass Cutting
- Slitting Operations
- Munitions Demilitarization Application

1.2 Developed Water-jet Cutting Systems in Mechanical Engineering Department in Çukurova University

1.2.1 Prototype of Water-jet Cutting System with 1400 Bar Operating Pressure

A very successful water-jet first prototype water-jet cutting system was designed and manufactured in Mechanical Engineering Department of University of Çukurova in 1999.

The first prototype had single layer double acting intensifier. So the pressure was limited to 140 MPa (1400 bars). The developed system has been able to cut the soft metals like aluminum and brass.

The water-jet cutting system consists of an intensifier, check valves for low and high pressures, a hydraulic pressure generation system, a programmable logic controller, a cutting nozzle, two proximity sensors, oil hoses and high pressure water hoses which is shown in Figure 1.6.

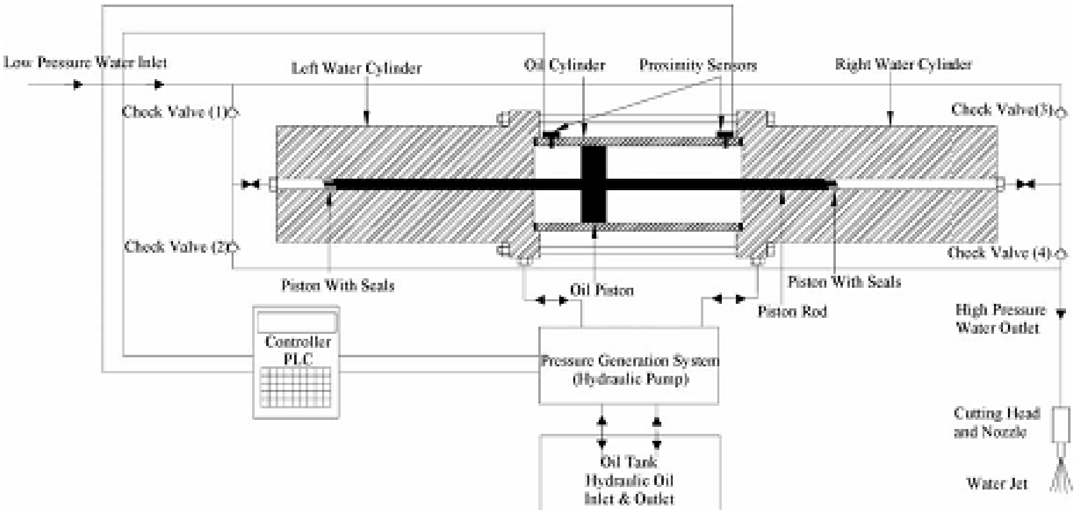

Figure 1.6.Water-jet Cutting System main parts

The intensifier of the prototype consists of four main parts: An oil cylinder which is located in the middle, two water cylinders that are located at the both sides of the oil cylinder and a piston which is located inside through the cylinders.

Adapters and "T" connections are auxiliary parts of the system. Check valves are located in "T" connections.

The operation principle is as follows: When the piston moves to the left high pressure is obtained on the left side. So the check valve 1, located inside the "T" connection 1, and the check valve 4, located inside the "T" connection 4, are opened. Other check valves are closed. When high pressure water is pumped to the line, low pressure water is sucked into the intensifier. At the end of the stroke, hydraulic directional control valve changes the direction of the piston. Then the high pressure is obtained on the right side. So the check valve 2, located inside the "T" connection 2, and the check valve 3, located inside the "T" connection 3, are opened. Other check valves are closed. As before, when high pressure water is pumped to the line, low pressure water is sucked into the intensifier. The continuous operation cycle of the intensifier provides continuous water through the nozzle.

1.2.2 Prototype of Water-jet Cutting System with 3000 Bar Operating Pressure

Although the first prototype water-jet cutting system in Çukurova University is operated successfully, for ultra-high pressures a new design is needed. So the second prototype water-jet cutting system for ultra-high pressures (3000 bars operating pressure) was designed and manufactured in the Mechanical Engineering Department of University of Çukurova in 2002 (Tutal, 2002).

Single layer intensifier can sustain only up to 1400 bars (Kırbas, 1999). The multi-layer intensifiers were used for the second prototype to sustain up to 3000 bars.

The intensifier of the second prototype consists of 6 main parts: an oil cylinder, located in the middle, 2 water cylinders, located at the both sides of the cylinders, piston assembly, located inside through the cylinders and heads and sealing heads located at the ends of the water cylinders.

At this prototype double layer water cylinder was used instead of single layer water cylinder. And more, adapters and T-Joint are canceled and instead of that heads and sealing had is designed and used which the design make very effective.

Figure 1.7 shows the prototype waterjet cutting system with 3000 bar operating pressure.

The working principle of the second prototype is the same as the first prototype. As the piston-assembly moves to one side the high pressure is generated and low pressure water is sucked into the intensifier. At the end of the stroke the hydraulic directional control valve changes the direction of the piston and high pressure is generated on the opposite side of the intensifier. The continuous operation cycle of the intensifier provides continuous water through the nozzle.

1.3 Redesign and Reverse Engineering Methodology

The reverse engineering and redesign methodology has three distinct phases. These phases are; reverse engineering, modeling and analysis, and redesign. The intent of the first phase, reverse engineering, is twofold. First, a product is treated as a black box, experienced over its operating parameters, and studied with respect to customer needs and predicted and/or hypothesized functionality, product components, and physical principles. The second step of the reverse engineering phase is to experience the actual product in both function and form. This sub-phase includes the full disassembly of the product, design for manufacturing analysis, further functional analysis, and the generation of final design specifications. The second stage of the methodology entails the development and execution of design models, analysis strategies, model calibration, and experimentation. The third and final stage of the methodology then initiates product redesign based on the results of the reverse engineering and modeling phases.

The following stages are used in Reverse engineering and redesign methodology:

Reverse Engineering Stage as follows:

a) **Investigation**: To create black-box model.

b) **Product tear-down and experimentation:** To disassembly the systems and prepare bill of material.

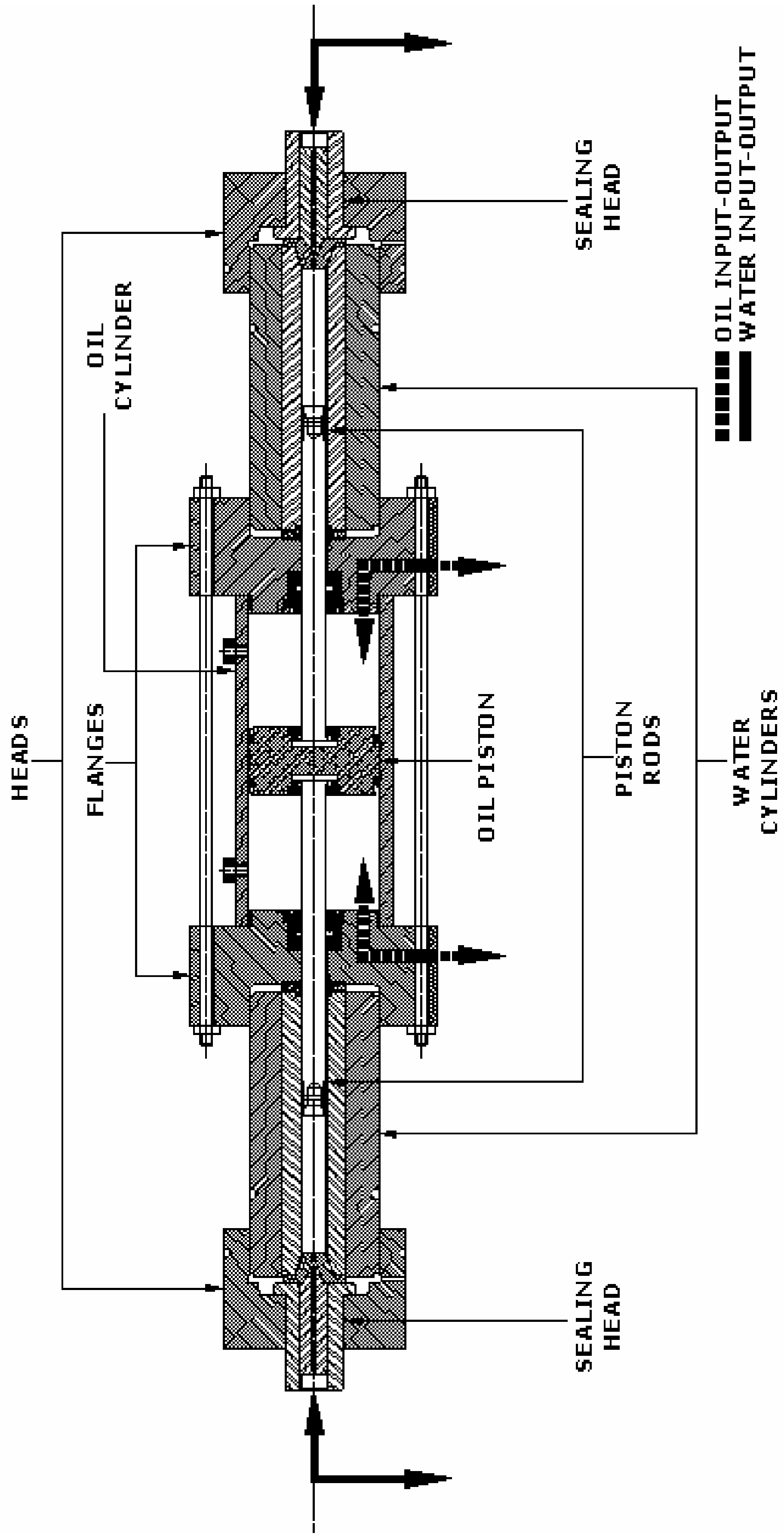

Figure 1.7 Prototype Waterjet Cutting System with 3000 Bar Operating Pressure (Tutal, 2002)

The disassembly is systematically executed, listing the order of disassembly, the component to be removed, tool usage, access direction and any permanent deformation caused by the disassembly. Each part is analyzed such way: possibility to redesign the part to be easier to assemble.

c) Functional analysis: To develop energy flow diagram and create sub-function chain for each flow.

d) Constraint propagation: To create morphological matrix.

e) Forming engineering specifications: To conceive input and output flows for each sub-functions and list for each sub-function row as the metrics for the sub-function.

Modeling and Analysis Stage as below:

f) Modeling and analysis: To perform adaptive or original design changes before creating and optimizing design for some system parts.

In this stage the developed software program that calculates the required dimensions of the inner and the outer diameter of pipes and other calculations will be used.

Redesign Stage is:

g) Redesign: To product integration of new concept based on the result parametric and adaptive redesign obtained from steps 1-7.

1.4 Scope of Study

The main part of the WJM is the intensifier. The intensifier receives low pressure water and can increase the water pressure up to 3000 bar. The water flow into the intensifier and output from the intensifier is controlled by check valves. There are four check-valves in a WJM system. Briefly, functions of check valves are to control high and low pressure flows. The intensifier, check-valves, and the cutting nozzle are connected to each other through many connections. All of these parts are subjected to ultra-high pressures and the parts require frequent servicing. As a result of this frequent disassembly of the parts are the essential requirements.

This thesis performs to design high pressure pipes, check valves and connection parts for water-jet system based on reverse engineering and redesign methodology. The study will not consider the design of the intensifier. It will concentrate on multi-layer design of check valves, and other connections. The study will consider both the stresses for minimum cost and ease of assembly and disassembly of the all connectors. In addition to these it will be the first study to apply "reverse engineering and redesign methodology" to connectors and ultra-high pressure check-valves.

The study can be divided into two parts to achieve the aim of the study. These are;

I) Determining the optimum dimensions of check valves, pipes and all connectors considering stresses.

II) Redesigning the parts for ease of assembly and disassembly, for reducing manufacturing and material costs etc using reverse engineering and redesign methodology.

The first aim will be achieved using a software program which was developed recently for shrink-fitted assemblies. Since all the connections and check valves are to be used under ultra high pressure, shrink-fitted assemblies will be used for the design of check valves. The program will be used to determine the thicknesses of double-layer cylinders based on the desired factor of safeties for optimum dimensions.

The second aim will be achieved using the methodology developed by Otto and Wood since it is gaining a wide acceptance internationally. Although, the definition of design and redesign varies and little agreement exists on how to use and apply a structured problem-solving methodology for design and redesign problems a methodology which has been developed by Otto and Wood will be used in this study as part of a methodology.

The methodology is expected to provide better designs for the target components of WJM system. It may lead to parametric, adaptive or original design changes in the following parts of the WJM system developed as first prototype in the mechanical Engineering department of Çukurova University.

- Head
- Sealing head
- Check valves and check valve housing blocks
- Pipes
- Connectors between intensifier and check valve block
- Other connection parts of the WJM system.

2. PREVIOUS STUDIES

2.1 Design Methodologies

In the literature a number of descriptive and prescriptive design methodologies have been developed for general engineering design problems (Pahl and Beitz, 1984; Pugh, 1991; Ullman, 1992; Ulrich and Eppinger, 1994; Asimow, 1962;.Altschuller, 1984; Dixon and Finger, 1989). Basic phase or "waterfall" models describe the entirety of design as sequences of stages from need analysis to conceptual design to final manufacture. Rather than adopting this universal view, other researchers consider different types of design process and methods. Pahl and Beitz (1984) describe a very detailed design process that works well for "medium" complexity systems. Ulrich and Eppinger (1994) describe formulation, evaluation, and completion of design decision, including customer and manufacturing information. Clausing (1994) describes a total quality approach. Papaiambros and Wilde (1988) describe methods to optimize product design models. Phadke (1989) describe experimental design methods.

The existing state of the art in design science is characterized by much stronger methods than theory. Some methods, such as those of axiomatic design are based on an explicit theory (Suh 1990, 2001). Other methods, such as Quality Function Deployment (Clausing 1994) are empirical in nature, derived from what appears to work in practice, rather than any underlying theory. Similarly, advances in design-for-x methods are typically ad hoc in nature, developed only in response to a particular need ‘‘for x’’. Herbert Simon, in criticizing the lack of theoretical basis behind design methods in the 1960s described such ad hoc methods as ‘‘cook-booky’’ (Simon 1996). This is in contrast to the methods used in other engineering sciences, and even mathematics, where methods to solve a particular problem, say of heat transfer, are based on an underlying theory, for example, Newton’s Law of Cooling. We propose that more rigorous methods in design should likewise be based on theory first, and then validated and refined by their use in industry.

While these methodologies are applicable to different types of design problems, they tend to emphasize either design problems that seek “original solutions” or well-posed parametric formulations. This emphasis provides a foundation for teaching engineering design; it also provides a possible approach for establishing corporate design processes in industry. For the class of problems known as redesign (adaptive, variant, etc.), however, an emphasis on original design may be too general. Sferro, Bolling, and Crawford (1993) argue the legitimacy of this claim, based on an analysis of the current variant design processes in the automobile industry. They develop a new variant design methodology, referred to as Direct Engineering, to replace more general original design methods.

Redesign is a common situation. As a result on decisions made at design reviews the details of the design are changed many times as prototypes are developed and tested. There are two categories of redesign: fixed and updates. A fixed is a design modification that is required due to less than acceptable performance once the product has been introduced into the market place. On the other hand, updates are usually planned as a part of the products life cycle before the product is introduced to the market. An update may be add capacity and improve performance to the product are improve its appearance to keep it competitive.

The most common situation in redesign is the modification of an existing product to meet new requirements. For example, the banning of the use of fluorinated hydrocarbons refrigerants because of the "ozone-hole problem" required the extensive redesign of refrigerant system. Often redesign result from failure of the product in service. A much simpler situation is the case where one or two dimensions of a component must be changed to match some change made by the customer of the part. Yet another situation is the continuous evolution of a design to improve the performance (Dieter 1999).

Organizations are seeking ways to redesign their business processes in order to streamline and simplify operations and remain competitive in a changing environment. The major motivation behind the sudden popularity of such process redesign efforts is the recognition that markets are becoming increasingly global and competitive, and that reacting to changes in these markets quickly is critical for an organization’s survival. Many of the benefits reported by firms that have undertaken these redesign efforts are measured along the lines of reduction in labor force, quicker service, reduced cycle time, etc. Hammer and Champy discuss several cases: IBM’s credit, Ford’s AP function, and Kodak’s product development, to illustrate how organizations have dramatically improved their performance by radically redesigning their business operations. Many of these organizations have combined both process simplification and the use of appropriate information technologies to realize these gains. Rowan also describes how Texas Instruments has redefined its manufacturing, product design, customer support, etc. in the design of a chip, which supports multiple technologies, in a shorter time frame. This case emphasizes the need to integrate multiple activities across business functions to meet a single outcome – the satisfaction of customer needs. The general implication from many of these case studies is that business processes have become complex and unwieldy over time, and only a radical departure from its current environment can help an organization realize significant gains in performance. One of the major issues in redesign is, hence, the identification of a process vision that will radically simplify the current business processes and yield dramatic gains in performance.

As with original design, redesign problems include the process step of "gathering customer needs", "specification planning and development", "benchmarking", "concept generation", "product embodiment", "prototype construction and testing", and "design for manufacturing", but they also focus on additional step, referred to here as "reverse engineering" (Ingle, 1994). Reverse engineering initiates the redesign process, where in a product is predicted, observed, disassembled, analyzed, tested, experienced, and documented in terms of its functionality, form, physical principles, manufacturability and assemblability. The intend of this process step is to fully understand and represent the current instantiation of a product. Based on the resulting representation and understanding a product may be evolved, either at the subsystem, configuration, component or parametric level.

Reverse engineering and redesign methodology focus on the process steps needed to understand and represent a current product. Extension of contemporarily techniques in engineering design is utilized at a number of stages in the redesign process to meet this goal (Wood and Otto, 1999).

To motivate the need for a redesign methodology, consider the abstraction of product evolution in the market place, depicted in Figure 2.1 one can determine a critical metric that is useful for evaluating a product and its market competition, and the plot of the performance for each product as a function of the time when each product was introduced. The metric values will naturally fail as an S-curve in time (Foster, 1986; Betz, 1993). At first, a new innovative product will enter a market domain as a new concept. The plotted curve will remain somewhat "flat" for a certain period of time, representing the time for the competition to respond. Next, a rapid profusion of innovation occurs, and many products are launched in time.

The lower leg of the "S" is forming. The new technology, however, eventually tops out, physical laws of the process dominate, and the engineers cannot extract more performance. The slope of the "S" tops out again, and the curve becomes flatter.

Depending on the competitive environment, a product development team must redesign their product at two levels into the product's future. The first level is as described along an individual S-curve and includes parametric, variant, and minor adaptive changes in the manufacturing, components, materials, geometry, assemblies, and subassemblies. These changes are a direct response to customer needs and feedback (Ashley, 1994). The second level, on the other hand, reflects a discontinuous jump in the product characteristics. Discontinuities of this type result from the introduction of new technologies, new production processes, or a fundamental change in product architecture.

The bottom line of product evolution with respect to S-curves is that all products must change (both nonlinearly along an S-curve and discontinuously between them) to remain competitive. We propose that a systematic methodology will lead to a better understanding of product evolution and how to execute effective change with reduced cycle time. The next subsection introduces the structure of such a methodology.

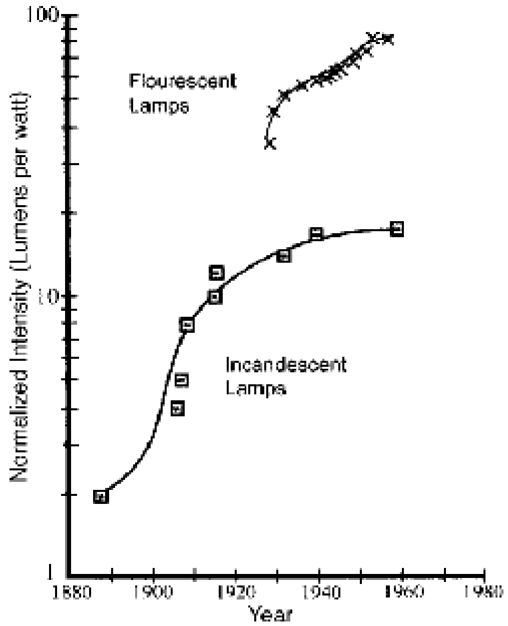

Figure 2.1 Product Evolutions along S-Curves, from Betz, (1993).

2.1.1 Redesign and Reverse Engineering Methodology

Figure 2.2 shows the general composition of our reverse engineering and redesign methodology. Three distinct phases embody the methodology: reverse engineering, modeling and analysis, and redesign. The intent of the first phase, reverse engineering, is twofold. First, a product is treated as a black box, experienced over its operating parameters, and studied with respect to customer needs and predicted and/or hypothesized functionality, product components, and physical principles. The second step of the reverse engineering phase is to experience the actual product in both function and form. This sub-phase includes the full disassembly of the product, design for manufacturing analysis, further functional analysis, and the generation of final design specifications.

The second stage of the methodology entails the development and execution of design models, analysis strategies, model calibration, and experimentation. The third and final stage of the methodology then initiates product redesign based on the results of the reverse engineering and modeling phases. Parametric redesign may be pursued using optimization analysis of the design models. Alternatively or in concert, adaptive redesign of product components and subassemblies may be pursued.

Beyond parametric or adaptive redesign, an original redesign effort may be needed to satisfy the customer needs. An original redesign, in this context, implies that a major conflict exist between the customer needs and the current product in the market. Because of this conflict, it is deemed that an entirely new product concept is needed. Functional analyses from the reverse engineering phase will direct the redesign effort.

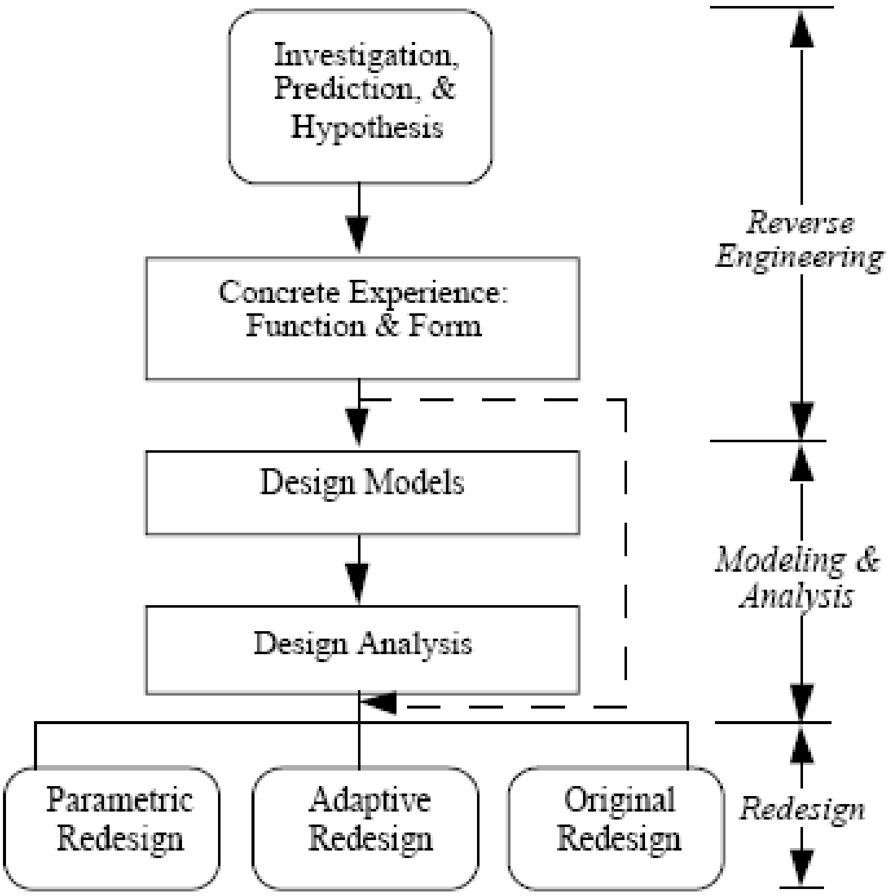

Figure 2.2 Reverse Engineering and Redesign Methodology (Otto and Wood, 1999)

It is based on a ten-step process with three primary phases. Each phase provides a clear set of tasks to seek new product configurations, combining contemporary design techniques with new applications and extensions developed

2.2 Design for Manufacture, Assembly-Disassembly and Maintenance Perspectives.

2.2.1 Design for Assembly

The essential of assembly is based on the premise that the lowest assembly cost can be achieved by designing a product in such a way that it can be economically assembled the most appropriate assembly system. The manual assembly is the most suitable for our design due to the economical reasons. The following design for assembly criterions are suggested by Corbett (1987).

- Minimize the number of parts and fixing, design variants, assembly movements and assembly directions.
- Provide suitable lead-in chamfers, automatic alignment, easy access for locating surfaces, symmetrical parts or aggregate asymmetry and simple handling and transportation.
- Avoid visual obstruction, simultaneous fitting operations, parts which will tangle or nest, adjustment which effects prior adjustments and the possibility of assembly errors.

2.2.2 Design for Disassembly

Disassembly is defined by Brenna, Gupta and Taleb (1994) as, the process of systematic removal of desirable constitute parts from an assembly while ensuring that there is no impairment of parts due to the process. Leonard (1991) reported that two basic methods of disassembly could be used: reverse assembly and brute force. In the case of reverse assembly, if a fastener is screwed in, then it is screwed out; if two parts are snap fit together, then they are snapped apart. While in the case of brute force, parts are just pulled or cut.

