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DESIGN AND ANALYSIS OF VANE SPRING STRUCTURES

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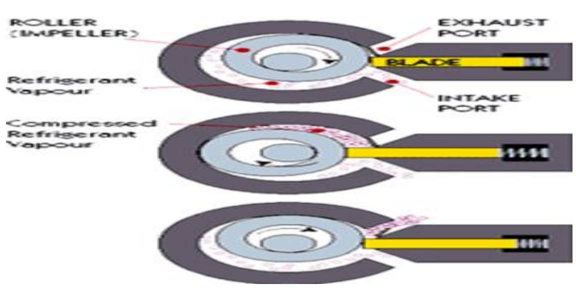
Abstract

In the rotary compressor there is a vane. The function of the vane is to maintain a proper contact with the roller. For that purpose a compressible helical spring is employed. The spring is to be analyzed, designed as per the limiting conditions.
This project deals with the design and analysis of the vane spring structures which are used in the rotary compressor by using ANSYS solver.
ANSYS was used for analyzing the vane spring. Emphasis was on finding a better solution and an alternate solution for the conditions prevailing for the system.
The design procedure starts with evaluation of the given spring parameters in static and fatigue conditions. The safe parameters are evaluated for a better solution. Then the model of the spring is made using CATIA software tools for testing in static and dynamic conditions. An alternate design is also worked out for the same design conditions there by providing an option for compressor design in future.
Index Terms: *vane spring, Catia, Ansys, static analysis, dynamic analysis .*

1. INTRODUCTION

1.1 ROTARY COMPRESSOR

In a rotary compressor the refrigerant is compressed by the rotating action of a roller inside a cylinder the roller rotates eccentrically (off-centre) around a shaft so that part of the roller is always in contact with the inside wall of the cylinder. A spring-mounted blade is always rubbing against the roller. The two points of contact create two sealed areas of continuously variable volume inside the cylinder.



Rotary compressor

At a certain point in the rotation of the roller, the intake port is exposed and a quantity of refrigerant is sucked into the cylinder, filling one of the sealed areas. As the roller continues to rotate the volume of the areas.

As the roller continues to rotate the volume of the area the refrigerant occupies it reduced and the refrigerant is compressed.

When the exhaust valve is exposed, the high-pressure refrigerant forces the exhaust valve to open and the refrigerant is released. Rotary compressor is very efficient because the action of talking in refrigerant and compressing refrigerant occur simultaneously.

1.2 SPRING

A spring is defined as an elastic body, whose function is to distort when loaded and to recover to its original shape when load is removal. The important application of springs is

- To cushion, absorb or control energy due to either shock (or) vibration as in car spring, railway buffers, aircrafts landing gears, shock absorbers and vibration dampers.
- To apply forces as in brakes, clutches and spring loaded valves.
- To control motion by maintain in contact between two elements as in cams and followers.
- To measure forces, as in spring balance and engine indicators.
- To store energy as in watcher, toys etc.

Advantages of using a helical compression spring:

- They are easy to manufacture.
- They are available in wide range.
- They are reliable.
- They have constant spring rate.
- Their performance can be predicted more accurately.
- Their characteristics can be varied by changing their dimensions.

Terms used in compression helical spring:

1) Solid length (lS):

When the compression spring is compressed until the coils come in contact with each other, then the spring is said to be solid. The solid length of a spring is the product of total number of coils and the diameter of the wire.

2) Free length (lf):

The free length of a compression spring is the length of the spring in the free or unloaded condition. It is equal to the solid length plus the max deflection or compression of the spring and the clearance between the adjacent coin.

3) Wire diameter (d):

The diameter of the wire that is wounded into a helix.

4) Coil diameter (D):

The mean diameter of the helix, i.e., (D outer +D inner/2).

5) Spring index (c):

The spring index is defined as the ratio of the mean diameter of the coil to the diameter of the wire.

6) Spring rate (k):

The spring rate (or) stiffness or spring constant is defined as the load required per unit deflection of the spring.

7) Pitch (P):

The pitch of the spring is defined as the axial distance between adjacent coils in uncompressed state.

8) Helix angle (α):

The angle between the coils and the base of the spring.

$\propto = \tan^{-1}(p/\pi)$

9) Total coils (nf):

The number of coils or turns in the spring.

10) Active coils (na):

The number of coils which actually deform when the spring is loaded, as opposed to the inactive turns at each and which are in contact with the spring seat or base.