2.2.3 Design for Manufacture

The selection of appropriate processes for the manufacture of particular part is based upon the matching of the required attributes of the part and the various process capabilities. These processes include raw material selection, process selections, modular design, Standard component usage, multi-use part development, separate fasteners usage and assembly direction minimization. Kirkland (1998) provided the factors which influence of designer's selection of a particular material which includes

- Raw material selection
- Process selection
- Developed a modular design.
- Use standard component
- Design part to be usable
- Avoid a separate fastener
- Minimizing assembly directions (all parts should be assembled from one direction whenever possible)

2.2.4 Design for Maintainability

Maintainability can be defined as the probability that a failed system can be repaired in a specific interval of down time. The basic objective of design of maintainability is to assure that the product can be maintained throughout its useful life-cycle at reasonable expense without any difficulty.

The following is a list of design for manufacture guide lines that provides designers with specific guidance regarding design requirements.

1. General design features
    - The design shall preclude the possibility of damage to the equipment during maintenance and servicing.
    - Minimize the needs for special tools.

- Part reference designations shall be located next to each part legibly and permanently.
- Keying, size or shape shall be used to ensure that removable parts are reassembled in the correct position.
- Guide pins shall be provided for alignment of modules or high-density connectors.
- Handless shall be provided for removable units weighting over 4.5 kg pounds or whose shape makes them difficult to handle.
- Sharp edge, corners, or protrusions that could case injury to personal shall be avoided.

2. Mounting and location of units
   - Provide for the removal and replacement of line replaceable unit (LRU) without removal of unfailed units.
   - Provide for the removal and replacement of line LRUs without interrupting critical functions.
   - Provide clear access to all LRU locations. Mount units to chassis or structure rather than on other units.
   - Mount heavy units as low as possible. Label access for limits.

2.3 Problem Solving Method

2.3.1 Theory of Inventive Problem Solving (TRIZ)

TRIZ (Theory of Inventive Problem Solving) is a problem solving method based on logic and data, not intuition, which accelerates the project team's ability to solve these problems creatively. TRIZ also provides repeatability, predictability, and reliability due to its structure and algorithmic approach.

Altshuller who is creation of TRIZ, identified creative problem which is the solution to the problem caused another problem.

TRIZ is an international science of creativity that relies on the study of the patterns of problems and solutions, not on the spontaneous and intuitive creativity of individuals or groups. More than three million patents have been analyzed to discover the patterns that predict breakthrough solutions to problems.

The research has proceeded in several stages. The three primary findings of this research are as follows:

- Problems and solutions are repeated across industries and sciences. The classification of the contradictions in each problem predicts the creative solutions to that problem.
- Patterns of technical evolution are repeated across industries and sciences.
- Creative innovations use scientific effects outside the field where they were developed.

In the course of solving any one of technical problem, the 40 Principles of Problem Solving are the most accessible "tool" of TRIZ (see table 2.1). These 40 principles involve some general solutions which were found against some contradictions faced by designers in the world, and lead designer to create solutions against these contradictions encountered during problem solving. A contradictions matrix (database) is also other tool of TRIZ with which designers can decide which principles can solve the contradiction and then follow these principles.

Table 2.1 Principle of TRIZ

1. Segmentation	**A.** Divide an object into dependant parts
	B. Make an object sectional-easy to assemble or disassemble
	C. Increase the degree of fragmentation or segmentation
2. Taking out	**A.** Extract the disturbing part or property from an object
	B. Extract the only necessary part (or property) of an object
3. Local quality	**A.** Change of an object's structure from uniform to non-uniform
	B. Change an action or external influence from uniform to non-uniform
	C. Make each part of an object function in conditions most suitable for operations
	D. Make each part of an object fulfil a different and/or complementary useful function
4. Asymmetry	**A.** Change the shape or properties of an object from symmetrical to asymmetrical
	B. Change the shape or properties of an object to suit external asymmetries (i.e. ergonomic features)
	C. If an object is asymmetrical, increase its degree of asymmetry

5. Merging	**A.** Bring closer together (or merge) identical or similar objects or operations in space
	B. Make objects or operations contiguous or parallel; bnng them together in time
6. Universality	**A.** Make an object perform multiple functions; eliminate the need for other parts
7. Nested doll	**A.** Place one object inside another
	B. Place multiple objects inside another
	C. Make one part pass (dynamically) through a cavity in the other
8. Anti-weight	**A.** To compensate for the weight of an object, merge it with other objects that provide lift
	B. To compensate for the weight of an object, make it interact with the environment (use aerodynamic, hydrodynamic, buoyancy and others)
9. Prior counteraction	**A.** When it is necessary to perform an action with both harmful and useful effects, this should be replaced with counteractions to control harmful effects
	B. Create beforehand stresses in an object that will oppose known undesirable working stresses later on
10. Prior action	**A.** Perform the required change of an object in advance
	B. Pre-arrange objects such that they can come mto action from the most convenient place and without losing time for their delivery
11. Cushion in advance	**A.** Prepare emergency means beforehand to compensate for the relatively low reliability of an object ('belt and braces')
12. Equipotentialy	**A.** If an object has to be raised or lowered, redesign the object's environment so the need to raise or lower is eliminated or performed by the environment
13. The other way round	**A.** Invert the action used to solve the problem
	B. Make movable parts fixed, and fixed parts movable
	C. Turn the object (or process) upside down
14. Speheroidality-Curvature	**A.** Move from flat surfaces to spherical ones and from parts shaped as a cube (parallel piped) to ball-shaped structures
	B. Use rollers, balls, spirals
	C. Go from linear to rotary motion (or vice versa)
	D. Use centrifugal forces
15. Dynamics	**A.** Change the object (or outside environment) for optimal performance at every stage of operation
	B. Divide an object into parts capable of movement relative to each other
	C. Change from immobile to mobile
	D. Increase the degree of free motion
16. Partial or excessive action	**A.** If 100 percent of an object is hard to achieve using a given solution method then, by using 'slightly less' or 'slightly more' of the same method, the problem may be considerably easier to solve
17.Another dimension	**A.** Move into an additional dimension - from one to two - from two three
	B. Go from single storey or layer to multi-storey or multi-layered

	C. Incline an object, lay it on its side
	D. Use the other side
18. Mechanical vibration	**A.** Cause an object to oscillate or vibrate
	B. Increase its frequency (even up to the ultrasonic)
	C. Use an object's resonant frequency
	D. Use piezoelectric vibrators instead of mechanical ones
	E. Use combined ultrasonic and electromagnetic field oscillations
19. Periodic action	**A.** Instead of continuous action, use periodic or pulsating actions
	B. If an action is already periodic, change the periodic magnitude or frequency
	C. Use pauses between actions to perform a different action
20. Continuity of useful action	**A.** Carry on work without a break. All parts of an object operating constantly at full capacity
	B. Eliminate all idle or intermittent motion
21. Rushing through	**A.** Conduct a process, or certain stages of it (i.e. destructible, harmful or hazardous operations) at high speed
22. Bessing in disguise	**A.** Use harmful factors (particularly, harmful or hazardous operations) at high speed
	B. Eliminate the primary harmful action by adding it to another harmful action to resolve the problem
	C. Amplify a harmful factor to such a degree that it is no longer harmful
23. Feedback	**A.** Introduce feedback to improve a process or action
	B. If feedback is already used, change its magnitude or influence in accordance with operatmg conditions
24. Intermediary	**A.** Use and intermediary carrier article or intermediary process
	B. Merge one object temporarily with another (which can be easily removed)
25. Self-service	**A.** An object must service itself by performing auxiliary helpful functions
	B. Use waste resources, energy or substances
26. Copying	**A.** Replace unavailable, expensive or fragile object with available or mexpensive copies
	B. Replace an object, or process with optical copies
	C. If visible optical copies are used, move to infrared or ultraviolet copies
27. Cheap short-living objects	**A.** Replace an expensive object with a multiple of inexpensive objects, compromising certain qualities, such as service life
28. Replace mechanical system	**A.** Replace a mechanical system with a sensory one
	B. Use electnc magnetic and electromagnetic fields to mteract with the object
	C. Replace stationary fields with moving; unstructured fields with structured
	D. Use fields in conjunction with field-activated (e.g. ferromagnetic) particles

29. Pneumatics and hydraulics	**A**. Use gas and liquid parts of an object instead of solid parts (i.e. inflatable, filed with liquids, air cushion, hydrostatic, hydro-reactive)
30. Flexible membranes/thin films	**A.** Use flexile shells and thin films instead of three-dimensional structures
	B. Isolate the object from its external environment usmg flexible membranes
31. Porous materials	**A.** Make an object porous or add porous elements (inserts, coating, etc.)
	B. If an object is already porous, use the pores to introduce a useful substance or function
32.Colour change	**A.** Change the colour of an object or its external environment
	B. Change the transparency of an object or its external environment
	C. In order to improve observability of things that are difficult to see, use coloured additives or luminescent elements
	D. Change the emissivity properties of an object subjected to radiant heating
33. Homogeneity	**A.** Objects interacting with the main object should be of same matenal (or matenal with identical properties)
34. Discarding and recovering	**A.** After completing their function (or becoming useless) reject objects, make tern go away, (discard them by dissolving, evaporating, etc) or modify during the process
	B. Restore consumable/used up parts of an object dunng operation
35. Parameter change	**A.** Change the physical state (e.g. to a gas, liquid, or solid)
	B. Change the concentration or density
	C. Change the degree of flexibility
	D. Change the temperature or volume
	E. Change the pressure
	F. Change other parameters
36. Phase transition	**A.** Use phenomena of phase transitions (e.g. volume changes)
37. Thermal expansion	**A.** Use thermal expansion, or contraction, of matenals
	B. Use multiple matenals with different coefficients of thermal expansion
38. Accelerated oxidation	**A.** Replace common air with oxygen-enriched air
	B. Replace enriched air with pure oxygen
	C. Expose air or oxygen to ionizing radiation
39. Inert atmosphere	**A.** Replace a normal environment with an inert one
	B. Add neutral parts, or inert additives to an object
40. Composite materials	**A.** Change from uniform to composite materials

In the TRIZ method, 39 engineering parameter seen in Table 2.2 which is mentioned above is arranged in matrix format and obtained a square matrix 39X39 sized which is called contradictions matrix. Engineering parameters are existed at the contradiction matrix's rows and columns. Rows are improving features, columns are worsening features. The designers can decide which contradictions can solve and then follow the 40 TRIZ principles.

Table 2.2 Engineering Parameters

1. Weight of moving object	21. Power
2. Weight of stationary object	22. Loss of Energy
3. Length of moving object	23. Loss of substance
4. Length of stationary object	24. Loss of Information
5. Area of moving object	25. Loss of Time
6. Area of stationary object	26. Quantity of substance/the matter
7. Volume of moving object	27. Reliability
8. Volume of stationary object	28. Measurement accuracy
9. Speed	29. Manufacturing precision
10. Force (Intensity)	30. Object-affected harmful factors
11. Stress or pressure	31. Object-generated harmful factors
12. Shape	32. Ease of manufacture
13. Stability of the object's composition	33. Ease of operation
14. Strength	34. Ease of repair
15. Duration of action of moving object	35. Adaptability or versatility
16. Duration of action by stationary object	36. Device complexity
17. Temperature	37. Difficulty of detecting and measuring
18. Illumination intensity	38. Extent of automation
19. Use of energy by moving object	39. Productivity
20. Use of energy by stationary object	

2.3.2 Use Value Analysis

These concepts are used to mark the requirement list and evaluation process. Every solution for the objectives is given a number between 1 and 10 for use-value analysis. The number given indicates how the solution reacts to the objective (weak, good, very good, ideal etc.), as can be seen in Table 2.3. These numbers then is used for evaluation optimum solution.

Table 2.3 Points awarded in use-value analysis (Pahl and Beitz, 2005)

Value Scale	
Points	Meaning
0	absolutely useless solution
1	very inadequate solution
2	weak solution
3	tolerable solution
4	adequate solution
5	satisfactory solution
6	good solution with few drawbacks
7	good solution
8	very good solution
9	solution exceeding the requirement
10	ideal solution

2.4 WJC Redesign Applications

In this section, redesign applications and examples of water-jet cutting is discussed and examined.

Herbig et al., (1999) stated that the reduced number of parts is one of the biggest advantages of the redesign for the end user. In Herbig's (1999) study, the redesign of existing suction and pressure valve (check valves) with exactly same parts but different configuration. The reduced number of parts is one of the biggest advantages of the redesign for the end user. The calculations enabled the determination of the influence of flange design on the resulting stresses (stresses due to the auto frettage, working condition and flange connection) under load condition.

The Standard high pressure pump (3,4 L/min, 400 MPa) normally is used for water-jet cutting application that is pump concerning life time of high pressure components, high reliability and easy of maintenance and service. A new generation of high-pressure pumps for water-jet cutting applications, several influencing cutting parameters have been considered. The requirements for these new high pressure pumps can be summarized as follows: high reliability of all components, easy of maintenance and service low pressure increase after switching of the cutting head

(<10 MPa) and low pressure fluctuation (<10 MPa) the first requirement was realized by using Standard components and redesign of the high pressure intensifier. The second and third requirements were achieved by adjusting the hydraulic drive to the behavior of the high-pressure system, especially to the intensifier (Herbig et al., 1999).

The design of the intensifier depends primarily on the main parameters: pressure and maximum flow rate under working condition which is shown in Figure 2.3

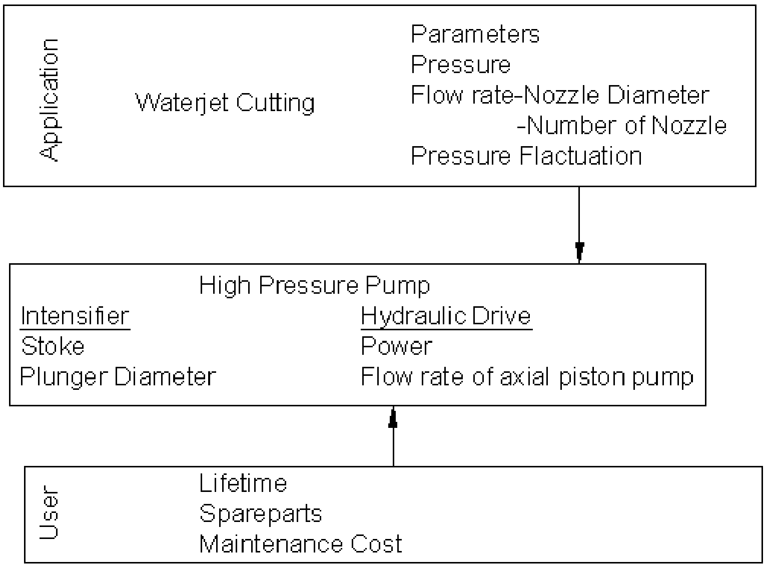

Figure 2.3 Design of intensifier Main Parameters (Herbig et al., 1999)

One of the most important requirements on high-pressure intensifier is low maintenance costs and high easy of service. This is primary realized by effective design of the wear components. During the last several years many design modifications were carried out to increase easy of service, decrease maintenance costs and reduce the number of different wearing parts. The design goals for identical suction and pressure valves are: quick reaction time reduced medium velocity in the cross section of the check valves and no change of the diameter of the parts. The function of the pressure valves is the opposite of the function of the suction valve but it is possible to assemble the same parts in reverse order. in this case one can use absolutely the same parts for different valves with different function. However, it is now even more necessary to pay attention to assembling the parts in the right order. This is, however, far better than the earlier design, where the user was burdened with using the right parts in each valve. This is one of the biggest advantages for users who are not specialist in high-pressure pumps or high-pressure technology. The set-up of the hydraulic system supports high reliability and easy maintenance. The design of the steel frame guarantees good access to all components and the steel frame itself serves as fixing for all components. Due to the design of the frame and of the flange connections of the intensifier, performing required maintenance work is very easy. Wearing parts, like high-pressure seals and check valves, can be replaced in very short time for further reduction in service downtime it is possible to install a second intensifier as a stand-by unit. The switchover to the other intensifier can be done manually or automatically (Herbig et al., 1999).

One of the important tools of the redesign is maintainability of the water-jet cutting systems. New design should be made the equipment much more "maintenance friendly ". Maintenance was an important factor in the operational cost of an intensifier. The life of high-pressure seals should be limited to 40-200 hours. By using a sliding cylinder as a shown in Figure 2.4, the piston should be retracted completely the cylinder body, giving free access to the seals. With this design, replacement time is drastically reduced to approximately 10 minutes compared with previous models in which the complete unit has to be disassembled with took 1 to 2 hours (Hoogstrate and van Luttervelt, 1997).

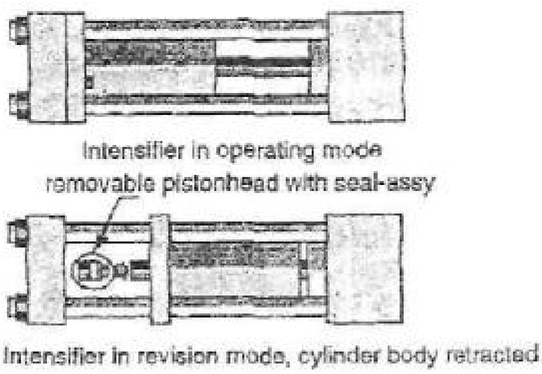

Figure 2.4 Maintenance of Intensifier Removable Piston Head Seals (Hoogstrate and van Luttervelt, 1997)

2.5 Ultra-High Pressure Applications

Higher pressures, up to 1030 MPa, can easily be generated in the industry. Several manufacturers offer commercial units for operation at pressures over 690 Mpa. However, these units are not intended for continuous operation, but for static applications.

Raghaven and Ting (1991) presented results on thin sheet aluminum, steel and titanium cutting with water-jets up to 690 MPa and showed that pressures over 300 MPa were needed to cut these materials. To increase the capability of water-jets, many techniques have been tested including polymer additives pulsating jets.

Hashish, Steele and Bothell (1996) addressed water-jet cutting at pressures up to 690 Mpa. Commercially available systems were capable of working at maximum pressure of 379 MPa it was observed that thin sheet metal (1.6 mm thick) can effectively be cut with water-jet. Higher quality surface are produced as the pressure increases. It was also observed that excessive plastic deformation occurs near the edges. Several composites were cut at 690 MPa without delaminations observed at 379 MPa. Increasing the stand of distance increases the cutting speed. This is attributed to the droplet impact effect that becomes dominant at large stand of distance. The use of 0,025 mm-diameter jets was explored. Tests were also conducted to include cutting with up to 690 MPa abrasive-water-jets (AWJ). Cuts with 690 MPa AWJs confirmed the linear trend of the effect of pressure on cutting rate. Most importantly, the abrasive consumption was significantly reduced when 690 MPa jets were used. This study is concluded that super-pressure plain water-jets were demonstrated to cut a wide range of metallic thin sheets (1.6 mm) at rates up to 4.23 m/s. it was observed that the cutting speed increases as the stand of distance increases. However, cuts displayed rounded edges at the top of surface of the material. It was demonstrated that composite cutting at 690 MPa produced no delamination (as is observed at 414 MPa) and a relatively high cutting speed could be maintained. it was observed that cutting efficiency, expressed as material unit power or per unit water volume, improved as the pressure was increased and diameter of the water-jet was reduced. Cuts made with 0,025 mm-diameter water-jets at 690 MPa were demonstrated showing thin kerfs 0,076-0,127 mm wide. Super water enhanced the coherency of elevated pressure. Cutting with super pressure AWJs showed great promise for minimizing abrasive usage and reducing the kerfs width.

2.6 Stress Analysis in Multi-Layer Cylinders

A method for finding stresses in thick-walled cylinders which is more direct than the usual technique was produced by French and Widden (1994). The stresses in thick-walled cylinders are not in themselves of sufficient direct practical importance to justify a place, but a good case can be for their inclusion as a simple example exhibiting many of the characteristics of more important problems, particularly superposition and the effects of pre-stressing and plasticity.

Equations for the radial σ_r and tangential σ_t stresses at radius r in a thick-walled cylinder;

$$s_r = A - B / r^2 \quad (2.1)$$

$$s_t = A + B / r^r \quad (2.2)$$

Where, A and B are material constants.

Then, provided σ_r and σ_t have opposite signs, the maximum shear stress τ at radius r is given by

$$t = (s_r - s_t) / 2 = B / r^2 \quad (2.3)$$

For a cylindrical annulus between radii r1 and r_2, where r_2>r1, the increase in the radial stress σ_r across the annulus is where t_1 is the maximum shear stress at the inner radius n.

$$s_{r2} - s_{r1} = (A - B / r^2) - (A + B / r^2) = t_1 (1 - r_1^2 / r_2^2) \quad (2.4)$$

Denoting this decrease in pressure by Δp, it is obtained

$$\Delta p = t_1(1 - r_1^2 / r_2^2) = t_1(1 - 1/q_{21}^2) \tag{2.5}$$

This simple equation gives the drop in radial pressure Δp over the layer between r_1 and r_2 in terms of the maximum shear stress t_1 at the inside and the ratio $q_{21} = r_2/r_1$ of the outside to the inside radius.

The equations given very much simplify calculations of the stresses and deflections in thick-walled cylinders of known sizes under given pressure and of the residual stresses remaining after a cylinder has been yielded by heavy pressure. They also greatly facilitate the design of cylinders to withstand given pressures (French and Widden, 1994).

2.6.1 Multi-Layer (Shrink-Fit) Studies

Paliwal, D. N. et al. has developed a computer aided design method in Turbo Pascal for multilayer cylindrical high-pressure vessels. The design based on Tresca and Von Misses' conditions. An algorithm is given for different procedures in software. His program calculates the thickness of the various layers as well as the residual and net stresses across the thickness for given shrink-fit allowances. For a given internal pressure and number of layers (2 and 3) and shrink-fit, the program yields the thicknesses of various layers as well as the residual stresses and net stresses in various layers. Optimization in terms of machining cost of various layers can also be carried out using this program. Programs are also included for the design at flat heads and reinforcement around openings and for graphical representation of stresses across the thickness (Paliwal, 1993).

He derived an equation according to the Lame' and Manning and Labrow's (Manning, 1971) equations and had a relation of;

$$(P_i - P_o) = \left(\frac{M}{2}\right)^{1/2}\left[\frac{K_1^{\,2}-1}{\left(4K_1^{\,4}+2K_n+1\right)^{1/2}}+...+\frac{K_n^{\,2}-1}{\left(4K_n^{\,4}+2K_n+1\right)^{1/2}}\right] \qquad (2.6)$$

And this equation was solved assuming,

(a) radius ratios to be same or

(b) pressure difference at each layer to be the same and as a result

$$(P_i - P_0) = \frac{n(M/2)^{1/2}(K^2-1)}{\left(4K^4+2K+1\right)^{1/2}} \qquad (2.7)$$

Where, K is ratio of outer radius to inner radius, M is Maxwell's factor and n indicates the number of layers.

Chen, C. T. P. used a new theoretical model for reverse yielding in high strength steel to obtain residual stresses in a closed-end thick-walled cylinder subjected to high internal pressure. The numerical results of residual hoop stresses are also obtained by using the isotropic and kinematical hardening models. A comparison of different models in the theoretical prediction of residual stress is given. The new results indicate that the influence of the Bauschinger and hardening effects on the residual stresses are more significant for cases involving larger wall ratio and higher internal pressure (Chen, 1986).

Barlow assumed that the circumferential stress is uniformly distributed across the wall section and that the internal pressure acts over the outside diameter. The basic assumptions are incorrect, but the stresses computed are always on the safe side. And he suggested a formula for wall thicknesses that

$$t = \frac{PD_2}{2sE} \qquad (2.8)$$

Where t indicates wall thickness, σr is stress distributed, E is the modulus of elasticity of pressure vessel material and D_2 shows outside diameter and P is internal pressure (Siemon, 1958).

Clavarion had a formula for stress distributed as

$$s = P\frac{R_1^{\,2}(1-2m)+R_2^{\,2}(1+m)}{R_2^{\,2}-R_1^{\,2}} \tag{2.9}$$

For μ =1/3 (steel) this formula can be written

$$R_2 = R_1\sqrt{\frac{3s+P}{3s-4P}} \tag{2.10}$$

Where R_1 and R_2 are inner and outer radii respectively, P is internal pressure and μ shows Poisson's Ratio and it is 0.333 for steel, wrought iron and brass, and 0.250 for cast iron.

Clavarion's formula is based on the ‶Maximum Strain Theory" of elastic failure that claims that failure is caused when a certain maximum strain, produced by the combined stresses, is reached. It assumes longitudinal tension as well as circumferential compression to be acting and applies to cylinders with closed ends only. For high temperatures and other cases where the value of Poisson's Ratio differs from 0.333, the formula should be revised by inserting the correct value (Siemon, 1958)

Von Bach suggested a formula for calculation of tangential unit stress as

$$s_t = \frac{m-2}{m}\frac{R_1^{\,2}}{R_2^{\,2}-R_1^{\,2}}P+\frac{m+1}{m}\frac{R_2^{\,2}+R_1^{\,2}}{R_2^{\,2}-R_1^{\,2}}\frac{1}{x^2}P \tag{2.11}$$

Where s_t is the tangential unit stress at any point distant *x* from the center, and *m* is reciprocal of Poisson's Ratio and defined as 1/μ. P shows the internal pressure (Siemon, 1958).

s_t become a maximum for x=R₁ and m=10/3, the formulation becomes,

$$R_2 = R_1\sqrt{\frac{mS+(m-2)P}{mS-(m+1)P}} \tag{2.12}$$

$$R_2 = R_1\sqrt{\frac{S+0.4P}{S-1.3P}} \tag{2.13}$$

Birnie has developed a formula based on the same assumptions as Clavarion's formula except that the longitudinal stress is equal to zero. It applies only to cylinders with open ends and is used for the calculation of guns and it was defined as,

$$s = P\frac{D_1^{\,2}(1-m)+D_2^{\,2}(1+m)}{D_2^{\,2}-D_1^{\,2}} \tag{2.14}$$

Where, σ is stress distributed, *P* is internal pressure D_1 and D_2 indicates the inner and outer diameters respectively and μ is the Poisson's Ratio (Siemon, 1958).

2.6.1.1 Stress, Strain, and Interface Pressure in Multi-Layer Structures

The shrink fit fastening process is widely used in industry to produce tight, precision assemblies where other fastening methods are neither necessary nor practical. It can also be used in research to produce a predictable residual stress state.

Stress analysis of thick-walled cylinders under internal and/or external pressure, of course, received considerable attention. Conway and Farnham (1967) have accounted for non-uniformity of interface pressure and friction in shrink fits of rings on elastic shafts. Elastic-plastic cylinder behavior, similarly, has been extensively studied. Johnson and Mellor (1973) have included in their book an extensive bibliography of work in this area. The work of Bland (1956) forms the basis for recent research. Odenö (1969) has studied the behavior of elastic-plastic disks under thermal gradients. Later Chen considered the problem of analysis of an elastic-plastic annular plate, comparing the predictions of both flow and deformation plasticity theories. Gamer and Lance (1983) have studied in detail the stresses in a shrink fit in which the disk as well as the ring may become plastic.

The stresses and deformations in a thin ring-disk shrink fit assembly have been analyzed by Gamer and Lance (Fig. 2.5). The interface pressure for the presumed plane stress state has been found as a function of the interference of the fit.

$$p = \frac{1}{2}\frac{s_0}{1+h\dfrac{s_0}{E}}\left[h\left(\frac{1}{a}-\frac{s_0}{E}\right)+In\frac{EI}{s_0 a}\right]+\frac{s_0}{2}\left(1-\frac{EI}{s_0 a}\frac{a^2}{b^2}\right) \tag{2.15}$$

Where p is interface pressure, σ_0 is the initial tensile yield stress $\sigma_0 = \sigma_t - \sigma_r$ and, η is work hardening parameter, E is Young's modulus, I is interference, a and b are inner and outer radii of disk respectively.

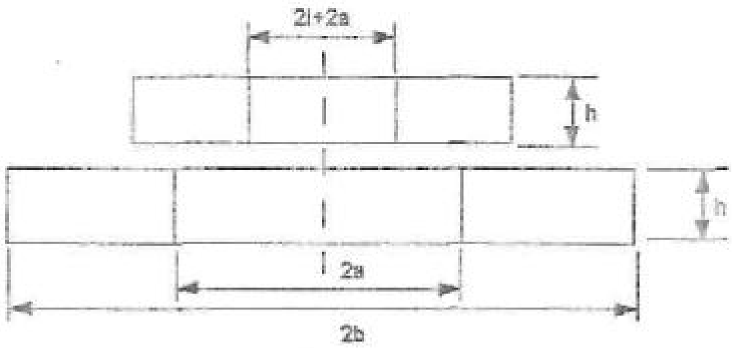

Figure 2.5 Shrink Fit Geometry (Gamer and Lance, 1983)

Interferences large enough to induce plastic deformations in the ring are accounted for. The ring material was assumed to be a linear work-hardening material that obeys Tresca's yield condition.

Since thermally assembled elastic-plastic shrink fits are found frequently in mechanical engineering, the determination of the stress distribution has been the subject of several investigations. Due to the complexity of the problem, numerical techniques, and finite element methods are used.

Thermal assembly of an elastic-perfectly plastic shrink-fit consisting of a solid inclusion and a hub whose axial length is small compared to the interface radius was investigated by Mack and Bengeri (1994) which is seen in Figure 2.6. Hence, as usual in the analytical treatment of shrink fits, a solid disk and a ring in a state of plane stress were used as a model, it is presumed that both components exhibit the same material properties and retain their circular symmetry through the assembly. All the results were based on Tresca's yield condition and the flow rule associated with it.

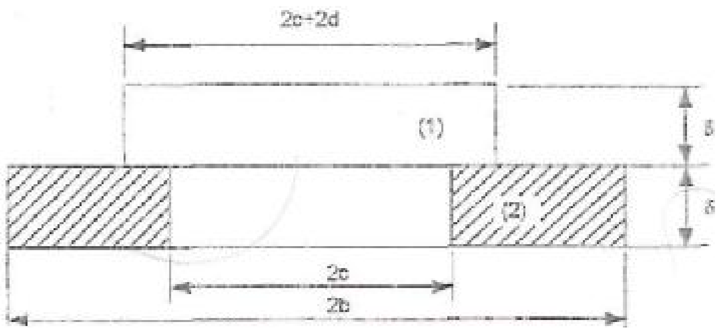

Figure 2.6 Sketch of the shrink fit prior to the assembly
1) Inclusion 2) Hub (Mack and Bengeri, 1994)

In. the study, the radial, tangential stresses, and universal displacement were given below equations.

$$s_r = C - \frac{K}{r^2} + E\left(-\frac{a}{r^r}\int Trdr + \int \frac{e_r^{pI}}{r} dr \right) \tag{2.16}$$

$$s_t = C + \frac{K}{r^2} + E\left(\frac{a}{r^r}\int Trdr - aT + \int \frac{e_r^{pI}}{r} dr + e_r^{pI} \right) \tag{2.17}$$

$$u = \frac{1}{E}\left[(1-n) s_r r + \frac{2K}{r} \right] + 2\frac{a}{r}\int Trdr \tag{2.18}$$

Where; C and K are constants of integration under boundary conditions, r is radius, E is Young's modulus, T is temperature, a is coefficient of (linear) thermal expansion, v is Poisson's ratio, and ε^{pI} is plastic strain component and calculated as; (Mack and Bengeri, 1994)

$$e_r^{pI} = \frac{1}{E}\left(s_t - s_r - \frac{2K}{r} \right) - 2\frac{a}{r^2}\int Trdr + aT \qquad (2.19)$$

Of particular interest is that the unloading process, which is due to the temperature dependence of the yield stress, has only a minor influence on the stress distribution. Hence, for engineering purposes a sufficient approximation of the residual stresses and, especially, of the interface pressure can be calculated isothermally, and unloading need not be taken into account. It should be mentioned that the same applies to several types of shrink fit with hollow inner components.

The determination of the stress distribution in internally assembled elastic-plastic shrink fits constitutes an important problem, which has received considerable attention in applied mechanics. An advantage of the numerical techniques is the possibility to use complex models of the material behavior and the geometrical properties. On the other hand, analytical or semi-analytical methods are more appropriate to study the essential physical features of the problem.

The shrink fit with hollow inner component is modeled by two elastic-perfectly plastic rings (Fig. 2.7), which are in a state of plane stress and retained their circular symmetry throughout the assembly by Bengeri and Mack (1994). It was assumed that both parts exhibit the same material properties and come into contact which each other immediately. Furthermore, an interference sufficient small for one of the rings to remain elastic was presumed.

Although, in principle, all the material properties depend on the temperature, the dependence of the yield stress was the most pronounced one within the temperature range under consideration. Therefore, the other material properties were taken to be constants (Bengeri and Mack, 1994). Figure 2.7 shows sketch of the shrink-fit prior to the assembly.

Obviously, the mounting process was influenced not only by the temperature dependence of the yield stress, but also by several other parameters. Although certain combinations of their values will cause a qualitatively different behavior of me components, the results of the study hold for a wide class of shrink fits.

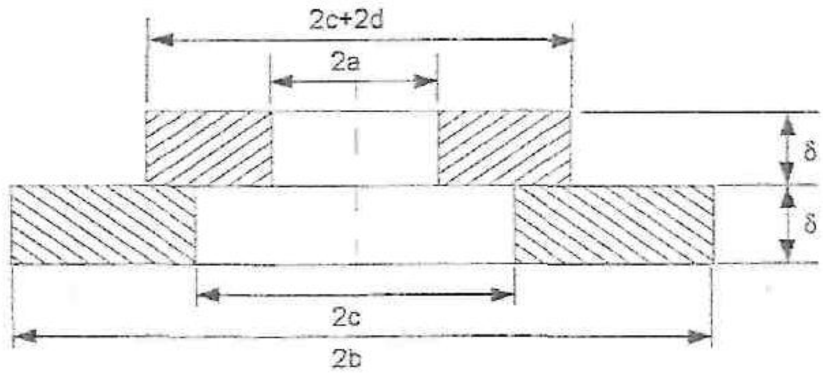

Figure 2.7 Sketch of the shrink fit prior to the assembly (Bengeri and Mack, 1994)

For a linearly temperature dependent yield stress for $\sigma_t > 0 > \sigma_r$ Tresca's yield condition takes the form

$$s_t - s_r = s_0(1 - bT) \tag{2.20}$$

Where; σ_0 denotes the uni-axial yield limit at reference temperature and β a material parameter. And then, the radial and tangential stresses become

$$s_r = s_0\left(\ln\frac{r}{c} - b\int\frac{T}{r}dr\right) + C \tag{2.21}$$

$$s_t = s_0\left[1 + \ln\frac{r}{c} - b\left(T + \int\frac{T}{r}dr\right)\right] + C \tag{2.22}$$

Where; r is radius, C is the constant of integration under boundary conditions.

For $0 > \sigma_r > \sigma_t$ the radial and tangential stresses become;

$$s_t = -s_0(1 - bT) \tag{2.23}$$

$$s_r = -s_0 \frac{1}{r}\int (1 - bT)dr + \frac{R}{r} \tag{2.24}$$

Where; R denotes a constant of integration under boundary conditions.

The main finding is that the reduction of the yield stress at elevated temperatures has only a small influence on the residual stresses and; in particular, on the interface pressure. Hence, for engineering purposes a sufficient approximation of the latter can be calculated isothermally and unloading need not to be taken into account. This seems quite remarkable, since it was shown that the temperature dependence of the yield stress has a pronounced effect on the stress distribution in a shrink fit subject to a temperature cycle (Bengeri and Mack, 1994).

3. MATERIAL AND METHOD

This chapter provides methods to develop high pressure connection parts using redesign methodology, developed by Otto and Wood, provides systematic approach for the product development that matches customer needs. However, the procedure has been modified by Geren et al (2007) to meet the needs in the new product development. In addition to that redesign activities are supported by suitable concept generation techniques such as TRIZ. In this study the customer needs have not been considered because the product is a prototype development. Therefore, design concepts must be generated by considering the prototype development. Figure 3.1 shows the redesign methodology used in this study.

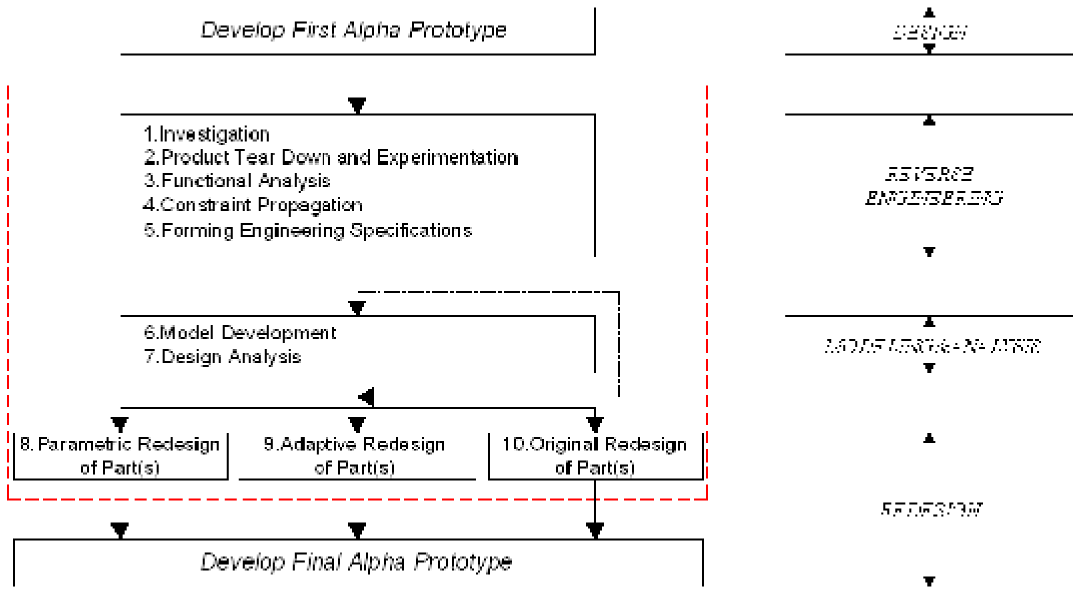

Figure 3.1 Modified Reverse Engineering and Redesign Procedure (Geren et al,2007)

Following parts of the WJM system will be redesigned based on the previous prototype which is available in the Mechanical Engineering Department of Çukurova University.

- Cup-Head
- Sealing head
- Check valves and check valve housing blocks

- High pressure pipes
- Connectors between nozzle and check valve block
- Connectors between intensifier and check valve block
- Other connection parts of the WJM system.

The parts provided above will be introduced in the following sections. Figure 3.2 shows the exploded view of whole system and Figure 3.3 shows the assembled view of whole system.

3.1 Reverse Engineering and Redesign Methodology

Design is to formulate a plan for the satisfaction of human needs**.** Redesign is product development that matches customer needs. The definition of design and redesign varies and little agreement exists on how to use and apply a structured problem solving methodology for design and redesign problems (Ingle 1994, Otto and Wood 1999). The reverse engineering and redesign methodology focuses on the process steps needed to understand and represent a current product. Extensions of contemporary techniques in engineering design are utilized at a number of stages in the redesign process to meet the goal. Redesign problems include the all process steps of original design together. Three distinct phases embody the methodology; reverse engineering, modeling and analysis, and redesign.

The original redesign procedure has to be modified to adopt into this study. Figure 3.1 illustrates the modified procedure suggested by Geren at al (2007) will be used in this study. It starts with the ‘design’ step, which is named ‘the development of a first alpha-prototype WJM system’. Then, reverse engineering and redesign methodology is applied. This methodology is actually allowed evaluating parts to obtain usable product redesign specifications for the design of the final alpha prototype.

The methodology is expected to provide better designs for the target components of WJM system. It may lead to parametric, adaptive or original design changes on the high pressure pipes, connections and check valve of the WJM, developed as previous prototype in the Mechanical Engineering Department of

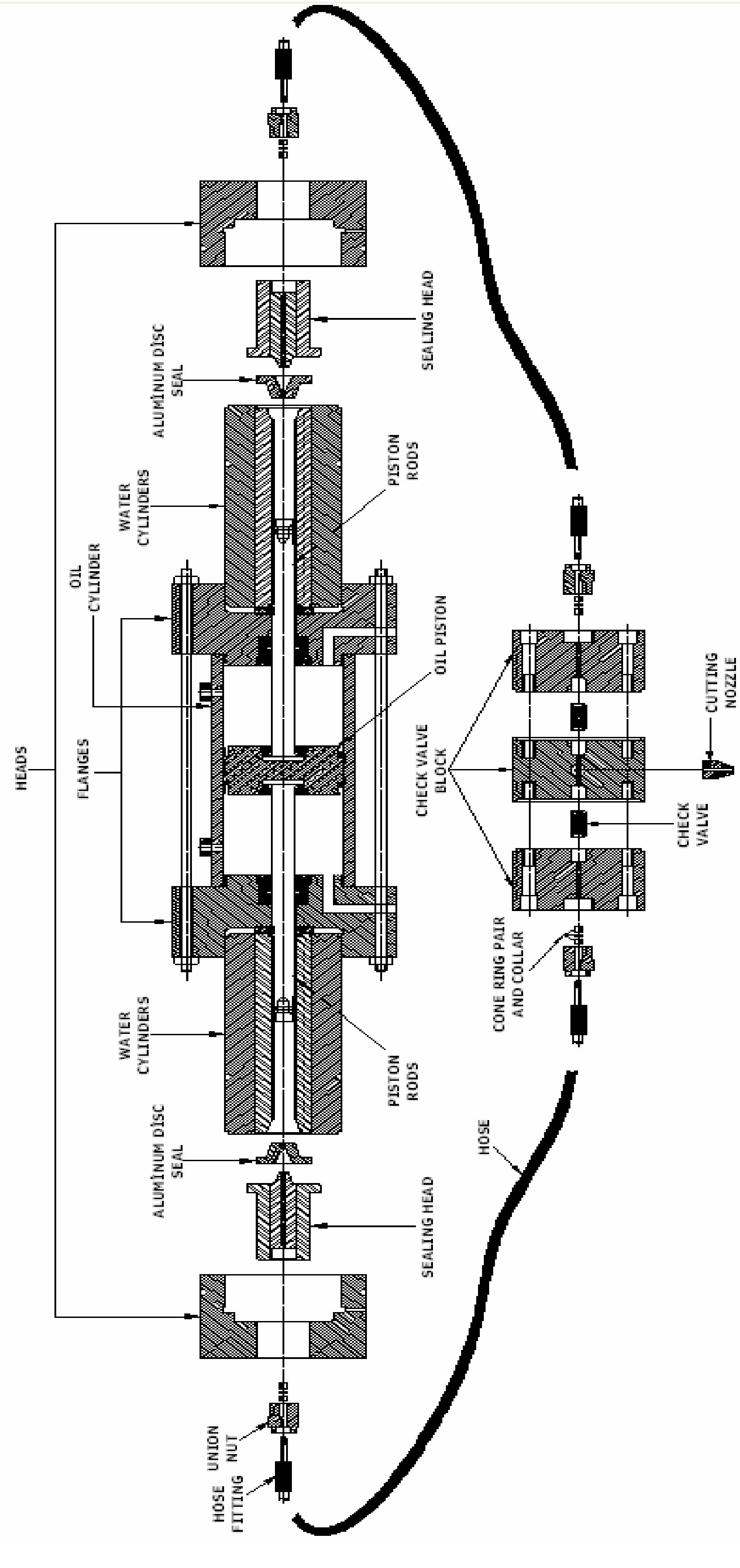

Figure 3.2 3000 bar operating pressure intensifier with exploded view of high pressure parts.

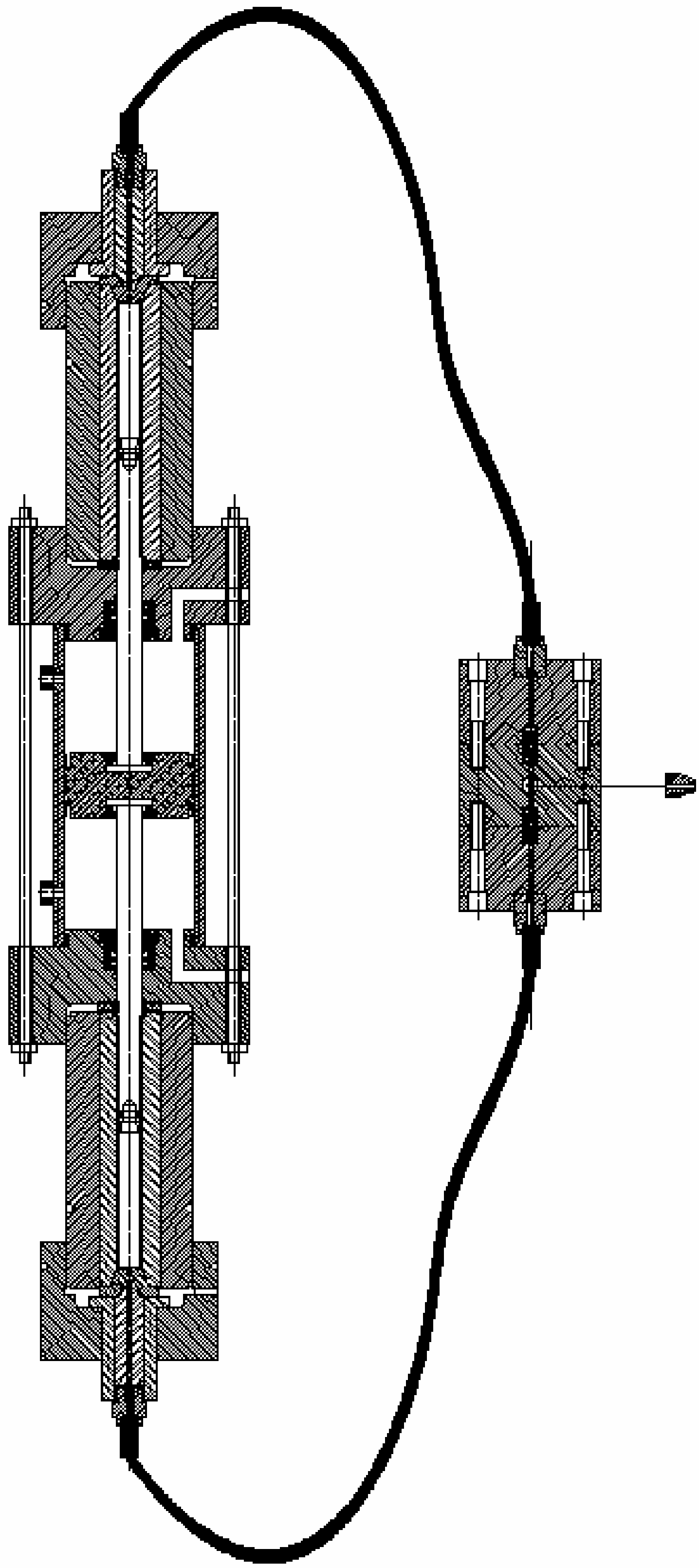

Figure 3.3 The assembled view of 3000 bar operating pressure intensifier with high pressure parts.

Çukurova University. The main intention is to simplify the design of the target components for ease of assembly and disassembly, reduce the manufacturing and material cost.

3.1.1 Reverse Engineering

The intent of the first phase, reverse engineering, is twofold. First, a product is treated as a black box, experienced over its operating parameters, and studied with respect to customer needs and predicted and/or hypothesized functionality, product components, and physical principles. The second step of the reverse engineering phase is to experience the actual product in both function and form. This sub-phase includes the full disassembly of the product, design for manufacturing analysis, further functional analysis, and the generation of final design specifications. The intent of these process steps is to fully understand and represent the current instantiation of a product. Based on the resulting representation and understanding, a product may be evolved, either at the subsystem, configuration, component or parametric level.

3.1.1.1 Investigation

The first step of the reverse engineering provides us with the necessary investigation, predictions and hypotheses for a successful product evolution. Project description and what is the expected level of completion to be accomplished. The redesign process begins with a set of customer needs and target specifications.

The black box model is the starting point for the first step of the redesign methodology. Based on the problem statement and black-box model, prototype needs, expectation and requirements are identified, gathered and organized for the product.

The mission statement for the product, prototype needs and requirement list which must be relative statement for the product, prototype needs and requirement list which must be relative importance and preliminary product specifications are important inputs to the process is prepared and abstracted considering the weakness of the previous prototype. Table 3.1 shows the general requirements of the product.

Table 3.1 General requirement list

Geometry	Size, height, breadth, length, diameter, number, arrangement
Material	Properties of the initial and final product, auxiliary materials
Safety	Direct safety principles
Production	Possible dimensions, preferred production methods, achievable quality
Assembly	Special regulations, installation, connections
Maintenance	Servicing intervals (if any), exchange and repair
Costs	Manufacturing costs (labor cost, cost of material), waste of material

Product needs list and weight/diagnosis weakness is shown Table 3.2. In this table the needs of the high pressure parts are marked from 1 to 10 to identify weakness by considering value analysis. The meaning of 1 is very inadequate solution, the meaning of 10 is ideal solution. Then, current product is marked as same way. Marking is made using value analysis. This table point out the weakness of the previous prototype and superiority of new design for all of the parts which is mentioned above, the table is to be prepared for each part separately.

Part name is defined at first row and second column. Then, at the first column product needs general title, at the second column sub-title is explained based on prototype. Then, at the third column there are the new designs rating and at the fourth column current designs rating is available.

3.1.1.2.Product Tear Down and Experimentation

Product tear down is applied at the second step of reverse engineering. The disassembly is systematically executed, listing the order of disassembly, the component to be removed, access direction and any permanent deformation caused by the disassembly.

Table 3.2 Product needs list and weight/diagnosis weakness

Part Name **Product Need**		**Current Design Weight**	**New Design Weight**
I.Size	a) Light weight for ease of assembly b) Optimum dimension	x	x
II.Safety	a) Able to stand maximum pressure	x	x
III.Reliability	a) High reliability of all parts subjected high pressure	x	x
IV.Sealing	a) Ease of removing plastically deforming soft metals b) Plastically deformed main part	x	x
V.Replacement	a) Ease of replacement high pressure parts b) Lower time for replacement	x	x
VI.Maintenance	a) Ease of assembling and dismantling b) Lower assembling and dismantling time	x	x
VII.Manufacturing	a) Ease of manufacturing high pressure parts for WJM system	x	x
VIII.Cost	a) Manufacturing and labor cost	x	x

Each part is analyzed in three ways:

- Possibility to eliminate the part
- Possibility to reduce manufacturing cost
- Possibility to redesign the part for ease of assembly and disassembly

In Figure 3.4 the part's exploded view which is built up high pressure parts except intensifier of WJM system is shown.

Cup-Head

The head is fixed to the water cylinder by metric threads. Hence, both water cylinder and heads are threaded. While tightening the head towards to water cylinder, aluminum conic disc seal which is providing sealing action between head and water cylinder is compressed by the help of sealing head. Briefly, the head is used to apply clamping force to the sealing head.

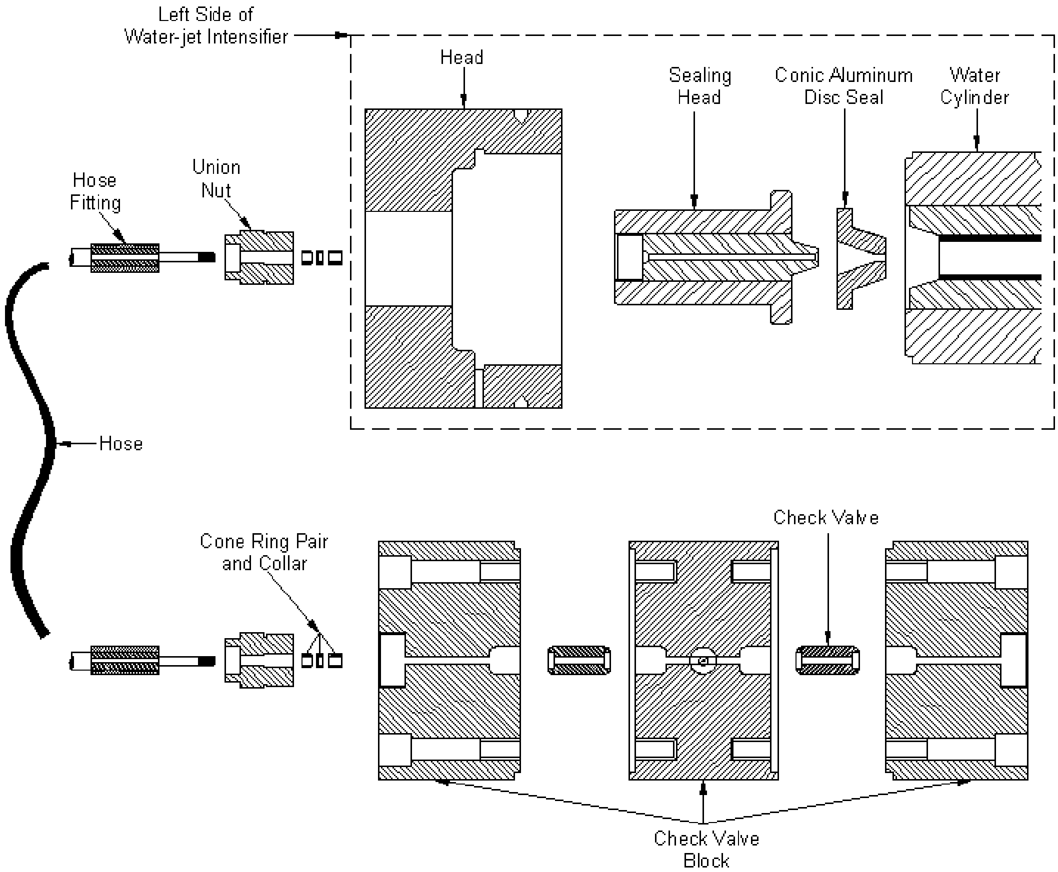

Figure 3.4 Exploded view of high pressure connection parts

In addition to this, piston sealing head which enables sealing between piston head and water cylinder must be replaced frequently. Hence, access to the piston head seals has to be eased. Consequently, head's assembly and disassembly must be easy. But the head is too heavy in the current design. The schematic view of assembling the head, the sealing head and left water cylinder is shown in Figure 3.5.

Sealing Head

Sealing head is located in between the water cylinder and the head. Sealing head had been designed for the system pressure of 3000 bars. The design procedure which was used for multi layer structures had also been used for the design of the sealing head. Sealing head had been made of stainless steel to protect them from the risk of corrosion.

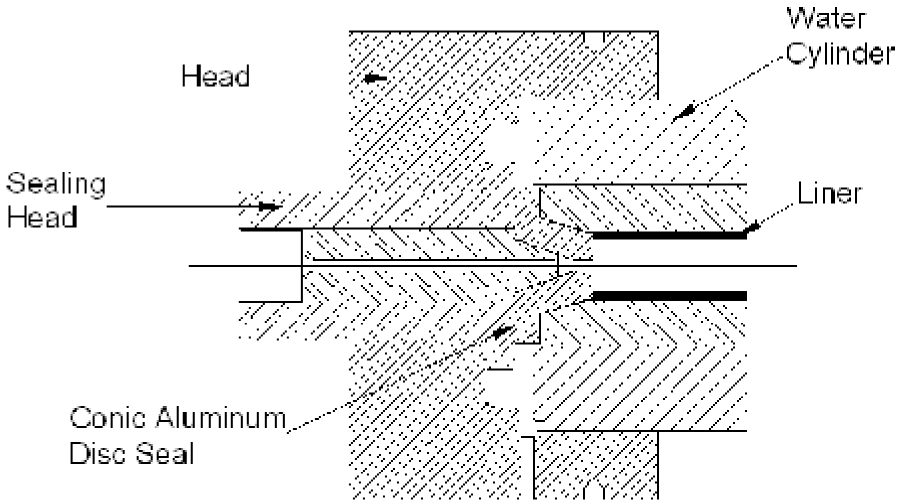

Figure 3.5 Schematic views of intensifier-cup head assembly

Sealing head compress aluminum disk seal when the head is tightened toward the water cylinder for sealing between water cylinder and head. Beside this, the sealing head is provided refilling of low pressure water and discharging of high pressure water from water cylinder to the check valve by the help of union nuts and hoses.

Manufacturing of the sealing head is not easy due to shrink fitted multi layer structure. Addition to this, the sealing head material is stainless steel which is an expensive material. Also, using sealing head means too more extra parts and addition time for assembling and disassembling.

Check Valve and Check Valve Block

The intensifier receives low pressure water and can increase the water pressure up to 3000 bar. The water flow into the intensifier and output from the intensifier is controlled by check valves. Four of the check-valves are located into a cylindrical check-valve block. Briefly, functions of check valves are to control high and low pressure flows.

The current design uses commercially available low pressure hydraulic check valves (with 700 bar burst pressure). In the current design, the low pressure check valves were modified and the existing springs were replaced with a stainless steel one. The check valves were shrink-fitted into their holes, machined in the block which consists of three main separable sub-blocks. This type of design enabled the use of low pressure standard commercial check valves instead of high cost ones designed for ultra high pressure applications. Sealing was provided using plastically deformed soft copper cone rings on both sides of each check valve.

Shrink-fit design is applied to commercial low pressure check valve to increase their operating pressure. Shrink-fit assembly of the check valves is not easy and takes considerable time. This is because after heating a sub-block of check valve without changing metallurgical structure, check valve is placed into machined hole at the block and then sub-block is waited to cool for completing shrink-fit assembly process. Same procedure is applied to other half of the check valve group. Consequently, applying shrink fitting process is not easy and considerable time is spent. Beyond this, when check valves wear, they must be replaced with the new one. After shrink fit process, replacement of check valves is very difficult. This is mainly due to the difficulty of separating check valves and sub-blocks from each other. The schematic view the check valve and the check valve block is shown in Figure 3.6.

High Pressure Hoses

High pressure hoses are fixed to the sealing head and check valve block, check valve block and cutting nozzle to transfer high pressure water from intensifier.

High pressure hoses purchased from ‘‘STIR STAR-Germany’’ endure 3200 bar operating pressure. These hose’s outer cover is polyamide, inner cover is polyoxymethylene. Addition to these, 8 layers of high tensile steel wire exist between outer cover and inner cover. These specifications are made the hose more expensive. Besides, this hoses come from abroad and cost increase relatively.

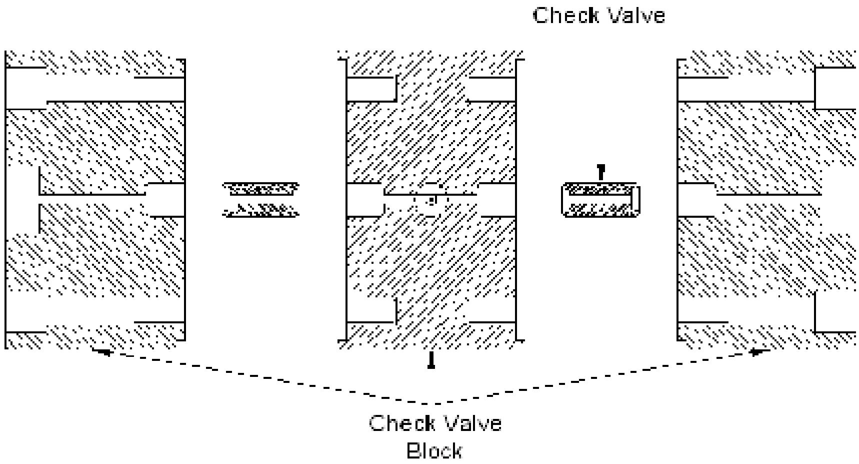

Figure 3.6 Schematic view of check valve and check valve block

3.1.1.3 Functional Analysis

By considering the results obtained from steps 1 and 2 of the redesign methodology which provided detailed information regarding component function, assemblability, physical parameters, manufacturing processes so on. The needs of high pressure connections parts is abstracted then identified and ranked for product improvement. Functional analysis is the key instrument for building the expectation. Briefly, using the results obtained from steps 1 and 2, required features and functions are defined which is added to the high pressure parts.

Most design challenges are too complex to solve as a single problem and can be divided into sub-problems to create a more specific description of what the elements of the product do in order to implement the overall function of the product. The goal is to describe the functional elements of the product without implying a specific technological working principle for the product concept.

3.1.1.4 Constraint Propagation

With the data obtained from previous steps, improvements are obtained for all components. Functional descriptions of components prescribed by the product needs were updated by adding, removing or altering functions. Based on the new function structure, adaptive solutions are obtained in a number of ways. An example morphological matrix of the solution principles is presented in Table 3.3 only for the significant functions.

Each row of the matrix corresponds to a sub-function from step 2. The ‘current parts’ column represents solutions for the previous prototype and the ‘possible solutions’ column provides solutions for the new prototype for each significant function. This table allows one to plan for design changes without violating functional requirements.

Briefly, in this step, alternative solutions are produced for desired specifications and functions from current prototype.

Table 3.3 Example of Morphological matrix

Function	**Part Name**	**Current Solution**	**Possible Solution**
-Allow easy access to piston seal head -Apply clamping force to the sealing head	**Head**	Threaded cylinder end cups	x
-Allow refilling of low pressure water and discharging of high pressure water from water cylinder to the check valves -Clamp and deform aluminum disc seal	**Sealing Head**	Shrink-fitted double layer cylindrical structure	x
-Control the direction of high and low pressure flows	**Check Valve**	Modified commercial low pressure cylindrical check valves fitted as shrink-fit	x
-Connect high and low pressure hoses	**Check Valve Block**	Cylindrical 3 part separable block	x
-Flow high and low pressure water	**Pipes**	Multi layer hoses for high pressure	x

3.1.1.5 Forming Engineering Specifications

In this stage the input and the output flows are conceived for each sub-function. To reach the target, for each sub-function suitable design specifications are identified which is indicated in the first step. The specifications which are obtained in this step provide clear goals for evolving the high pressure connection parts.

3.1.2 Modeling and Analysis

Virtual and physical modeling of the product provides in-depth insights into its operation and possible improvements that may be achieved parametrically. In this step, analysis strategies are developed for solving the models.

Engineering specifications, requirements obtained from previous steps were used to perform adaptive or original design changes before creating and optimizing design for some of the high pressure parts. These parts are head, pipes, check valve, check valve block and so on.

3.1.2.1 Design of Head

One of the important problems is the replacement of piston head seals. There must be easy access to the piston head seals because of having limited life and requiring frequent replacement. Consequently, for easy replacement and servicing requirement the head design gains important.

The head is fixed to the water cylinders by metric threads. Hence, both water cylinders and heads are threaded. The design of the head is based on threaded joints. Consequently, design of head has been carried out based on threaded stresses. Therefore, obtaining the thread dimensions tension, shear and bearing loading conditions must be considered. Figure 3.7 shows the current cup-head.

For tension, force "F" seen at Figure 3.8 applied on the head is,

$$F = P.A \tag{3.1}$$

Where, P is system pressure, A is inner cylinder area. Tensile stress which is exerted on the head is,

$$s = \frac{F}{A_t} \tag{3.2}$$

Where, A_t is tensile stress area of threaded heads and tension force of "F" is applied on the heads.

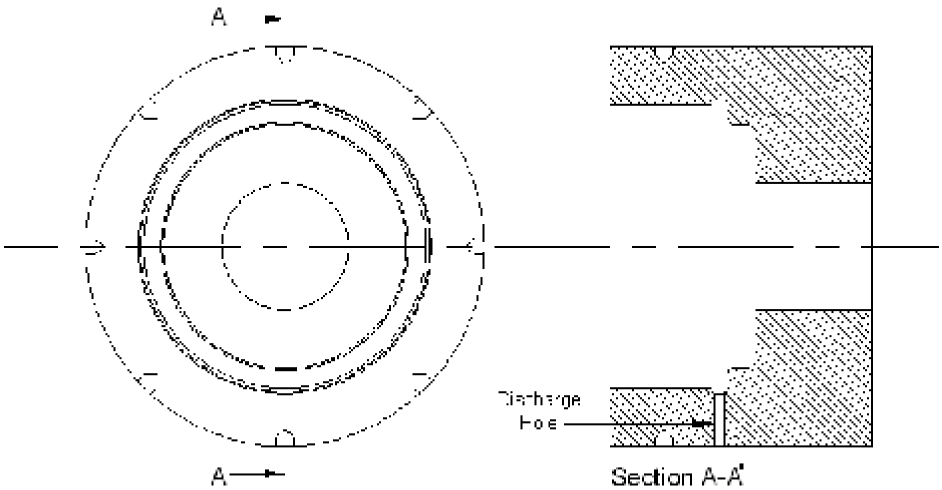

Figure 3.7 The section of current cup-head.

The heads are also subjected to fluctuating stresses because of the repeated low and high pressure then maximum and minimum stresses can be found as,

$$s_{\max} = \frac{F}{A_t}, s_{\min} = 0 \tag{3.3}$$

Mean and amplitude stress can be found as,

$$s_{mean} = s_{amp} = \frac{F}{2A_t} \tag{3.4}$$

The factor of safety is determined which is work under dynamic loading as,

$$n = \frac{1}{\frac{s_{mean}}{S_{ut}} + \frac{s_{amp}}{S_e}} \tag{3.5}$$

Where S_e is endurance limit and given as a function of some modifiers,

$$S_e = k_a.k_b.k_c.k_d.k_e.k_f.S_e^{'} \tag{3.6}$$

Where,

k_a : Surface factor which is a function of surface finish.

k_b : Size factor which is a function of diameter.

k_c : Reliability factor which is a function of reliability, safety and service life.

k_d : Temperature factor is a function of operating temperature.

k_e : Stress concentration factor which is a function of discontinuities.

k_f : Miscellaneous effects factor which is a function of residual stresses, corrosion, plating, fretting etc.

$S_e^{'}$: Endurance limit of a rotating beam which is based on material.

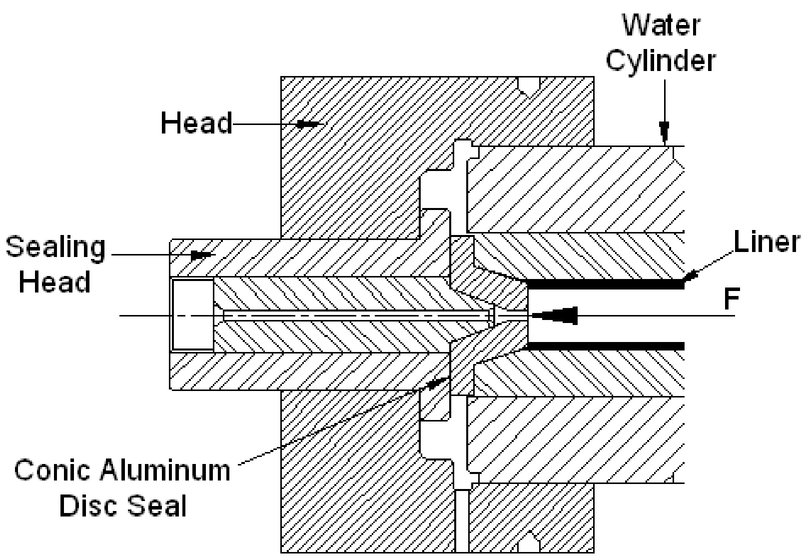

Figure 3.8 Applied force on head.

All factors are given by figures in Shigley (1986). For axial loading $S_e^{'}$ can be calculated by this formula;

$$S_e^{'} = \left[0{,}566 - 9{,}68.10^{-5}.S_{uc}\right]S_{uc} \qquad S_{uc} \geq 400Mpa \tag{3.7}$$

Since, $S_{uc} \cong S_{ut}$ for most steels. It is also important to remember that k_b is taken as unity in Eq. (3.6) when Eq. (3.7) is used because these effects have been accounted for (Shigley, 1986).

In addition to above shear stress on the threads can be found as,

$$t = \frac{2F}{p.d_r.h} \tag{3.8}$$

Where, F is system force, d_r is minor diameter of the thread and h is height of the threads. As the same way, shear stresses on the threads act on the heads under fluctuating stresses. So that, maximum and minimum shear stresses can be found as,

$$t_{\max} = \frac{2F}{p.d_r.h}, t_{\min} = 0 \tag{3.9}$$

Since amplitude and mean shear stresses can be written as,

$$t_{mean} = \frac{t_{\max} + t_{\min}}{2}, t_{amp} = \frac{t_{\max} - t_{\min}}{2} \tag{3.10}$$

Endurance modifying factors can be taken as Eq. (3.6). Using the maximum distortion energy theory endurance limit for shear can be found as,

$$S_{se} = 0{,}577.S_e \tag{3.11}$$

Since the loading is simple. Shear loading factor of safety for shear can be found as,

$$n = \frac{S_{se}}{t_a} \tag{3.12}$$

The design of the heads must also be checked for bearing thread stresses too, bearing stress s_b should not exceed 15 MPa for design of the bearings stress of the threads and hence for this case,

$$s_b = \frac{4F}{p.\left(d^2 - d_i^2\right).z} \leq 15MPa \tag{3.13}$$

F is system force at design pressure, d and d_i is major and minor diameter of the threads respectively, " z " is number of engaged threads.

3.1.2.2 Design of High Pressure Pipe

To decrease production cost high pressure pipe is designed instead of high pressure hoses. High pressure connection pipes that are used in water jet cutting systems operate under fluctuating stresses. This is because as the one side of high pressure pipes are subjected to high pressure during the compression stage; other side is subjected to low pressure during refilling stage. Continuous operation results in repeated sinusoidal fluctuating stresses. Therefore, the high pressure pipes are subjected to repeated low and high pressures which resulting fluctuating stresses on the walls.

Referring Figure 3.9 the designation of inside radius of pipe is ‘‘ a ’’, outer radius of ‘‘ b ’, internal pressure ‘‘ p_i ’’ and external pressure ‘‘ p_o ’’. Then it can be shown that tangential and radial stress exist on the pipe are,

$$s_t = \frac{p_i a^2 - p_o b^2 - a^2 b^2 (p_o - p_i)/r^2}{b^2 - a^2} \tag{3.14}$$

$$s_r = \frac{p_i a^2 - p_o b^2 + a^2 b^2 (p_o - p_i)/r^2}{b^2 - a^2} \tag{3.15}$$

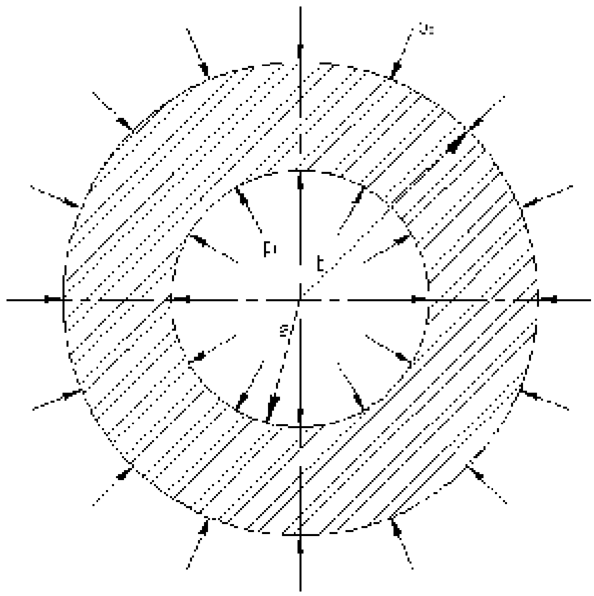

Figure 3.9 Designation of pipe dimension, external and internal pressures

When external pressure is equal to zero then the equations will be,

$$s_t = \frac{a^2 p_i}{b^2 - a^2}\left(1 + \frac{b^2}{r^2}\right) \tag{3.16}$$

$$s_r = \frac{a^2 p_i}{b^2 - a^2}\left(1 - \frac{b^2}{r^2}\right) \tag{3.17}$$

These equations are plotted in Figure 3.10 to show stresses over the wall thickness. The maximum stresses occur at the inner surface, where $r = a$.
Their magnitudes are,

$$s_t = p_i \frac{b^2 + a^2}{b^2 - a^2} \tag{3.18}$$

$$s_r = -p_i \qquad (3.19)$$

Addition to these formulas, longitudinal stress exists when pressure is exerted on the pipes wall.

$$s_l = \frac{p_i a^2}{b^2 - a^2} \qquad 3.20)$$

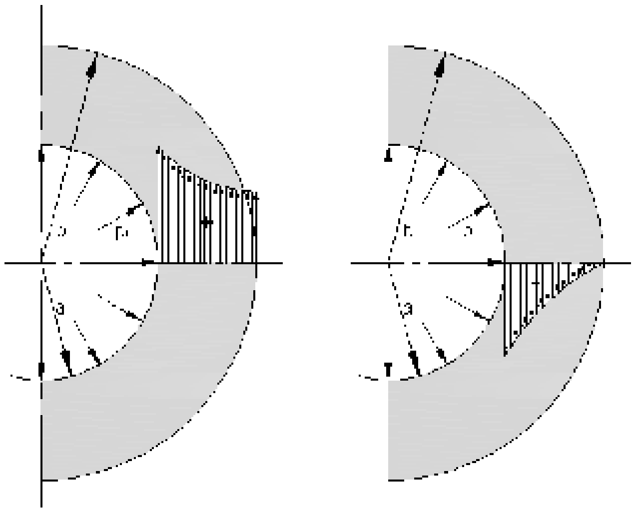

Figure 3.10 Stress distributions of pipe

For the failure analysis of ductile materials distortion-energy theory is used, since the distortion energy theory predicts failure most accurately. The distortion-energy theory states that, yielding begins when the total strain energy stored in the stressed element reaches a critical value, and it becomes,

$$s' = \frac{1}{\sqrt{2}}\sqrt{(s_1 - s_2)^2 + (s_1 - s_3)^2 + (s_2 - s_3)^2} \qquad (3.21)$$

Minimum stress value is zero when there is no pressure inside pipe. In other means, this condition occurs during refilled pipe. Maximum stress occurs when the pressure is generated during pressure intensification stage of the intensifier. Using the distortion-energy theory, the stresses values are obtained as,

$$s_{\min} = 0 \tag{3.22}$$

$$s_{\max} = \frac{1}{\sqrt{2}}\sqrt{\left(s_r - s_t\right)^2 + \left(s_r - s_l\right)^2 + \left(s_t - s_l\right)^2} \tag{3.23}$$

In fatigue failure analysis, the amplitude and mean stress value are calculated based on Shigley 1986,

$$s_{mean} = \frac{s_{\max} + s_{\min}}{2} \tag{3.24}$$

$$s_{amp} = \frac{s_{\max} - s_{\min}}{2} \tag{3.25}$$

And due to this criterion the factor of safety of the pipes are,

$$n = \frac{1}{\dfrac{s_{mean}}{S_{ut}} + \dfrac{s_{amp}}{S_e}} \tag{3.26}$$

To find S_e Eq. (3.6) and (3.7) can be used.

3.1.2.3 Design of Check Valve and Check Valve Block

In the current design, check valves and check valve block was designed and built up considering cylindrical shrink-fit process.

In this study, to design check valve and check valve block a program which was written for multi layer cylindrical structures which is called DPro-MLWJI by KUNI A. (2008) is used. But this program is adapted for conical shrink-fit process.

3.1.3 Conical Shrink-Fit

3.1.3.1 Theory of Multi-Layer Structures for Conical Form

Conical shrink-fit connections are often used in practices. Outer cylinder which inside is operated conic and inner cylinder which outside is operated similar conicity is fastened each other with an axial force. This force can be created by tightening with any nut. Figure 3.11 shows the conical pre-shrink-fit dimensions.

In the conical shrink-fit process, contact surfaces are assumed to be continuous and circular. So, surface pressure is assumed to act equally each point.

Figure 3.12 shows the surface pressure, forces act on the inner and outer cylinder, conical post-shrink dimensions.
The normal force which acts perpendicular to the surfaces and friction force are,

$$F_N = p.r_m.h.p \tag{3.27}$$

$$F_f = m.F_N \tag{3.28}$$

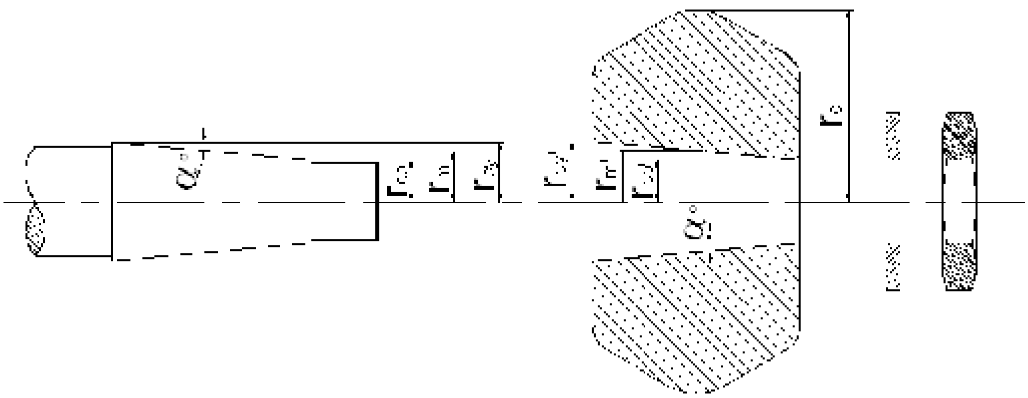

Figure 3.11 The dimensions before shrink-fit

Where, p is surface pressure, r_m is mean radius noting that $r_m = (r_{cy} + r_{co})/2$, h is conic surface width and m is friction coefficient.

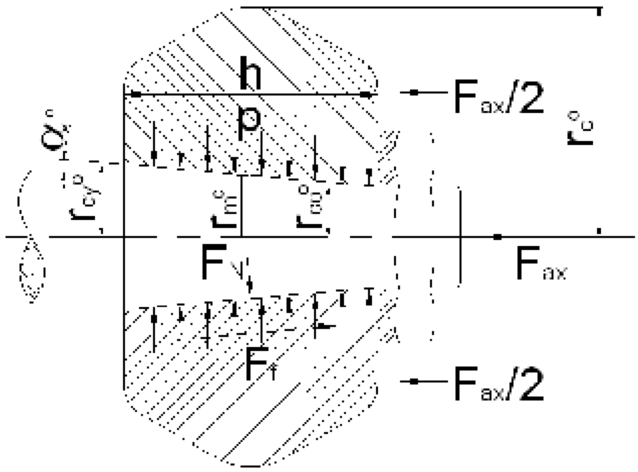

Figure 3.12 Forces of conical shrink-fit

The relation between cylindrical and conical radius is,

$$\tan a = \frac{r_{cy} - r_{co}}{h} \Rightarrow r_{co} = r_{cy} - h \tan a \tag{3.29}$$

Where, a is conic surface slope angle.

Not to create any plastic deformation on the structures, surface pressure must be $p \le p_{all}$. Outer cylinder which inside is operated conic is considered as a thick-walled vessel which is exposed to ‘‘ p ’’surface pressure. In this case, $Q_o = r_m / r_o$ is dimensional constant, maximum equivalent stress is,

$$s_{o\max} = p \frac{\sqrt{3 + Q_o^{\ 4}}}{1 - Q_o^{\ 2}} \le s_{all} \tag{3.30}$$

Considering above equation the allowable stress is,

$$s_{all} = \frac{s_y}{S} \tag{3.31}$$

Where, S is safety factor and equal to $S = 1{,}1....1{,}3$.

The important point is the interface pressure must not exceed the plastic deformation region. The maximum interface pressure is,

$$p_{\max} \le p_{all} = s_{all} \frac{1 - Q_o^{\ 2}}{\sqrt{3 + Q_o^{\ 4}}} \tag{3.32}$$

Theoretical radius difference is,

$$\Delta d = \frac{r_m p}{E_i}\left(\frac{1+Q_i^{\ 2}}{1-Q_i^{\ 2}} - n_i\right) + \frac{r_m p}{E_o}\left(\frac{1+Q_o^{\ 2}}{1-Q_o^{\ 2}} - n_o\right) \tag{3.33}$$

In the practices, for providing ‘‘ p ’’ surface pressure, axial force which is applied to the nut have to overcome horizontal component of normal forces and frictional forces seen in Figure 3.13.

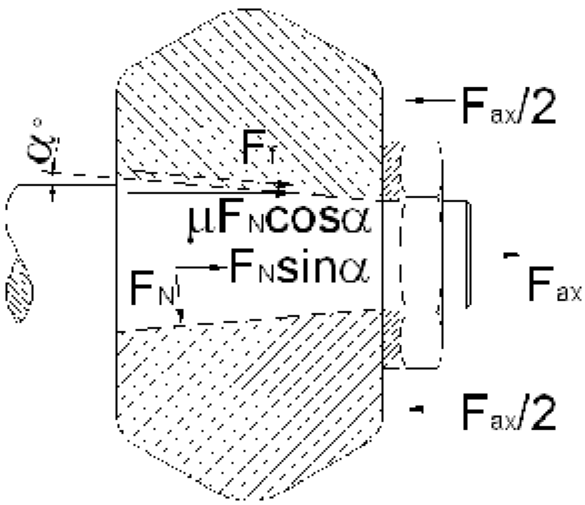

Figure 3.13 Schematic views of axial forces

$$F_{ax} = F_N \left(\sin a + m\cos a\right) \tag{3.34}$$

To dismantle connection, requiring axial force by considering change of friction force's direction is,

$$F_{di} = F_N \left(\sin a - m\cos a\right) \tag{3.35}$$

Where, ‘‘ F_{di} ’’ is dismantling force.

Conical fitted connection does not unfasten itself. So, self locking must be provided. According to this, the self locking condition is,

$$\tan a \le m \tag{3.36}$$

Self locking condition is ensued in the form of $a \le r$. ‘‘ r ’’ is friction angle. Figure 3.14 shows the self locking condition schematically.

Disassemble of connection is difficult when ‘‘ a ’’ angle decrease. In the conical shrink-fit process, if required force to tighten the nut is high, pressurized oil method can be used. If pressurized oil is used between surfaces, the tightening force for assembly is smaller.

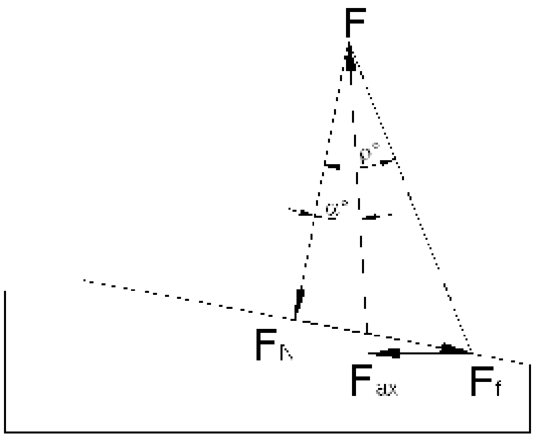

Figure 3.14 Schematic view of self-locking condition

Pressurized oil decrease friction coefficient therefore, required axial force is decreased. If $m_{dry} = 0{,}1$ at the dry condition, $m_{oiled} = 0{,}01$ or smaller value. According to this, required axial force can be decreased 10 or more times.

$$F_{dry} / F_{oiled} = m_{dty} / m_{oiled} \tag{3.37}$$

3.1.3.2 Development of Computer Program for Conical Shrink-Fit Process

To design check valve and check valve block a program which was written for multi layer cylindrical structures is adapted for conical shrink-fit process. This program is called CoSF-Pro means that ‘‘ Conical Shrink-Fit Program’’.

The interface pressure is calculated by the cylindrical shrink fit program using iteration. The difference between the conical shrink-fit program and the cylindrical shrink-fit program is mainly due to interface pressure calculation. The interface pressure is calculated from the equations 3.27 and 3.34. In addition to these, in the cylindrical shrink-fit program, total interface displacement is entered manually. But, in the conical shrink-fit program, the total interface pressure is calculated based on the summation of radial displacement of outer surface of inner cylinder and inner surface of outer cylinder.

To calculate interface pressure, axial force is entered manually in the conical shrink-fit program. In the conical shrink-fit program, the important point is the interface pressure should not exceed the plastic deformation region. Based on the equation 3.31 and 3.32 the axial force value is given by considering this important situation. In addition to this, the maximum internal pressure does not exceed 180 Mpa in practice.

3.1.3.3 Flow Diagram of the CoSF-Pro

The flow diagram of the program is given in Figure 3.15. To initiate the program, the inputs about ‘‘ p_o’’, ‘‘ p_i ’’, ‘‘ F_{ax} ’’, ‘‘ R ’’, ‘‘ t_1 ’’, ‘‘ t_2 ’’, ‘‘ h ’’, ‘‘ a ’’ and material properties are entered manually into the program. Then, press the

Numpad-Enter to run the program. After this step, all of the terms in the notebook will be calculated in a predefined order. The flow chart below shows the main steps of calculation sequence.

The program firstly reads the values and assigns the basic inputs, then calculates the ‘‘ p ’’ paramatically. Then, the program calculates ‘‘d_1’’ and ‘‘d_T’’ analytically, after that it is calculated numerically, too. Then according to the solution of both parameters, program calculates the all other parameters. But, the important point in here is, the results will be correct in only and only if numeric solution of the ‘‘d_1’’ and ‘‘d_T’’ exists in the analytical solution set.

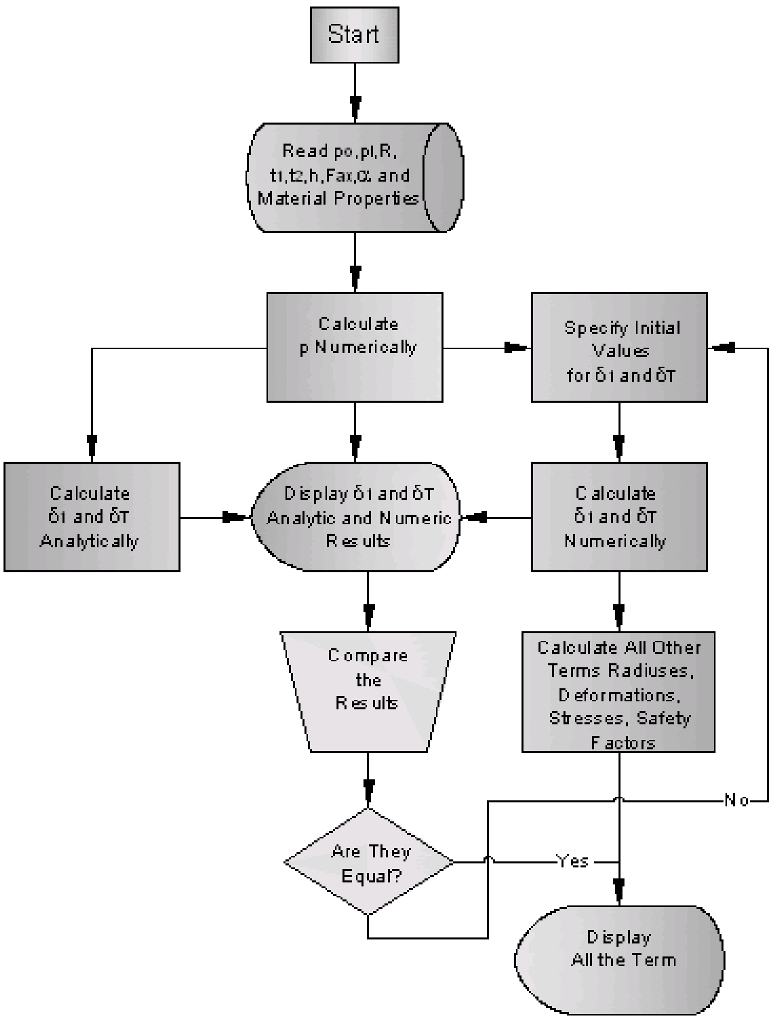

Figure 3.15 Flow chart of the conical shrink-fit program

3.1.3.4 Flow Diagram of the both Cylindrical and Conical Shrink-Fit Program

After opening the program, two directives are seen on the screen. First directive is ‘‘If the Problem Cylindrical Form Click on Cylindrical Shrink-Fit’’, second directive is ‘‘If the Problem Conical Form Click on Conical Shrink-Fit’’. In this point, according the type of problem, one directive is selected and run the program. Figure 3.16 shows the flow chart of the both cylindrical and conical shrink-fit program.

3.1.3.5 Explanation of the Steps of Program CoSF-Pro

The steps of the conical shrink-fit program as follows. The conical shrink-fit program and steps are seen in the Appendix A.

Step1: The inputs about the working conditions and design parameters are expressed.
Step2: The inputs about the material property constants are entered. Endurance limits are calculated using equations (3.6) and (3.7).
Step3: The “X” values collect the repeating terms in the equations. So it reduces the complexity of the equations.
Step4: This step calculates the pre-shrink fit dimensions.
Step5: In this step the interface pressure is calculated.
Step6: This step calculates stress constants that are calculated for the pre-shrink fit situation.
Step7: This step calculates the total deformation values of the inner and outer surfaces of the both cylinders.
Step8: In this step the displacement of the inner conical cylinder inner surface and total interface displacement are calculated. “Solve” command was used Mathematica solves the system by substituting the numeric constants. The total interface displacement and deformation values of inner surface of the inner cylinder are shown in output. One of them is real and the other is redundant.

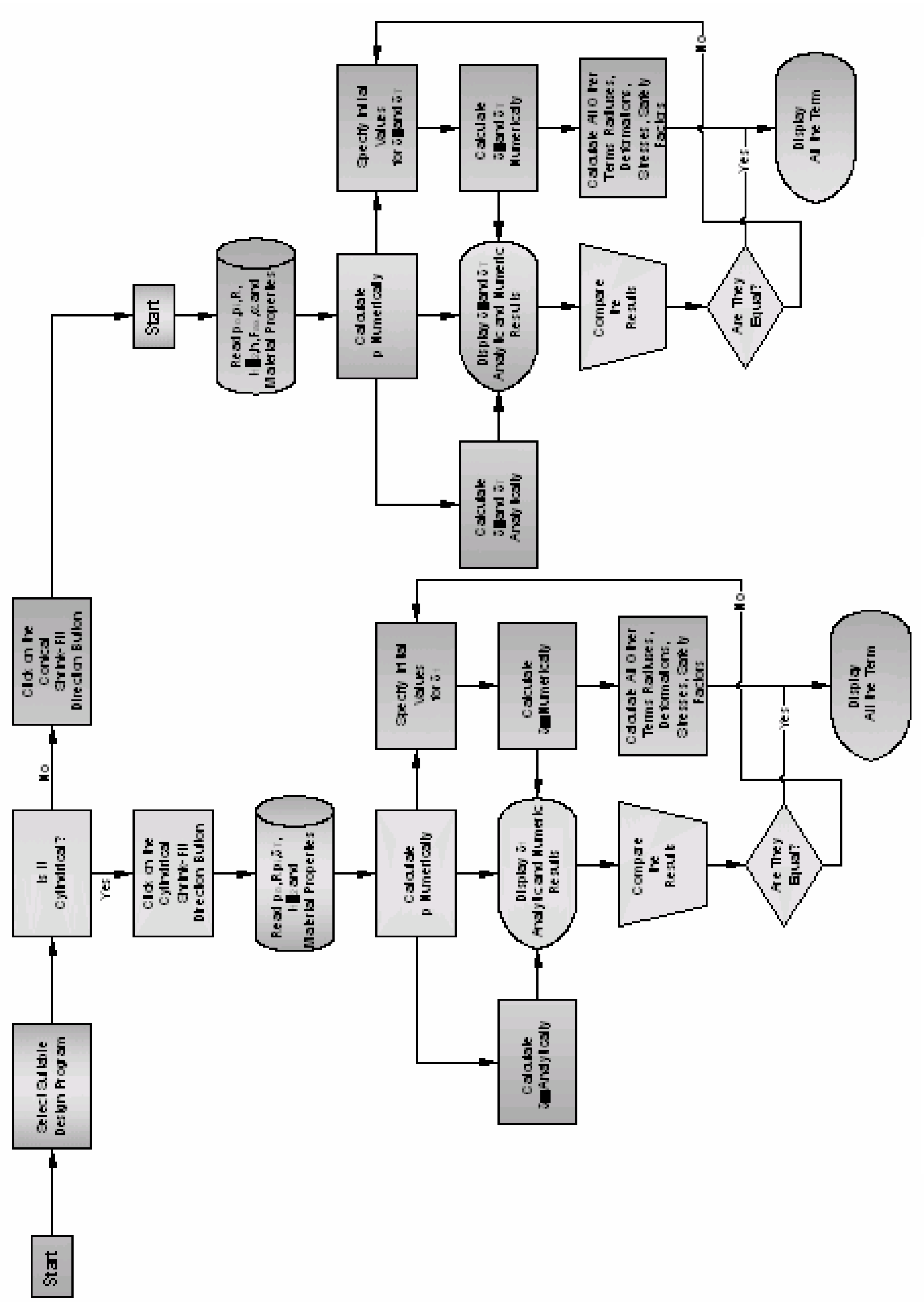

Figure 3.16 Flow chart of the both cylindrical and conical shrink-fit program

Step9: "FindRoot" command finds the root of the system by numerically way. The result or output is the subset of the previous output sets with only one element. Thus the solution is correct and accurate.

Step10: This step assigns the last found results for ''d_1'', '' p '' and ''d_T '' to them again. "%" means previous result here and "%%" means two times previous, "%%%" means three times previous.

Step11: This step calculates the post shrink fit dimensions.

Step12: The endurance limit modifying factors are calculated and assigned. The endurance limits of the cylinders are calculated.

Step13: This step calculates the stress constants under only interface pressure.

Step14: This step calculates the stress constants under only operating pressure.

Step15: This step calculates the radial, tangential, and longitudinal stresses for inner cylinder in only operating pressure and in only interface pressure separately.

Step16: This step calculates the total radial, longitudinal and tangential stresses of the inner cylinder.

Step17: This step calculates the minimum and maximum stresses for the inner cylinder; and using these, it calculates the mean and amplitude stresses.

Step18: In this step, the factor of safety of the inner cylinder is calculated.

Step19: This step calculates the radial, tangential, and longitudinal stresses for outer cylinder in only operating pressure and in only interface pressure separately.

Step20: This step calculates the total radial, longitudinal and tangential stresses of the outer cylinder.

Step21: This step calculates the minimum and maximum stresses for the outer cylinder; and using these, it calculates the mean and amplitude stresses.

Step22: In this step, the factor of safety of the outer cylinder is calculated.

Step23: This step represents of the all calculated variable in table form. "N" at the beginning of the sentence represents that, results must be numeric, not to be rational or other kinds.

Step24: Is to clear the all saved data to prevent the assignment errors in next iteration.

3.1.4 Redesign

As stated in the previous studies, the redesign process consists of 3 main steps. First 2 steps have been discussed up to this point. The last step discusses to achieve the new prototype in redesign process. Because of this, step 3 is given in chapter 4 (Result and Discussion). In the chapter 4, the high pressure connection parts of WJM system has been investigated by using reverse engineering and redesign methodology. Then, using the conical shrink-fit program, optimum dimension of the check valve and check valve block will be calculated.

4. RESULTS AND DISCUSSION

As mentioned in the previous chapters, the high pressure connection parts of WJM system has been investigated by using reverse engineering and redesign methodology. In this chapter, the results of the reverse engineering and redesign methodology steps was explained. Then, using the conical shrink-fit program, optimum dimension of the check valve and check valve block was calculated.

4.1 Results of the Reverse Engineering

4.1.1 Investigation

To achieve target specifications prototype needs, requirements and expectation have been identified and abstracted for new design. TRIZ (Theory of Inventive Problem Solving) method is used to find solution for high pressure connection parts.

For all of the high pressure parts, black box model is created which is shown in Figure 4.1 to identify the input and output flows of the product.

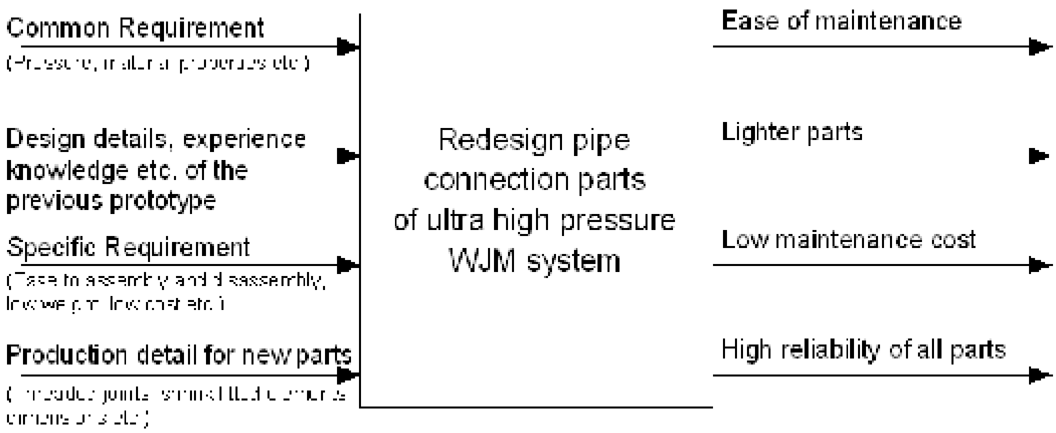

Figure 4.1 Black box model

The new design and current design specifications are compared each other. Then, these specifications are marked according to importance. To mark weight and current design value analysis is used as mentioned in chapter 2. In the value analysis, the values are given by designer according to the relative importance of the needs into the value scale in Table 4.1.

Table 4.1 Points awarded in use-value analysis

Value Scale	
Points	Meaning
0	absolutely useless solution
1	very inadequate solution
2	weak solution
3	tolerable solution
4	adequate solution
5	satisfactory solution
6	good solution with few drawbacks
7	good solution
8	very good solution
9	solution exceeding the requirement
10	ideal solution

4.1.1.1 Cup-Head and Sealing Head

The head is fixed to the water cylinder by metric threads. Between head and water cylinder, sealing head is available in the current design. The problems about the head and sealing head are total weight of head and sealing head is high, then related with weight, assembly is hard to achieve. Addition to these, manufacturing cost is high. So the weight of the head and sealing head must be decreased.

Figure 4.2 shows assembled view of the head, sealing head, conic aluminum disc seal and water cylinder of left side of intensifier unit.

4.1.1.2 High Pressure Pipes

High pressure hoses are fixed to the sealing head and check valve block, check valve block and cutting nozzle to transfer high pressure water from intensifier.

The hoses are used in the current design. However the hose is expensive and too many waiting for delivery.

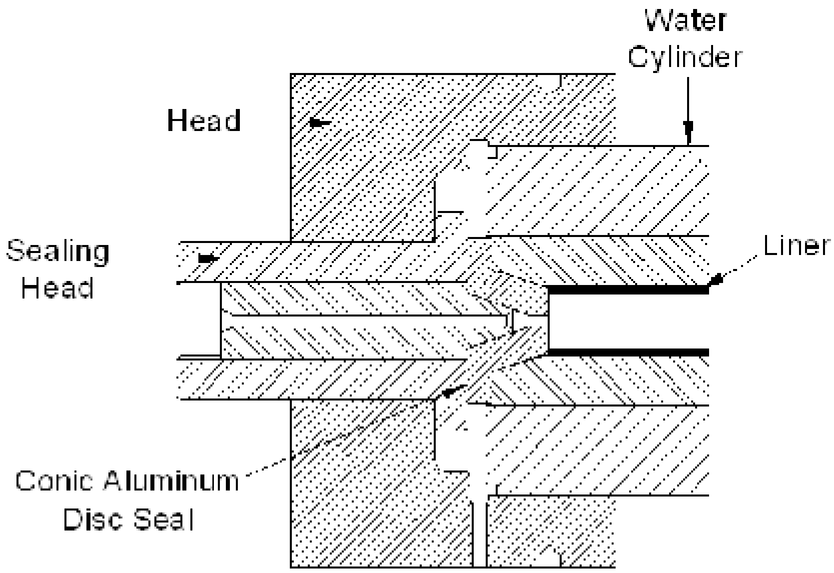

Figure 4.2 Assembled view of cup-head and sealing head of current design

4.1.1.3 Check Valve and Check Valve Block

Four of the check-valves are located into a cylindrical check-valve block. Check valve block is fixed to the sealing head with high pressure hoses. Addition to this, check valve block is connected the cutting nozzle with high pressure hose, too.

The problems about the check valve and check valve blocks are assembly and disassembly of the check valve and check valve block for the difficulty of shrink fit process and high weight of the check valve block.

4.1.2 Product Tear Down and Experimentation

Eliminating the parts, reducing manufacturing cost and redesign the high pressure parts for ease of assembly and disassembly which is outlined chapter 3 is applied to the current design in the product tear down and experimentation step. Figure 4.3 shows the exploded view of high pressure parts to apply product tear down in the current design.

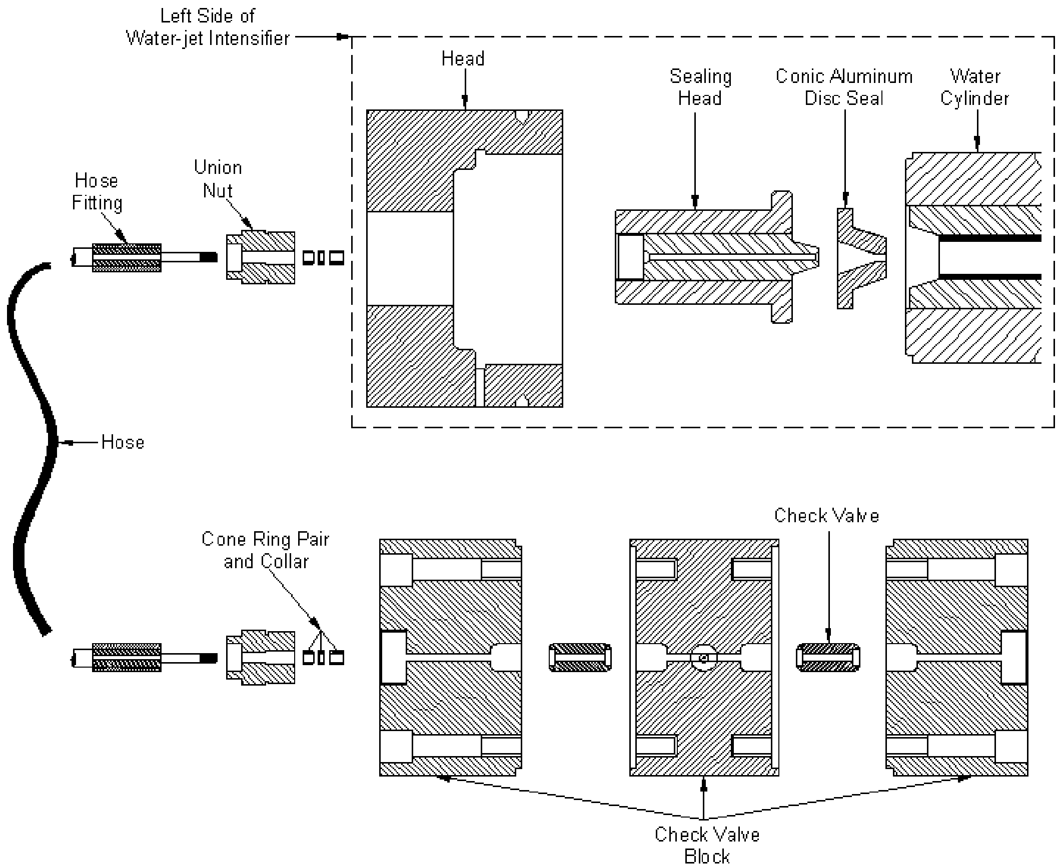

Figure 4.3 Exploded view of high pressure connection parts in the current design

4.1.3 Functional Analysis

Figure 4.4 gives the function mean tree to general understanding and then breaking the problem down into sub-problems of high pressure connections parts of current prototype.

The whole system function flows are to receive the filtered and conditioned pure water, to intensify the water pressure, and to send the high-pressure water to the nozzle. The function of high pressure connection parts is to take the high pressure water from water cylinder then send to nozzle. Figure 4.5 shows the function structure of high pressure connection parts.

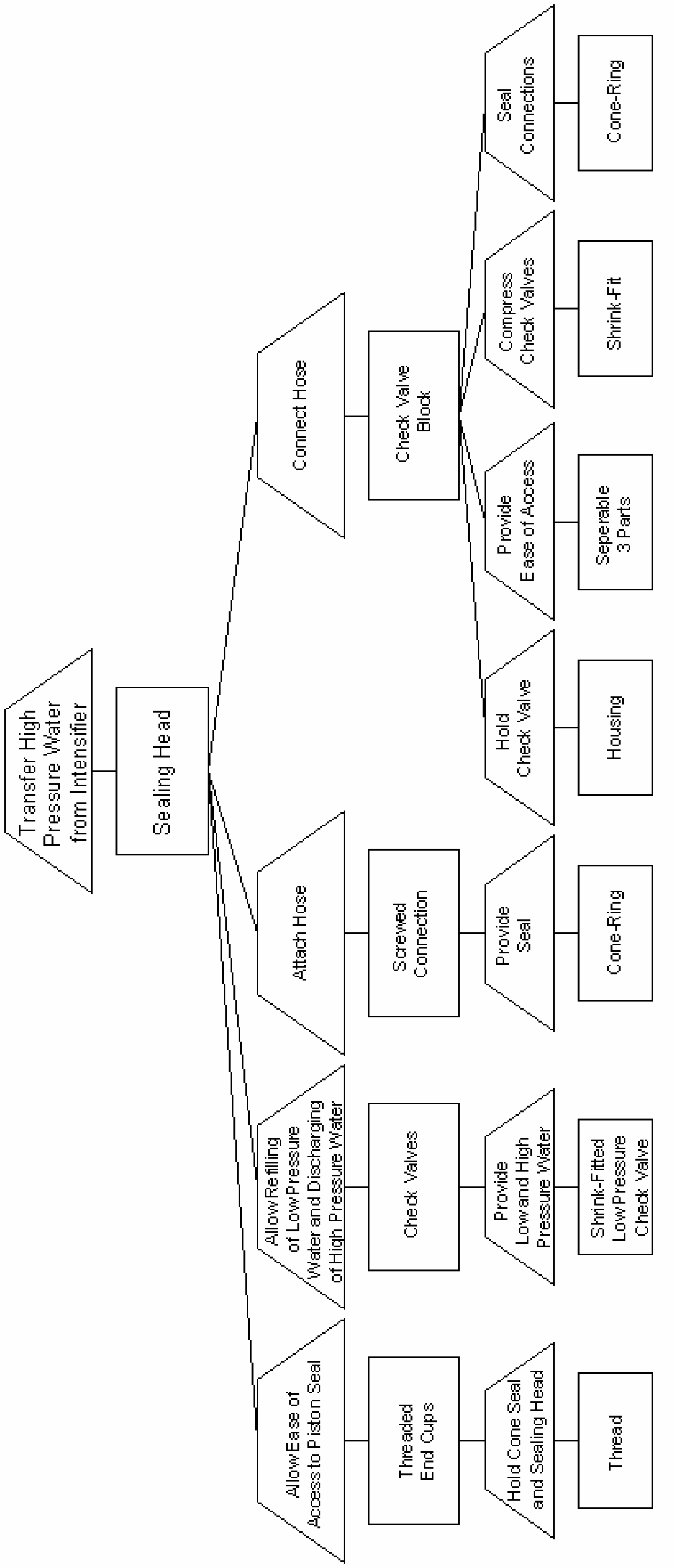

Figure 4.4 Function mean tree of high pressure parts of current prototype

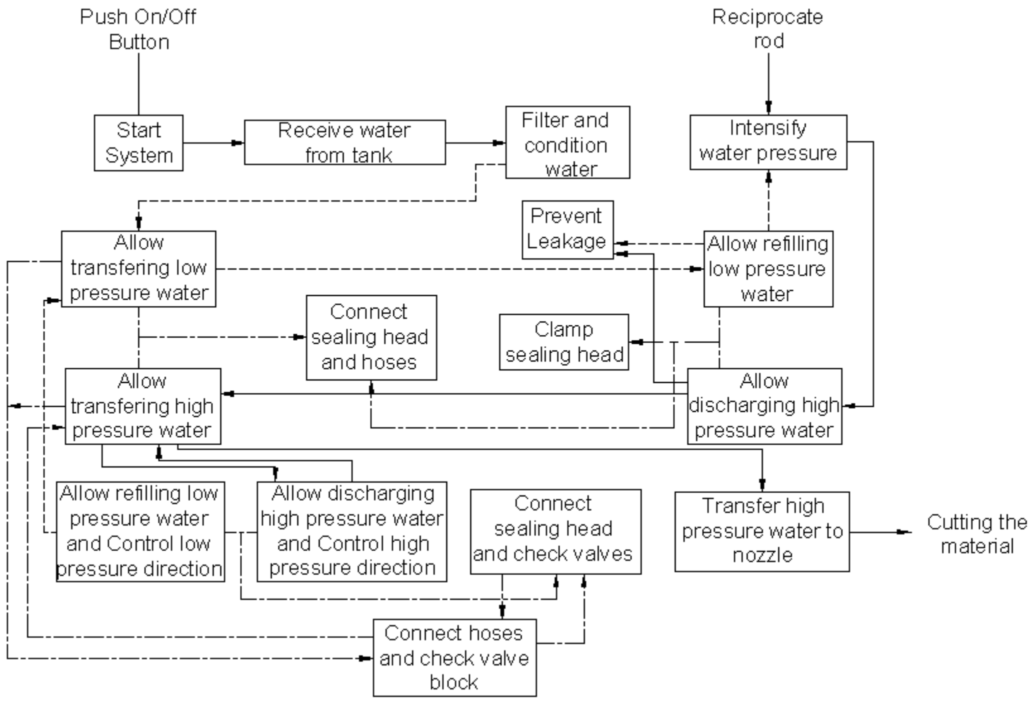

Figure 4.5 High Pressure Parts Function Structure – Actual Product

4.1.4 Constraint Propagation

In the new design, based on the previous prototype, alternative solutions are produced considering the function of the high pressure parts for desired specifications. Table 4.2 shows the morphological matrix for the current design and new design.

4.1.5 Forming Engineering Specifications

Based on comparison of the current and new design and requirements lists for every high pressure parts the target specifications are achieved.

4.2 Redesign

Based on the results obtained from steps 1-7, high pressure parts are redesigned. Focusing the prototype need ‘‘ease of assembly and disassembly, reducing the manufacturing cost of high pressure parts’’ is obtained. Figure 4.6

shows the exploded view of new designed whole system and Figure 4.7 shows the assembled view of new desgined whole system.

Table 4.2 Morphological matrix of high pressure parts

Function	Part Name	Current Solution	Possible Solution
-Allow easy access to piston seal head -Clamp and deform aluminum disc seal -Allow refilling of low pressure water and discharging of high pressure water from water cylinder to the check valves	**Head**	Threaded cylinder end cups	Threaded cylinder end cups with optimum dimension
-Control the direction of high and low pressure flows	**Check Valve**	Modified commercial low pressure check valves fitted as cylindrical shrink-fit	Modified commercial low pressure check valves fitted as conical shrink-fit
-Connect high and low pressure hoses -Holding check valves -Compressing check valves	**Check Valve Block**	-Cylindrical 3 part separable block -To house check valve operating cylindrical	Cylindrical 3 part separable lighter block -To house check valve operating inner side conic
-Flow high and low pressure water	**Pipes**	Multi layer hoses for high pressure	Metal ultra high pipes

4.2.1 Redesign of Cup-head

To design the head, the most important point is to consider replacement of piston head seals. The piston head seals have a limited life and these require frequent replacement. Therefore, the access of piston head seals must be easy. Consequently, considering the piston head seal, the head design gains important role. The primary goal of head design is ease of assembly and disassembly.

Based on the TRIZ method, the improving and worsening features has been decided. The improving feature decrease weight of stationary object and then the worsening feature are loss of subtance and quantity of subtance. Based on the contradiction matrix (see Chapter 2, Table 2.2) the ideal solutions techniques are 5,8,13,30 and 6,18,19,16 numbered inventive solutions seen in Table 4.3.

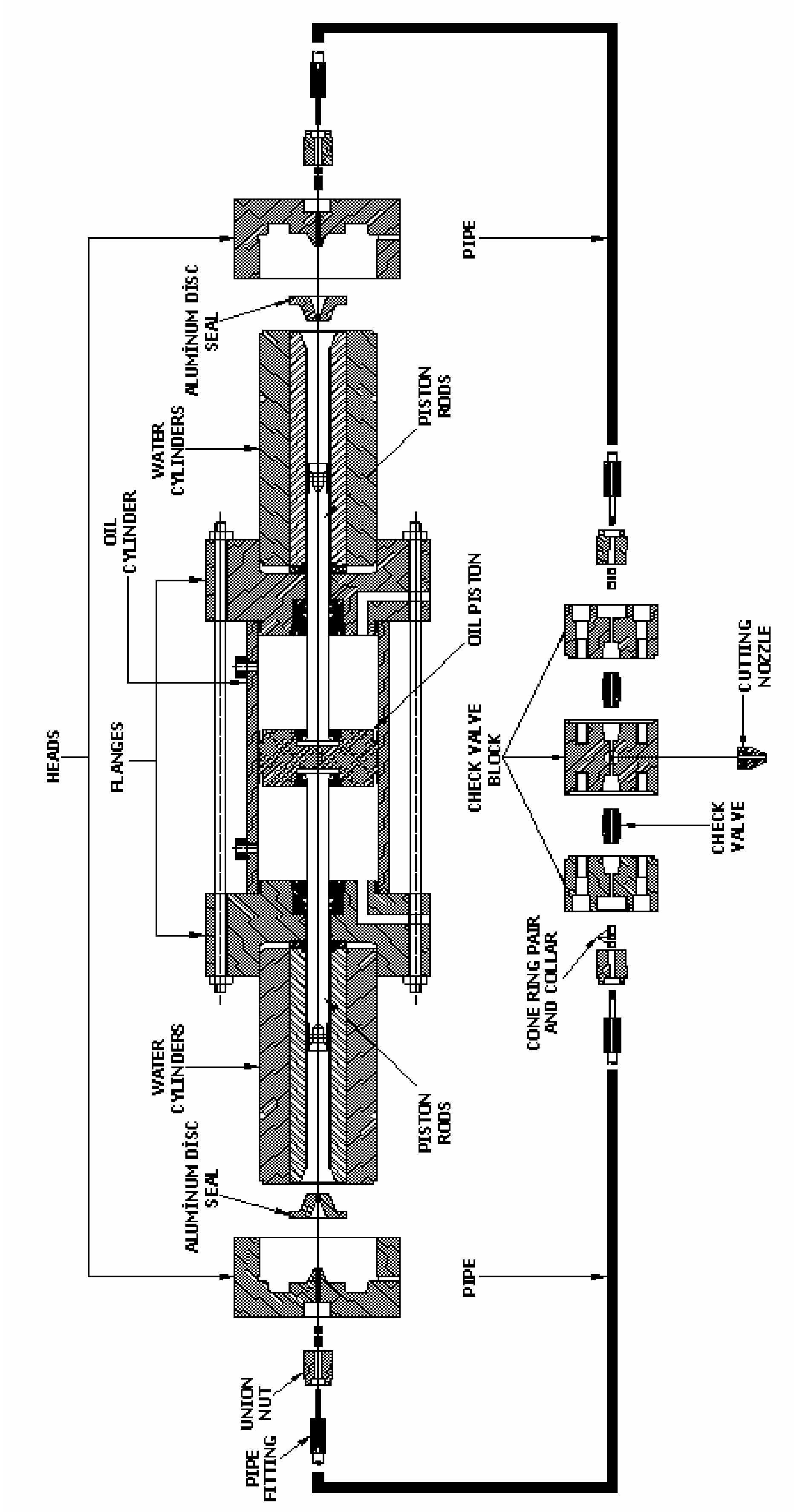

Figure 4.6 3000 bar operating pressure intensifier with exploded view of new design high pressure parts.

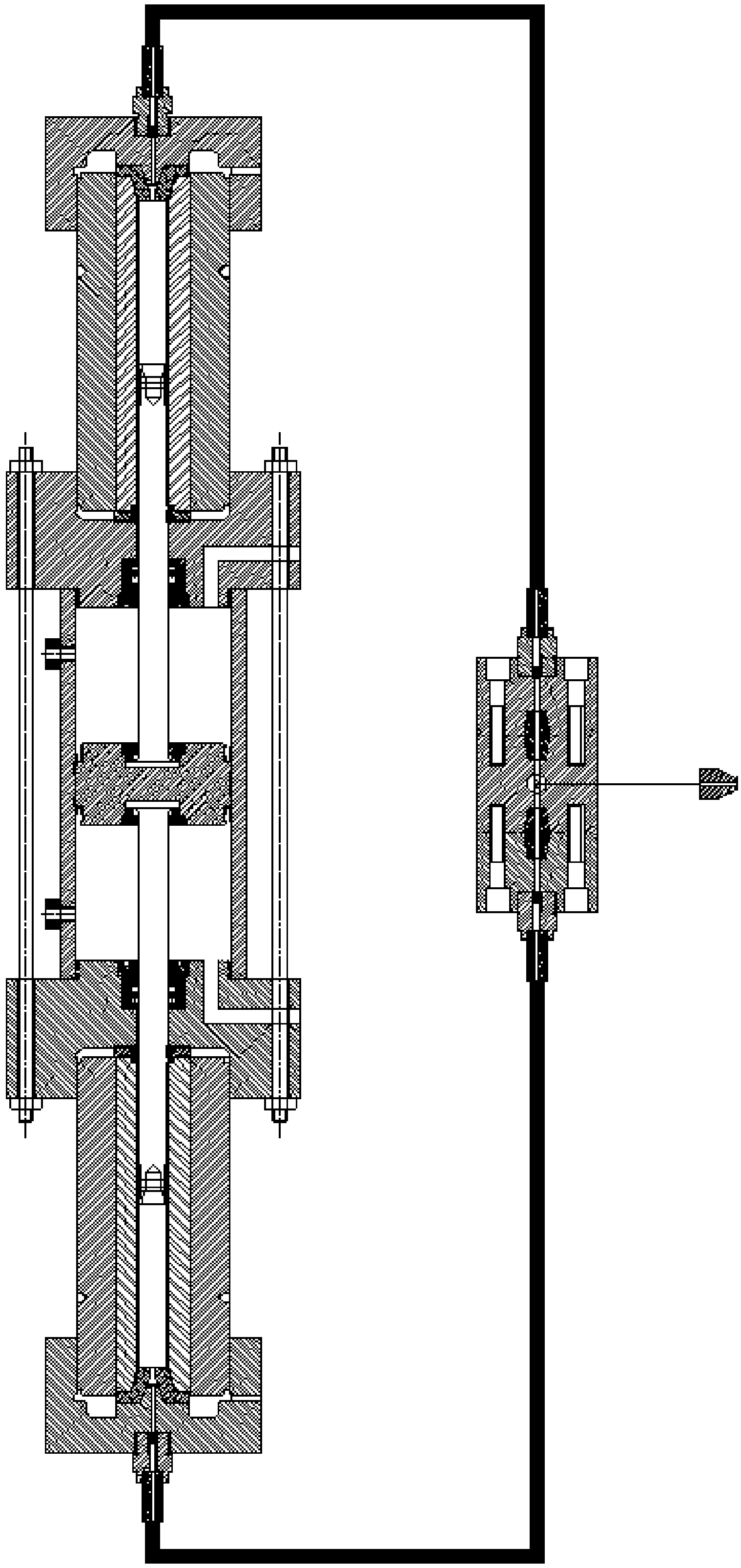

Figure 4.7 The assembled view of 3000 bar operating pressure intensifier with new design high pressure parts.

Based on the inventive solution ‘‘5 Merging’’, ‘‘6 Universality’’ ,‘‘8 Anti-weight’’ and ‘‘13 The other way round’’ the weight of the head is decreased and the sealing head is canceled. The sealing head is combined to the head.

Table 4.3 Inventive solutions for head and sealing head

5 Merging	8 Anti-weight
13 The other way round	30 Flexible shelles and thin films
6 Universality	18 Mechanical vibration
19 Periodic action	26 Copying

The comparison of the new designed ‘‘head-sealing head’’ combination and previous prototype head and sealing head is as follows.

- Weight of new designed head is lower than current design.
- Length of new designed head is lower than current design.
- The complexity of new designed head is a bit more than current design.
- Sealing head is eleminated in the new design of system.

Some analysis is made by considering the comparison of new design and current design;

- Ø Less material is used to manufacture the new head.
- Ø Less manufacturing time to manufacture the new head.
- Ø The total cost of the new designed head for material and labor is lower than the current design.
- Ø To reach piston sealing head, number of parts which are dismantled is lower. Therefore, the assembling and disassembling time can be lower.
- Ø To replace piston head seal 2 connections and 3 parts must be dismantled. But new design 2 connections and 2 parts must be dismantled.

The design of the heads is based on threaded joints. Design of heads has been carried out based on the threaded stresses. Therefore, the thread dimensions are obtained for tension, shear and bearing loading conditions. As a result of the calculation and design decisions of shear loading, M156X2 thread has been found to be suitable. In addition, this metric thread was also check under bearing loading condition. Design of threads based on the criteria of bearing loading was performed and 27 engaged threads were found to be sufficient for safe operation.

According to these comparison and considering value analysis product needs list and weight/diagnosis weakness has been prepared and shown Table 4.4. Figure 4.8 shows the new designed function mean tree. Figure 4.9 shows schematic views of assembled head, aluminum disc seal and water cylinder.

Table 4.4 Product needs list and weight/diagnosis weakness cup-head and sealing head

Part Name Product Need	Head and Sealing Head	Current Design Weight	New Design Weight
I. Size	a) Light weight for ease of assembly	4	7
	b) Optimum dimension	5	7
II. Safety	a) Able to stand maximum pressure	10	10
III. Reliability	a) High reliability of head and sealing head subjected high pressure	9	9
IV. Replacement	a) Ease of replacement piston seal head	6	8
	b) Lower time for replacement piston seal head	6	8
V. Maintenance	a) Ease of assembling and dismantling	5	7
	b) Lower assembling and dismantling time	5	7
VI. Manufacturing	a) Ease of manufacturing head and sealing head for WJM system	4	7
	b) Lower manufacturing time	5	7
VII. Cost	a) Manufacturing and labor cost	4	8

4.2.2 Redesign of High Pressure Pipes

Based on the TRIZ method, the improving feature decrease loss of time. In the new design instead of hoses, pipes are used to transfer high pressure water. Comparison of the new design pipes and high pressure hoses are given below.

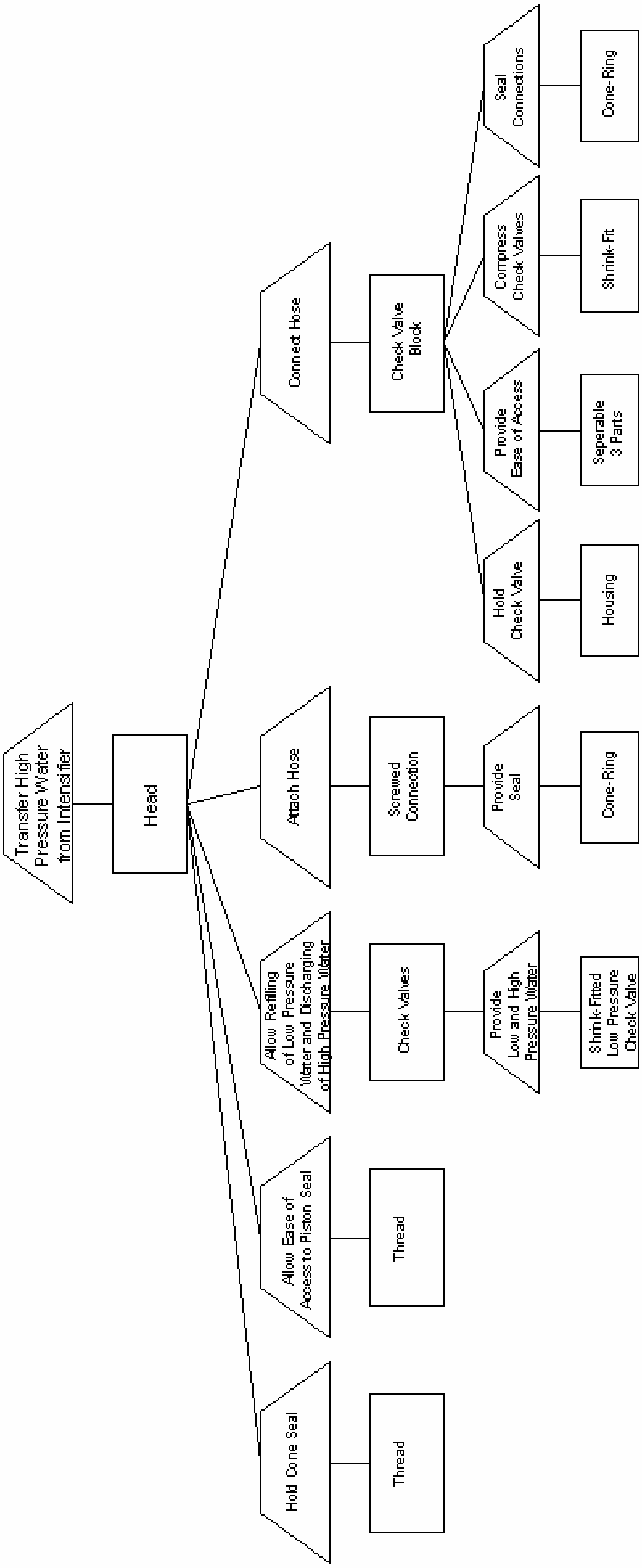

Figure 4.8 The function mean tree of new design high pressure connection parts.

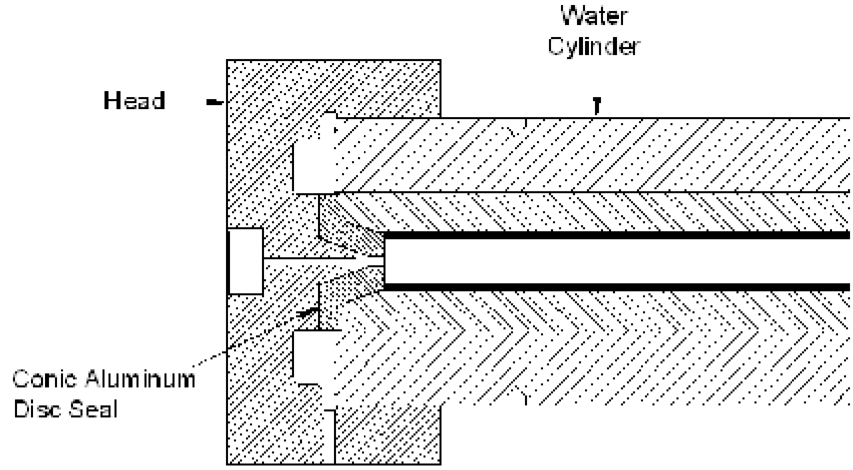

Figure 4.9 View of assembled cup-head and water cylinder in the new design

- High pressure hoses are not manufactured in our country. Hoses must be brought from abroad and there is not many manufacture company. Pipes are manufactured in our country.
- While importing high pressure hoses, delivery time can be long. But, pipes can be obtained easily in the market.
- Because of hoses special structure and importing from abroad hoses are more expensive than pipes.
- The high pressure hoses flexible but pipes are not. However, pipes can be bend easily in any work shop.
- Total cost of hoses is more than pipes.

According to above comparison product needs list and weight/diagnosis weakness has been prepared and shown in Table 4.5.

Table 4.5 Product needs list and weight/diagnosis weakness of hoses and pipes

Part Name **Product Need**	**High Pressure Pipes**	**New Design Weight**	**Current Design Weight**
I. Safety	a) Able to stand maximum pressure	10	10
II. Reliability	a) High reliability of high pressure pipes subjected high pressure	9	9
III. Replacement	a) Delivery time	7	3
IV. Cost	a) Total cost	8	4

High pressure pipes are fixed to the head and check valve block, check valve block and cutting nozzle to transfer high pressure water from intensifier. High pressure pipes are widely available from Turkish manufactures. The material for the selected pipes is ST52 steel which tensile strength is between 490-620 Mpa. Standard pipe sizes are selected to check it for 3000 bar operating pressure.To calculate the pipe stresses under high stresses the computer program which has been written previously (see Appendix C) has run to obtain optimum pipe diameter to resist the pressure. Table 4.6 shows the properties of materials and program inputs. Table 4.7 shows the output of the program results.

Table 4.6 The program inputs and material properties

Parameter	**Unit**	**Value**	**Parameter**	**Unit**	**Value**
p_o	Pa	0	a	m	5.10^{-3}
p_i	Pa	3000.10^5	t	m	$2,5.10^{-3}$
S_{ut}	Pa	6200.10^5	b	m	10.10^{-3}
S_Y	Pa	3550.10^5			

Table 4.7 The program outputs

Parameter	**Unit**	**Value**	**Parameter**	**Unit**	**Value**
s_r	Pa	-3000.10^5	$s_{\max}$	Pa	9353.10^5
s_t	Pa	7800.10^5	$s_{\min}$	Pa	0
s_l	Pa	2400.10^5	FS	--	1,326

4.2.3 Redesign of Check Valve and Check Valve Block

Based on the TRIZ method, for check valve the improving feature is shape and then the worsening feature are force and stress or pressure. Based on the contradiction matrix the ideal solutions techniques are 10,35,37,40 and 10,14,15,34 numbered inventive solutions. For the check valve blocks the inventive solution 16 is used. Based on the contradiction matrix the ideal solutions techniques are 5,8,13,30 and 6,18,19,16 numbered inventive solutions seen in Table 4.8.

Table 4.8 Inventive solutions for check valves and check valve block

10 Preliminary action	14 Spheroidality - Curvature
15 Dynamics	34 Discharging and recovering
35 Parameter changes	37 Thermal expansion
40 Composite materials	

Based on the inventive solution ‘’10 Preliminary action’’, ‘‘14 Spheroidality - Curvature’’ , ‘‘15 Dynamics’’ and ‘‘35 Parameter changes’’ is used. Based on these, both side of the check valves is designed conical.

Comparison between new designed check valve, check valve block and current design provides the following.

- In the previous design check valves are in cylindrical shapes, but in the new design check valves are in conical form.
- In the current designed check valves and block, the shrink fitting process is applied with heat, but in the new designed check valve and block the shrink fitting process is applied with axial pressing force.
- Sealing was provided using plastically deformed soft copper cone rings of check valve in the current design. Sealing of the new design provides the conical shape of the check valves.
- Weight of new designed check valve block is lower than current design.

Some analysis is made by considering the comparison of new design and current design;

Ø Less material is used to manufacture the new designed check valve block.

Ø The material cost of the new designed block is lower than the current design.

Ø The use of extra material like soft copper cone rings for providing sealing action is eleminated due to the conical shape of the check valve which enables the sealing action.

Ø Extra manufacturing time and process to manufacture soft copper cone rings for sealing is eleminated.

Ø To manufacture soft copper cone rings there is not extra cost for labor and acquiring material.

- Ø In the current design, the assembly time of the check valve and block is higher than new design. Because shrink fit process with heating takes considerable time. But in the new design, there is no waiting time for shrink fit.
- Ø To reach the check valves, in the current design, the disassembly time of the check valve and block is higher than new design due to dismantling check valve and check valve block is very difficult process after cylindrical shrink fit process.

According to these comparison and results product needs list and weight/diagnosis weakness has been prepared and shown in Table 4.9.

Table 4.9 Product needs list and weight/diagnosis weakness check valve and check valve block

Part Name **Product Need**	**Check Valve and Check Valve Block**	**Current Design Weight**	**New Design Weight**
I. Size	a) Light weight for ease of assembly	3	4
	b) Optimum dimension	6	6
II. Safety	a) Able to stand maximum pressure	10	10
III. Reliability	a) High reliability of check valve and block subjected high pressure	8	8
IV. Sealing	a) Ease of removing plastically deforming soft metals	3	8
	b) Elastically deformed main part	-	9
V. Replacement	a) Ease of replacement check valves	3	6
	b) Lower time for replacement	3	6
VI. Maintenance	a) Ease of assembling and dismantling	4	6
	b) Lower assembling and dismantling time	3	6
VII. Manufacturing	a) Ease of manufacturing check valve and block for WJM system	6	7
VIII. Cost	a) Manufacturing and labor cost	6	7

Check valve and check valve block is designed based on the conical form. The check valves are shrink-fitted into their conical form holes, machined in the block. Shrink fitting process is applied with axial pressing. Sealing was provided by elastically deformed check valves due to conical form on the both sides.

The optimum design parameters for conical form check valve and check valve block are found in this study. The design program which is called CoSF-Pro was run for different conical angle, different conicity length, different axial pressing and so on. Figure 4.10 shows the new designed check valve and check valve block and Figure 4.11 shows the exploded view of the new designed whole system of the high pressure parts.

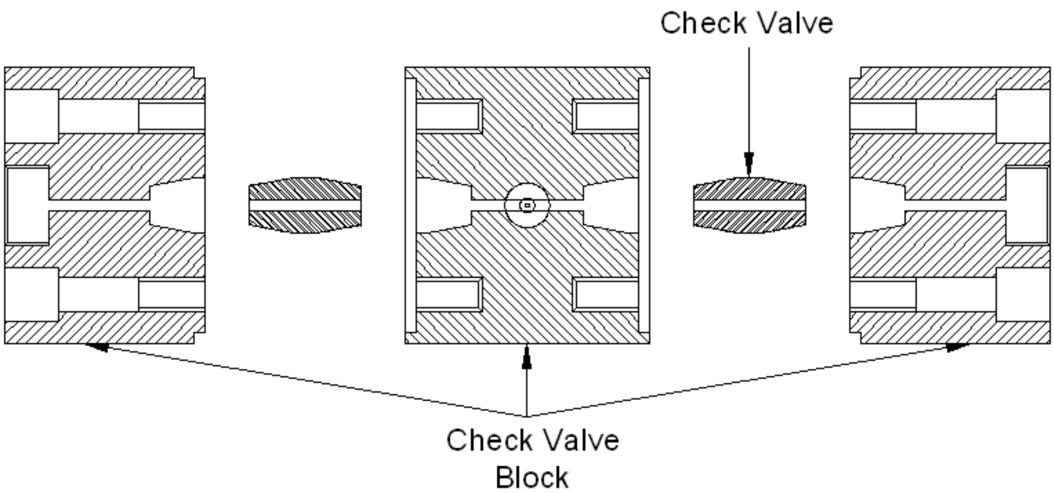

Figure 4.10 The new design check valve and check valve block

4.2.3.1 Methodology of Finding the Optimum Design Parameters

In the conical shrink-fit program, the important unknowns are interface pressure and total interface displacement. First of all, 5 degree value was assigned for conical angle and 10 kN value was assigned for the axial force. Then, axial force is increased in 5 kN increments starting with 10 kN. It was seen that increasing axial force causes increasing interface pressure and total interface displacement. In this approach, the consideration point is interface pressure which does not exceeded 180 Mpa. This process is repeated for 2.5 degree resolution to find results for different

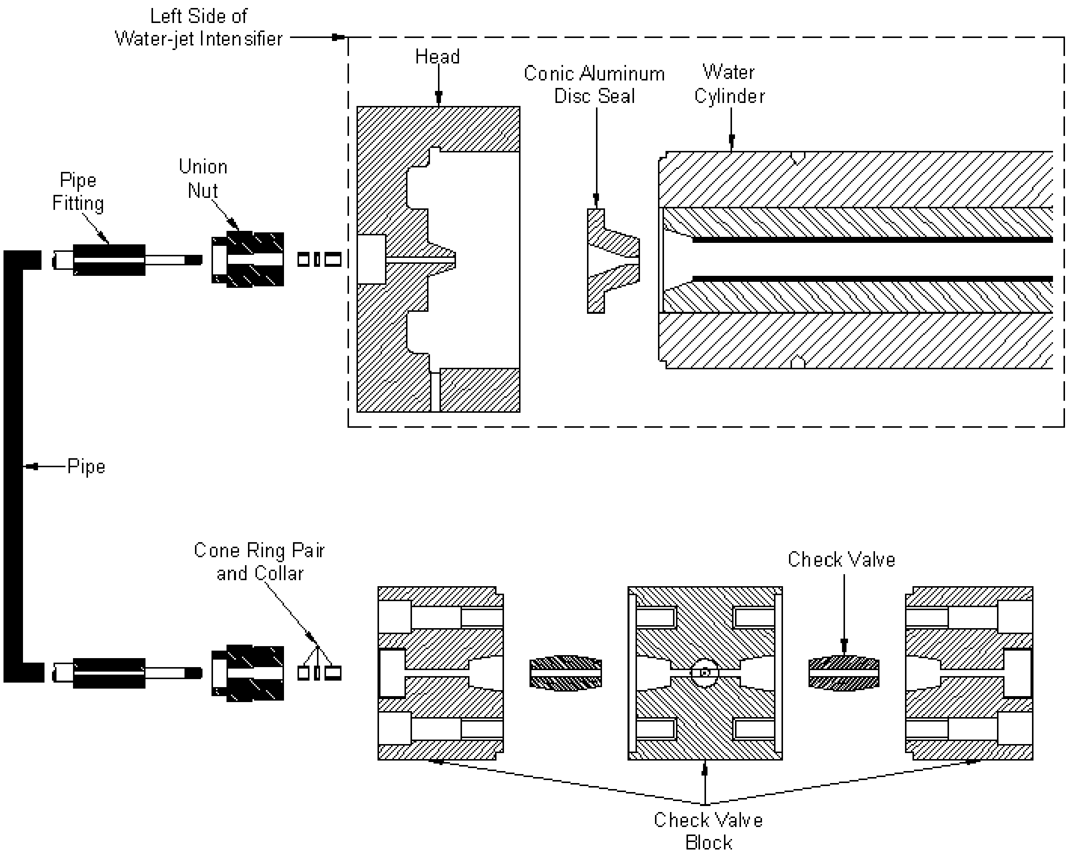

Figure 4.11 Expolded view of high pressure parts of new design

conical angles. In this process conicity length is equal to 20 mm. The results for interface pressure and total interface displacements are given in the Table 4.10.

Table 4.10 shows the internal pressure and total interface displacement changes against increasing axial force and conical angle. As seen in Table 4.7, the first calculation starts from the $a = 5^0$ and $F_{ax} = 10$ kN values. Then along the row, the axial force value is increased by 5 kN increments. Along the column, conical angle degree is increased by 2.5 degree increments. The internal pressure values and the total interface displacement values at that point were given.

As seen in the table, lower conical angle and higher axial force is increased the value of internal pressure and caused to exceed internal pressure up to 180 Mpa. The increasing axial force increases internal pressure. Beside this increasing conical angle decreases internal pressure. Considering these results, conical angle is chosen 10 degree and applied axial force is chosen as 20 kN. According to these results, factor of safety of inner and outer conical cylinders are found as 5.12 and 4.52 respectively.

Table 4.10 The internal pressure and total interface displacement values according to conical angle and axial force changes.

		F_{ax} (kN)					
		10		***15***		***20***	
		p	d_T	**p**	d_T	**p**	d_T
a (degree)	***5***	108.29	14.63	162.41	21.93	216.52	29.22
	7.5	83.74	11.31	125.59	16.96	167.44	22.61
	10	68.37	9.24	102.54	13.85	136.72	18.42
	12.5	57.86	7.83	86.78	11.72	115.70	15.63
	15	50.23	6.79	75.34	10.18	100.45	13.57
	17.5	44.46	6.01	66.68	9.01	88.91	12.01
	20	39.94	5.40	59.91	8.09	79.88	10.79
		F_{ax} (kN)					
		25		***30***		***35***	
		p	d_T	**p**	d_T	**p**	d_T
a (degree)	***5***	270.61	36.51	324.69	43.78	378.75	51.04
	7.5	209.28	25.25	251.11	33.38	292.93	39.51
	10	170.88	23.07	205.44	27.68	239.13	32.28
	12.5	144.62	19.53	173.53	23.43	202.44	27.33
	15	125.56	16.96	150.66	20.35	175.76	23.73
	17.5	111.13	15.01	133.35	18.01	155.56	21.01
	20	99.84	13.49	119.80	16.18	139.76	18.88

Table 4.11 shows the input of the program. The output of the program is given in Table 4.12.

Table 4.11 The inputs of the conical shrink-fit program

Parameter	Value	Parameter	Value	Parameter	Value
p_o (Pa)	0	**E_1 (Pa)**	207.E9	n_1	0,292
p_i (Pa)	3000.E5	**E_2 (Pa)**	207.E9	n_2	0,292
R (m)	0,0025	**S_{ut1} (Pa)**	1760.E6	m	0,06
t_1 (m)	0.0075	**S_{ut2} (Pa)**	1720.E6	a	10
t_2 (m)	0.001	**S_{y1} (Pa)**	1620.E6	**F_{ax} (N)**	20.E3
h (m)	0.002	**S_{y2} (Pa)**	1570.E6	--	--

Table 4.12 The outputs of the conical shrink-fit program

Parameter	Value (mm)	Parameter	Value (Mpa)
a	2,50353	**p**	136,72
b	10,0035	**pi**	300
c	9,98506	**S r11**	0
d	19,9851	**S r21**	-136,72
R	2,5	**S r12**	-300
G	9,99797	**S r22**	-14,2907
M	19,9938	**S t11**	-291,677
d_T	0,0184694	**S t21**	227,892
d_1	-0,00352805	**S t12**	309,53
d_2	-0,00556082	**S t22**	23,8204
d_3	0,0129086	**S r1**	-300
d_4	0,00878234	**S r2**	-151,011
t_1	7,5	**S t1**	17,8525
t_2	10	**S t2**	251,713
h	20	**S l1**	20,0087
r_m	10,0035	**S l2**	6,25429
r_{cy}	16,4805	**S 1max**	318,936
r_{co}	12,954	**S 2min**	319,037
r_o	19,9851	**S 2max**	351,545
Parameter	**Value**	**Parameter**	**Value**
a	10	**FS1**	5,12247
--	--	**FS2**	4,5204

4.2.3.2 Effect of Design Parameters

The effect of adjustable design parameters such as ‘‘ F_{ax} ’’ axial force, ‘‘ a ’’ conical angle, ‘‘ h ’’ conicity length are studied in this section. Briefly, an answer given to question ‘‘How these factors do affect the interface pressure and total interface displacement?’’. This section investigates the effect of design parameters on interface pressure and total interface displacement.

Effect of Axial Force on Interface Pressure and Total Interface Displacement

The effect of the axial force was investigated in this section. While doing this, the parameters are set accordingly based on the optimum point which was found in the chapter 4.2.3.1. The program CoSF-Pro was run and the results were given which seen in the Table B1 in the Appendix B. The results was plotted showing the relationship between axial force, interface pressure ‘‘ p ’’ and total interface displacement ‘‘d_T’’ is given in Figure 4.12.

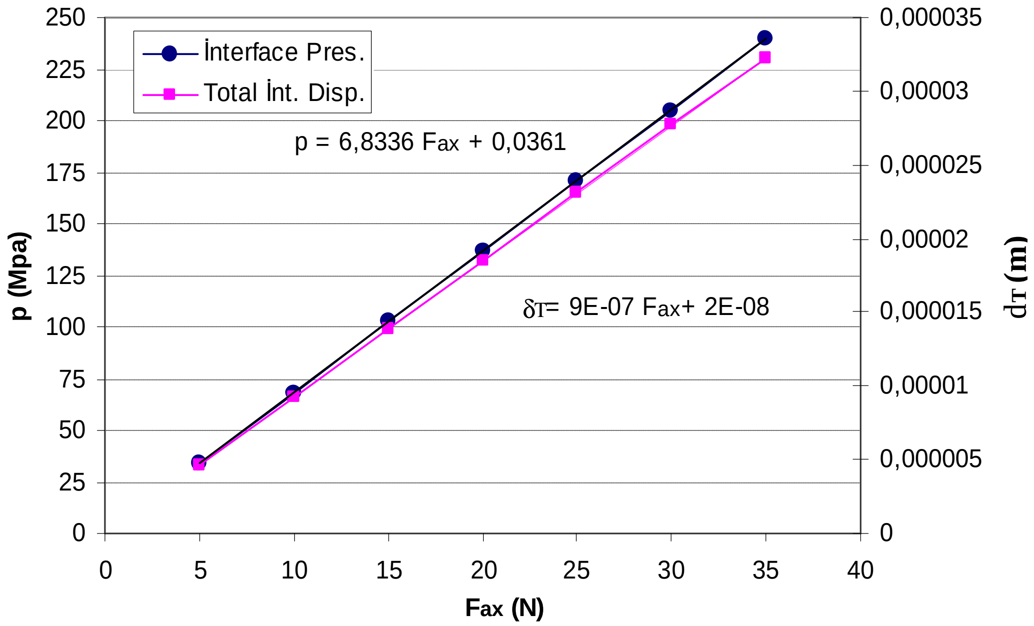

Figure 4.12 The relation between internal pressure, total interface displacement and axial force.

As seen in Figure 4.12 there is a linear relationship between interface pressure, total interface displacement and axial force. The left vertical axis represents the interface pressure values in MPa’s, the right vertical axis represents total interface displacement values in meter’s. The horizontal axis represents the axial force in Newton’s. The circular points are respresenting the coordinate points for internal pressure. The square points are representing the coordinate points for total interface displacement. The diagonal points are the calculated points. And the line connects the points is the regression lines and the formulas of these lines are given in the graph. As a result, interface pressure and total interface displacement is directly proportional with the axial force. It is clearly seen that increasing axial force causes to increase internal pressure and total interface displacement as expected.

Effect of Conical Angle on Interface Pressure and Total Interface Displacement

The effect of the conicity angle on the internal pressure and total interface displacement is investigated in this section. The conical angle is changed 2.5 degree increment and then the change of the interface pressure and total interface displacement is calculated by the developed computer program CoSF-Pro. The program was run and the results are given in Table B2 in Appendix B. Figure 4.13 was plotted to show the relationship between conical angle, interface pressure and total interface displacement.

As seen in Figure 4.13 the relationship between interface pressure, total interface displacement and conical angle is polynomial. The left vertical axis represents the interface pressure values in MPa’s, the right vertical axis represents total interface displacement values in meter’s. The horizontal axis represents the conical angle in degree’s. The diagonal points are the calculated points. The graph shows that increasing conical angle causes to decrease internal pressure and total interface displacement as expected.

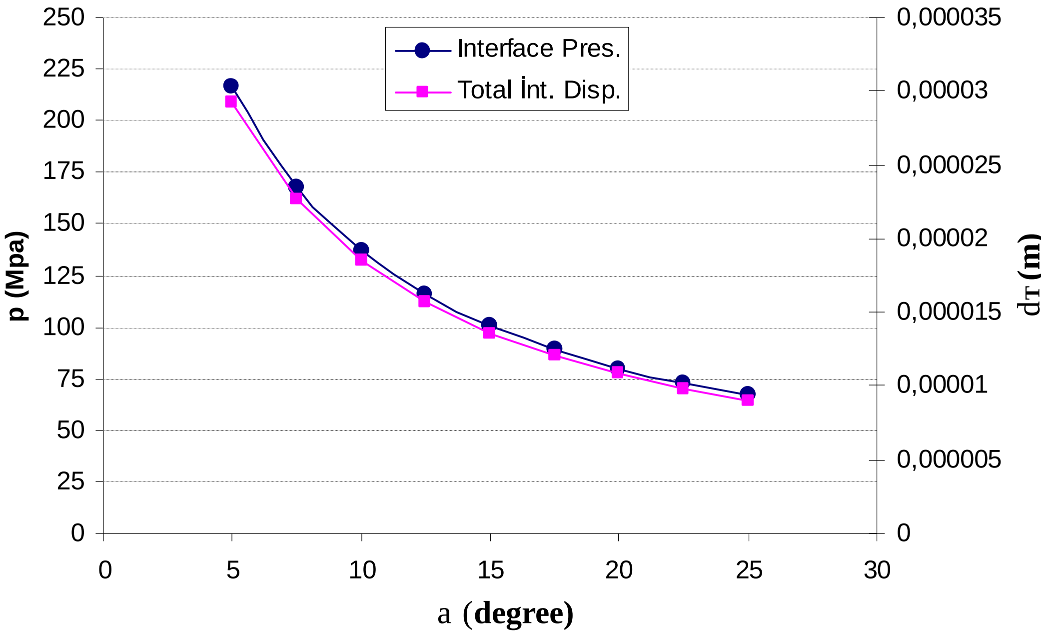

Figure 4.13 The relation between internal pressure, total interface displacement and conical angle

Effect of Conicity Length on Interface Pressure and Total Interface Displacement

In this section, the effect of the conicity length on the interface pressure and total interface displacement is investigated. The conicity length is changed 2.5 mm increments and then the change of the interface pressure and total interface displacement is calculated by the developed computer program CoSF-Pro. The results are given in Table B3 in Appendix B. Figure 4.14 was plotted to show the relationship between conicity length, interface pressure and total interface displacement.

As seen in Figure 4.14, the relationship between interface pressure, total interface displacement and conicity length is polynomial. The left vertical axis represents the interface pressure values in MPa's. The right vertical axis represents total interface displacement values in meter's. The horizontal axis represents the conicity length in milimeter's. The diagonal points are the calculated points. It is clearly seen that increasing conicity length causes to decrease internal pressure and total interface displacement.

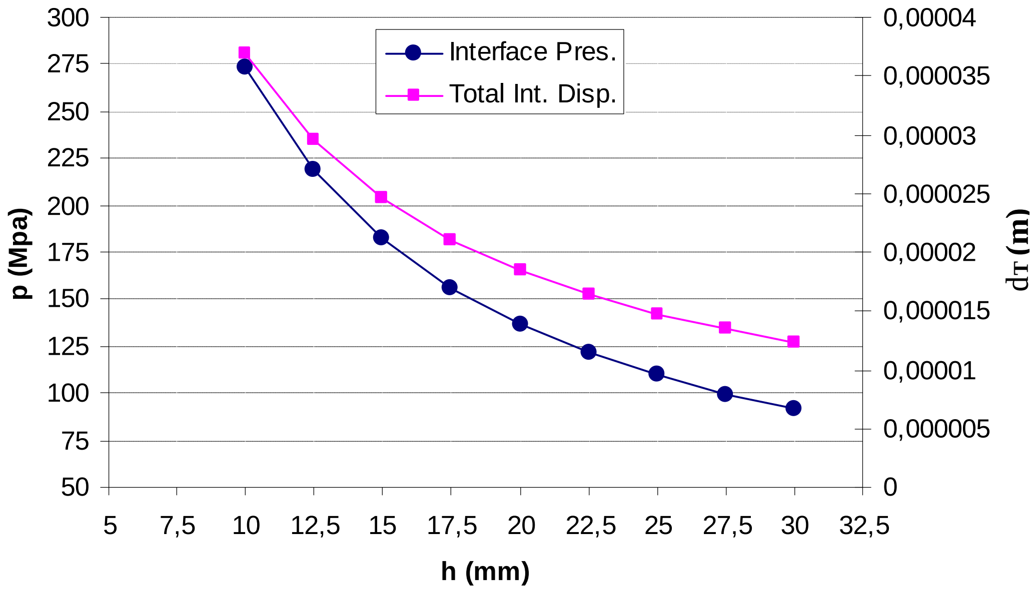

Figure 4.14 The relation between internal pressure, total interface displacement and conicity length

5. CONCLUSION

The use of high pressure water-jets as a cutting tool has recently increased in popularity. The popularity of the water-jet cutting comes from its various distinct advantages over the other cutting technologies such as no thermal distortion, high machining versatility, high flexibility, small cutting forces and ability to cut several materials without modifying their mechanical properties.

The first WJM system was designed for 1800 bar with single layer intensifier which is capable of cutting soft materials. After that a double layer system is designed for 3000 bar operating pressure constituting shrink fitted parts. In this study the objective was to redesign of high pressure pipes and connections for water-jet cutting system using reverse engineering and redesign methodology. The high pressure parts have been redesigned to eliminate the difficulties in maintenance, assembly-disassembly and ease of servicing and decrease total cost. In this study the head is redesigned based on the reverse engineering and redesign methodology developed by Otto and Wood. The sealing head is eleminated. Then, instead of high pressure hoses, high pressure pipes design is made considering standard dimension in Turkey. In the check valve and check valve block design conical form is chosen. The comparison of the new design and current design head is shown in table 5.1.

Table 5.1 The comparison of the new design and current design head

		Current Design	**New Design**
Weight of raw material of head and sealing head		46 kg+10 kg	36 kg
Weight of the head and sealing head		31,6kg+6,8 kg	23,8 kg
Number of parts to reach piston head seal		3 parts 2 connections	2 parts 2 connection
Total cost			
Head	Material	115 TL - $75.3	90 TL - $58.9
	Labor	90 TL - $58.9	110 TL - $72
Sealing Head	Material	110 TL - $72	-
	Labor	130 TL - $85.1	-

1 American dolar is 1.527 Turkish liras. Based on the table 5.1 the total cost of head and sealing head of current design is 445 TL – $291.3 and the total cost of head of the new design is 200 TL – $130.9. The advantage of the new design is aproximately % 124.

The comparison of the new design and current design check valve and check valve block is shown in table 5.2.

Table 5.2 The comparison of the new design and current design check valve and check valve block

	Current Design	**New Design**
Weight of raw material of check valve block	62 kg	40 kg
Weight of the check valve block	54,5 kg	31,9 kg
Total cost		
Material	155 TL - $101.5	100 TL - $65.5
Labor	100 TL - $65.5	100 TL - $65.5

Based on the table 5.2 the total cost of check valve block of current design is 255 TL – $167 and the total cost of check valve block of the new design is 200 TL – $131. The advantage of the new design is aproximately % 28.

Figure 5.1 shows exploded view of the left side of current design intensifier and high pressure parts and figure 5.2 exploded view of the left side of new design intensifier and high pressure parts.

A program which was written cylindrical shrink-fit parts in the Mathematica Programming language is adapted considering general strength formulas for conical shrink fitted parts. Then, to redesign check valve, this program is used. Then optimization of the conical angle, coniciy length and axial force were optimized by using the conical shrink-fit program. Beside these studies, the relationship between design parameters such as interface pressure and axial force, conical angle and conicity length is investigated using the conical shrink-fit program.

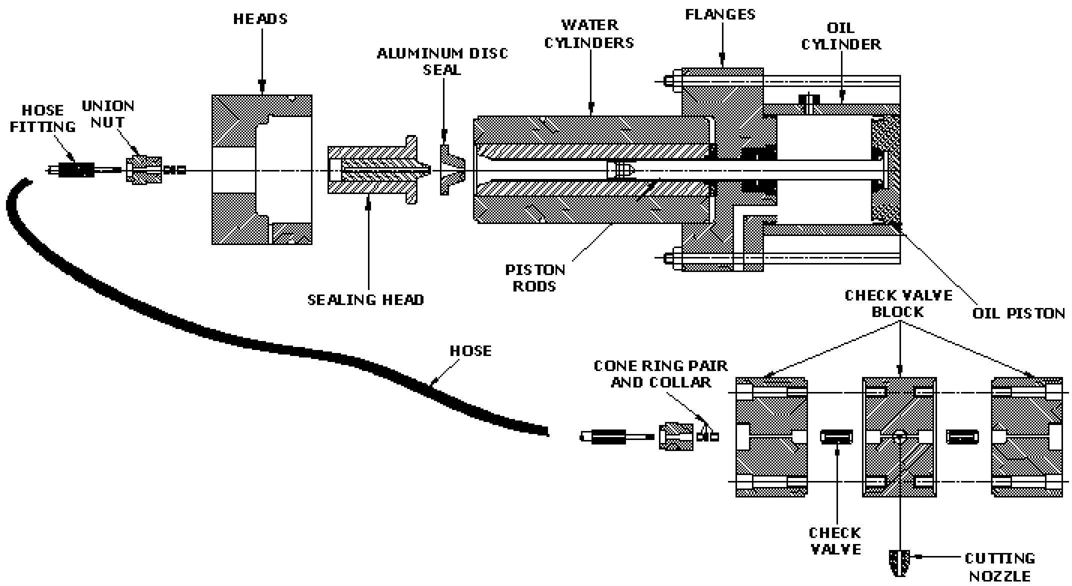

Figure 5.1 Exploded view of the left side of current design intensifier and high pressure parts

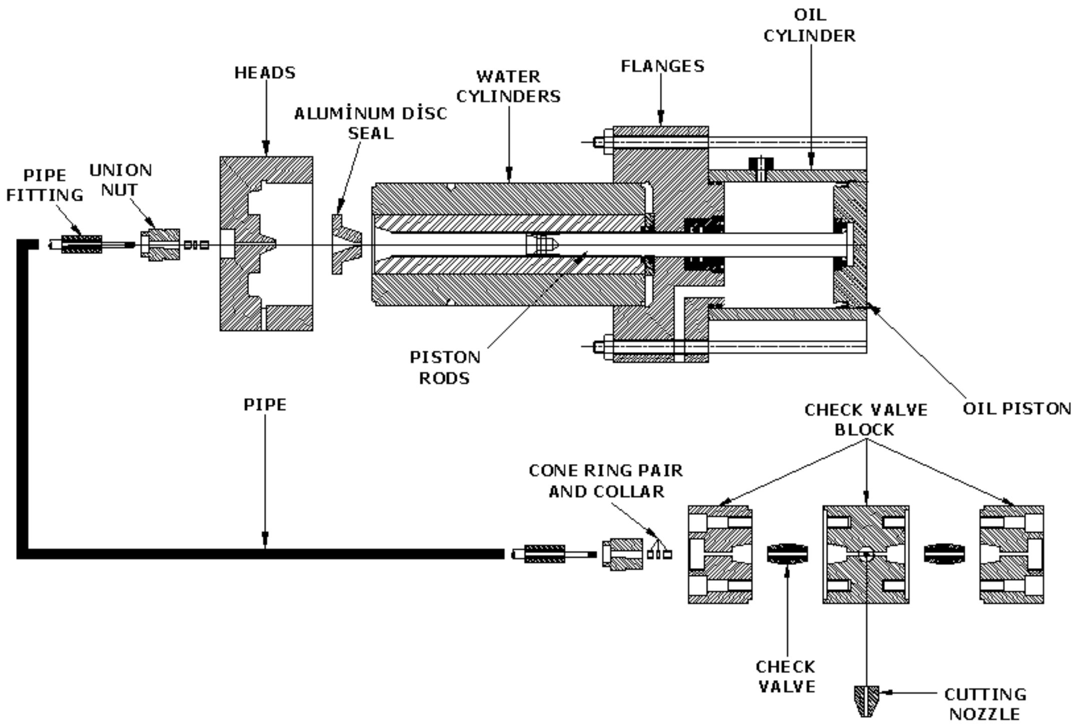

Figure 5.1 Exploded view of the left side of new design intensifier and high pressure parts

In future studies, the mass of some parts, especially heads and check valve block, should be decreased to increase the efficiency of system in great deal for easier maintenance, assembly and disassembly. The check valve block may be canceled and check valves might be mounted different places in the systems by searching new approaches.

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APPENDIX A

Steps of the Conical Shrink-Fit Design Program

Step1:

```
Off[General::spell1]
```

$po = 0;$

$pi = 3000 * 10^{5};$

$F_{ax} = 20 * 10^{3};$

$R = 2.5 * 10^{-3};$

$t1 = 7.5 * 10^{-3};$

$t2 = 10 * 10^{-3};$

$h = 20 * 10^{-3};$

$\alpha = 10;$

Step2:

$\mu = 0.06;$

$\nu 1 = 0.292;$

$\nu 2 = 0.292;$

$E1 = 207 * 10^{9};$

$E2 = 207 * 10^{9};$

$Sut1 = 1760 * 10^{6};$

$Sut2 = 1720 * 10^{6};$

$Sy1 = 1620 * 10^{6};$

$Sy2 = 1570 * 10^{6};$

$Sep1 = \left(\left(0.566 - 9.68 * 10^{-5} * (Sut1 / 10 \wedge 6)\right) * (Sut1 / 10 \wedge 6)\right) * 10 \wedge 6;$

$Sep2 = \left(\left(0.566 - 9.68 * 10^{-5} * (Sut2 / 10 \wedge 6)\right) * (Sut2 / 10 \wedge 6)\right) * 10 \wedge 6;$

Step3:

$X1 = \frac{E1}{1 - \nu 1^{2}}; \; X2 = 1 + \nu 1; \;\; X3 = 1 - \nu 1;$

$Y1 = \frac{E2}{1 - \nu 2^{2}}; \; Y2 = 1 + \nu 2; \; Y3 = 1 - \nu 2;$

Step4:

```
a = R - δ1;
b = a + t1;
c = b - δT;
d = c + t2;
rm = b;
ro = d;
rcy = 2 rm - h * Tan[(π / 180) * α];
rco = rcy - h * Tan[(π / 180) * α];
```

Step5:

$$F_N = \frac{F_{ax}}{(\mathrm{Sin}[(\pi/180)*\alpha] + \mu*\mathrm{Cos}[(\pi/180)*\alpha])};$$

$$p = \frac{F_N}{\pi*h*rm};$$

Step6:

$$C1 = \frac{b^2\ p\ X3}{(a^2 - b^2)\ E1};\ C2 = \frac{a^2\ b^2\ p\ X2}{(a^2 - b^2)\ E1};\ C3 = -\frac{c^2\ p\ Y3}{(c^2 - d^2)\ E2};\ C4 = -\frac{c^2\ d^2\ p\ Y2}{(c^2 - d^2)\ E2};$$

Step7:

$$\delta 2 = C1\ b + \frac{C2}{b};\ \delta 3 = C3\ c + \frac{C4}{c};\ \delta 4 = C3\ d + \frac{C4}{d};$$

Step8:

$$\mathrm{Solve}\left[\left\{\delta 1 == C1\ a + \frac{C2}{a},\ \delta T == C1*b + \frac{C2}{b} + C3*c + \frac{C4}{c}\right\},\ \{\delta 1,\ \delta T\}\right]$$

Step9:

$$\mathrm{FindRoot}\left[\left\{\delta 1 == C1\ a + \frac{C2}{a},\right.\right.$$
$$\left.\left.\delta T == \mathrm{Abs}\left[C1*b + \frac{C2}{b}\right] + \mathrm{Abs}\left[C3*c + \frac{C4}{c}\right]\right\},\ \{\delta 1,\ -0.0010\},\ \{\delta T,\ -0.0010\}\right]$$

Step10:

```
p = p /. %;
δ1 = δ1 /. %%;
δT = δT /. %%%;
```

Step11:

```
G = b + δ2;
M = d + δ4;
```

Step12:

```
ka = 1;
kb = 1;
kc = 1;
kd = 1;
ke = 1;
kf = 0.9;
Se1 = ka * kb * kc * kd * ke * kf * Sep1;
Se2 = ka * kb * kc * kd * ke * kf * Sep2;
```

Step13:

$$C1s = \frac{\nu 1 - 1}{E1}\ \frac{G^2\ p}{(G^2 - R^2)};$$

$$C2s = -\frac{\nu 1 + 1}{E1}\ \frac{G^2\ R^2\ p}{(G^2 - R^2)};$$

$$C3s = \frac{\nu 2 - 1}{E2}\ \frac{G^2\ p}{(G^2 - M^2)};$$

$$C4s = -\frac{\nu 2 + 1}{E2}\ \frac{G^2\ M^2\ p}{(G^2 - M^2)};$$

Step14:

$$C1ss = \left(M^2\ pi\ R^2\ Y2\ (X1\ X3 - Y1\ Y3) + G^2\ \left(pi\ R^2\ (X1\ X3 + Y1\ Y2)\ Y3 + M^2\ po\ X1\ X3\ (Y2 + Y3)\right)\right) / \left(X1\ \left(M^2\ (G - R)\ (G + R)\ X1\ X2\ X3\ Y2 + \left(G^2\ (G - R)\ (G + R)\ X1\ X2\ X3 - (G - M)\ (G + M)\ \left(R^2\ X2 + G^2\ X3\right)\ Y1\ Y2\right)\ Y3\right)\right);$$

$$C2ss = \left(G^2 R^2 \left(G^2 pi\ (X1\,X2 - Y1\,Y2)\ Y3 + M^2\ (po\ X1\,X2\ (Y2 + Y3) + pi\ Y2\ (X1\,X2 + Y1\,Y3))\right)\right) / \left(X1 \left(M^2\ (G - R)\ (G + R)\ X1\,X2\,X3\,Y2 + \left(G^2\ (G - R)\ (G + R)\ X1\,X2\,X3 - (G - M)\ (G + M) \left(R^2\,X2 + G^2\,X3\right) Y1\,Y2\right) Y3\right)\right);$$

$$C3ss = \left(M^2 po\ R^2\ X2\ (-X1\,X3 + Y1\,Y3) + G^2 \left(pi\ R^2\ (X2 + X3)\ Y1\,Y3 + M^2\ po\ X3\ (X1\,X2 + Y1\,Y3)\right)\right) / \left(M^2\ (G - R)\ (G + R)\ X1\,X2\,X3\,Y1\,Y2 - Y1 \left(-G^2\ (G - R)\ (G + R)\ X1\,X2\,X3 + (G - M)\ (G + M) \left(R^2\,X2 + G^2\,X3\right) Y1\,Y2\right) Y3\right);$$

$$C4ss = \left(G^2 M^2 \left(G^2 po\ X3\ (-X1\,X2 + Y1\,Y2) + R^2\ (pi\ (X2 + X3)\ Y1\,Y2 + po\ X2\ (X1\,X3 + Y1\,Y2))\right)\right) / \left(M^2\ (G - R)\ (G + R)\ X1\,X2\,X3\,Y1\,Y2 - Y1 \left(-G^2\ (G - R)\ (G + R)\ X1\,X2\,X3 + (G - M)\ (G + M) \left(R^2\,X2 + G^2\,X3\right) Y1\,Y2\right) Y3\right);$$

Step15:

$$\sigma r11 = X1 \left(C1s\ X2 - \frac{C2s\ X3}{R^2}\right);$$

$$\sigma t11 = X1 \left(C1s\ X2 + \frac{C2s\ X3}{R^2}\right);$$

$$\sigma r12 = X1 \left(C1ss\ X2 - \frac{C2ss\ X3}{R^2}\right);$$

$$\sigma t12 = X1 \left(C1ss\ X2 + \frac{C2ss\ X3}{R^2}\right);$$

$$\sigma l12 = pi\ \frac{R^2}{G^2 - R^2};$$

Step16:

$$\sigma r1 = \sigma r11 + \sigma r12;$$

$$\sigma t1 = \sigma t11 + \sigma t12;$$

$$\sigma l1 = \sigma l12;$$

Step17:

$$\sigma 1min = \sqrt{\sigma r11^2 + \sigma t11^2 - \sigma r11\ \sigma t11};$$

$$\sigma 1max = \sqrt{\left(\frac{(\sigma r1 - \sigma t1)^2 + (\sigma t1 - \sigma l1)^2 + (\sigma l1 - \sigma r1)^2}{2}\right)};$$

$$\sigma 1mean = \frac{\sigma 1max + \sigma 1min}{2};$$

$$\sigma 1ampl = \frac{\sigma 1max - \sigma 1min}{2};$$

Step18:

$$FS1 = \frac{1}{\frac{\sigma 1mean}{Sut1} + \frac{\sigma 1ampl}{Se1}};$$

Step19:

$$\sigma r21 = Y1\left(C3s\ Y2 - \frac{C4s\ Y3}{G^2}\right);$$

$$\sigma t21 = Y1\left(C3s\ Y2 + \frac{C4s\ Y3}{G^2}\right);$$

$$\sigma r22 = Y1\left(C3ss\ Y2 - \frac{C4ss\ Y3}{G^2}\right);$$

$$\sigma t22 = Y1\left(C3ss\ Y2 + \frac{C4ss\ Y3}{G^2}\right);$$

$$\sigma l22 = pi\ \frac{R^2}{M^2 - G^2};$$

Step20:

$$\sigma r2 = \sigma r21 + \sigma r22;$$

$$\sigma t2 = \sigma t21 + \sigma t22;$$

$$\sigma l2 = \sigma l22;$$

Step21:

$$\sigma 2min = \sqrt{\sigma r21^2 + \sigma t21^2 - \sigma r21\ \sigma t21};$$

$$\sigma 2max = \sqrt{\left(\frac{(\sigma r2 - \sigma t2)^2 + (\sigma t2 - \sigma l2)^2 + (\sigma l2 - \sigma r2)^2}{2}\right)};$$

$$\sigma 2mean = \frac{\sigma 2max + \sigma 2min}{2};$$

$$\sigma 2ampl = \frac{\sigma 2max - \sigma 2min}{2};$$

Step22:

$$FS2 = \frac{1}{\frac{\sigma 2mean}{Sut2} + \frac{\sigma 2ampl}{Se2}};$$

Step23:

```
N[{TableForm[{
    {"δ1=", δ1, "                    ", "a=", a},
    {"δ2=", δ2, "                    ", "b=", b},
    {"δ3=", δ3, "                    ", "c=", c},
    {"δ4=", δ4, "                    ", "d=", d},
    {"                                 "},
    {"rcy=", rcy, "                    ", "ro=", ro},
    {"rm=", rm, "                    ", "h=", h},
    {"rco=", rco, "                    ", "α=", α},
    {"                                 "},
    {"δT=", δT, "                    ", "R=", R},
    {"t1=", t1, "                    ", "G=", G},
    {"t2=", t2, "                    ", "M=", M}, {},
    {"p=", p, "                    ", "pi=", pi}, {},
    {"σr11=", σr11, "                    ", "σr12=", σr12},
    {"σr21=", σr21, "                    ", "σr22=", σr22}, {},
    {"σt11=", σt11, "                    ", "σt12=", σt12},
    {"σt21=", σt21, "                    ", "σt22=", σt22}, {},
    {"σl12=", σl12, "                    ", "σl22=", σl22}, {},
    {"σr1=", σr1, "                    ", "σr2=", σr2},
    {"σt1=", σt1, "                    ", "σt2=", σt2},
    {"σl1=", σl1, "                    ", "σl2=", σl2}, {},
    {"σ1min=", σ1min, "                    ", "σ1max=", σ1max},
    {"σ2min=", σ2min, "                    ", "σ2max=", σ2max},
    {"FS1=", FS1, "                    ", "FS2=", FS2}}]}]
```

Step24:

```
ClearAll[δ1, δ2, δ3, δ4, a, b, c, d, R, G, M,
        rcy, rm, rco, ro, h, α, p, pi, po, δT, t1, t2,
        X1, X2, X3, Y1, Y2, Y3, C1s, C2s, C3s, C4s, C1ss, C2ss, C3ss, C4ss,
        σr11, σr12, σr21, σr22, σt11, σt12, σt21, σt22, σl12, σl22,
        σr1, σr2, σt1, σt2, σl1, σl2, σ1min, σ1max, σ2min, σ2max, FS1, FS2]
```

APPENDIX B

The Tables

Table B1. The data of the relationship in between axial force ‘‘ F_{ax} ’’, total interface displacement ‘‘d_T ’’ and interface pressure ‘‘ p ’’ for conical angle ‘‘ $a = 10^0$ ’’ and conicity lengh ‘‘ $h = 20mm$ ’’

F_{ax} (kN)	p (Mpa)	d_T (mm)	FS1	FS2
5	34,1891	$4{,}62309.10^{-3}$	2,13489	12,1609
10	68,3721	$9{,}24235.10^{-3}$	2,70372	7,78019
15	102,549	$13{,}8578.10^{-3}$	3,60004	5,71849
20	136,72	$18{,}4694.10^{-3}$	5,12247	4,5204
25	170,885	$23{,}0772.10^{-3}$	7,91642	3,73737
30	205,044	$27{,}6812.10^{-3}$	13,2614	3,1856
35	239,197	$32{,}2814.10^{-3}$	22,3214	2,77583

Table B2. The data of the relationship in between conical angle ‘‘ a ’’, total interface displacement ‘‘d_T ’’ and interface pressure ‘‘ p ’’ for axial force ‘‘ $F_{ax} = 20kN$ ’’ and conicity lengh ‘‘ $h = 20mm$ ’’

a (°)	p (Mpa)	d_T (mm)	FS1	FS2
5	216,523	$29{,}2277.10^{-3}$	15,9172	3,03501
7.5	167,448	$22{,}6138.10^{-3}$	7,54697	3,80366
10	136.72	$18{,}4694.10^{-3}$	5,12247	4,5204
12.5	115,707	$15{,}6339.10^{-3}$	4,08743	5,18895
15	100,459	$13{,}5756.10^{-3}$	3,53144	5,81272
17.5	88,9107	$12{,}0163.10^{-3}$	3,189	6,39482
20	79,8803	$10{,}7968.10^{-3}$	2,95854	6,93806
22.5	72,6406	$9{,}81896.10^{-3}$	2,79364	7,44504
25	66,7206	$9{,}01925.10^{-3}$	2,67027	7,91809

Table B3. The data of the relationship in between conicity lengh ‘‘ h ’’, total interface displacement ‘‘ d_T ’’ and interface pressure ‘‘ p ’’ for axial force ‘‘ $F_{ax} = 20kN$ ’’ and conical angle ‘‘ $a = 10^0$ ’’

h (mm)	**p (Mpa)**	d_T **(mm)**	**FS1**	**FS2**
10	273,343	$36,8778.10^{-3}$	30,0436	2,45949
12.5	218,706	$29,5217.10^{-3}$	16,4744	3,00798
15	182,272	$24,6123.10^{-3}$	9,33135	3,53337
17,5	156,244	$21,1029.10^{-3}$	6,49846	4,03707
20	136,72	$18,4694.10^{-3}$	5,122247	4,5204
22,5	121,534	$16,4203.10^{-3}$	4,33859	4,88455
25	109,384	$14,7804.10^{-3}$	3,84051	5,43063
27,5	99,4423	$13,4384.10^{-3}$	3,49887	5,85967
30	91,1574	$12,3197.10^{-3}$	3,25112	6,27261

APPENDIX C

The program for pipes

```
Off[General::spelll]
```

$$pi = 3000 * 10^5;$$

$$a = 6 * 10^{-3};$$
$$t = 3 * 10^{-3};$$
$$b = a + t;$$

$$Sut = 620 * 10^6;$$
$$Se' = 0.5 * Sut * 10^6;$$

$$ka = 1;$$
$$kb = 1.189 * (2 * b)^{-0.097};$$
$$kc = 1;$$
$$kd = 1;$$
$$ke = 1;$$
$$kf = 1;$$
$$Se = ka * kb * kc * kd * ke * kf * Se';$$

$$\sigma r = -pi;$$
$$\sigma t = pi * \frac{b^2 + a^2}{b^2 - a^2};$$
$$\sigma l = pi * \frac{a^2}{b^2 - a^2};$$

$$\sigma min = 0;$$
$$\sigma max = \sqrt{\frac{(\sigma r - \sigma t)^2 + (\sigma r - \sigma l)^2 + (\sigma t - \sigma l)^2}{2}};$$

$$\sigma mean = \frac{\sigma max + \sigma min}{2};$$
$$\sigma amp = \frac{\sigma max - \sigma min}{2};$$

$$FS = \frac{1}{\frac{\sigma mean}{Sut} + \frac{\sigma amp}{Se}};$$

```
N[TableForm[{{"a=", a, "     ", "t=", t, "     ", "b=", b},
            {"σr=", σr, "     ", "σt=", σt, "     ", "σl=", σl},
            {"σmin=", σmin, "     ", "σmax=", σmax, "     ", "FS=", FS}}]]

ClearAll[a, t, b, σr, σt, σl, σmin, σmax, FS]
```