1.3 SPRING MATERIALS

The selection of the spring material is usually the first step in parametric spring design. Material selection can be based on a number of factors, including temperature range, tensile strength, and elastic modulus, fatigue strength, corrosion resistance, electrical properties, cost etc.
The helical springs design requires the following material properties as input:
Elastic modules(E)
Poisson's ratio(η)
Material mass density (ρ).
A short description of common spring materials is given in the following paragraphs.

1.3.1 High Carbon Steels:

High carbon springs steels are the most commonly used of all springs materials. They are least expensive, readily available, easily worked, and most popular. These materials are not satisfactory for high or low temperatures or for shock or impact loading.

Examples include:
Music wire
Hard drawn
High tensile hard drawn
Oil tempered
Carbon valve

1.3.2 Alloy Steels:

Alloy spring steels have a definite plate in the field of spring materials, particularly for conditions involved high stress and for applications where stock or impact load occurs. Alloy springs steels also can withstand higher and lower temp then the high carbon steels.

Examples include:

- Chrome vanadium
- Chrome silicon

Stainless spring steels have seen increased use in recent years.
Several new compositions are now available to withstand corrosion. All of these materials can be used for high temperatures upto 650o F.

1.3.3 Copper-base alloys:

These are important spring materials because of their good electrical properties combined with their excellent resistance to corrosion. Although these material are more expensive then the high-carbon and the alloy steels. They nevertheless are frequently used in electrical components and in subzero temperatures. All copper-base alloys are nonmagnetic **Examples include:**

- Phosphor bronze
- Beryllium copper
- Monel 400

1.3.4 Nickel based alloys:

These are especially useful spring materials to corrosion and withstand both elevated and below zero temperature application.
There non-magnetic characteristics is important for such devices as Gyroscopes, Chronoscopes, and indicating instruments. These materials have high elastic resistance and should not be used for conductors of electrical current.

Examples include

- A206 Alloy
- Inconel 600
- Inconel718
- Inconel X-750

1.4 MUSIC WIRE

The material chosen was music wire. Music wire is high grade uniform steel originally intended for the strings of musical instruments but now employed for the manufacture of springs it is the highest grade of the string wire and is made of acid opening hearth steel or electric steel, free from slag or dirt and low in sulphur and phosphorus. They are cold drawn and posses a high tensile strength and uniform mechanical properties. They can be applied in applications requiring good fatigue properties.

The properties of this material are as follows:

1.4.1 Physical properties:

- Spring density (ρ) =0.284 lb/in^3=7850 kgm^{-3}
- Mechanical properties
- Hardness Rockwell C=41.60
- Tensile strength (τt) =67500 psi=2378692.2×$10^3$$pa$
- Endurance limit (τe) =67500 psi = 4657963
- Young's modulus (E)=30500×103psi
  = 210290180×10^3pa
- Modulus of rigidity(G) =11600×103 psi
  = 79979216×10^3pa
- Poisons ratio (μ) =0.313

1.5 PROBLEM DEFNITION

In the rotary compressor there is a vane. The function of the vane is to maintain a proper contact with the roller. For that purpose a compressible helical spring is employed. That spring is to be analyzed designed as per the limiting conditions.

Analyse the design of the helical compressible spring subjected to:

- Maximum load on the spring(W max) =6.421025 lb =28.562 N
- Preload on the spring (W min)=1.34413 lb =5.9785 N
- Compressor working frequency=50 Hz
- Maximum allowable deflection(δ)=0.7475in =18.68mm

The analysis includes static and dynamic analysis. Also find a suitable alternate design for the application. From the analysis the best parameters of the spring are found for the conditions prevailing in the application.

1.5.1 Parameters of the given spring:

Free length=1.2435in=30.86mm
Coil diameter D=11.27mm
Wire diameter d=0.041in=1.025mm
Hence c=D/d=11
End condition is squared and ground
Active coils=na=6
Total coils=nt=8
Pitch p=0.17764in=4.441mm

1.5.2 Material Properties:

Music wire with tensile strength
(τt)=345000Psi=2378692.2×10^3pa
Modulus of rigidity=G=11600×10^3Psi=79979216×10^3Pa
Endurance limit(τe)=67500 Psi=4653963×10^3Pa
Spring density(ρ)=0.284 lb/m^3 $= 7850$ kgm^{-3}

2. INTRODUCTION TO CATIA

CATIA is software which is used for creation and modifications of the objects. In CATIA and design and modeling feature is available. Design means the process of creating a new object or modifying the existing one. Drafting means the representation or idea of the object. Modeling means converting 2D to 3D.

2.1. SALIENT FEATURES OF CATIA

2.1.1 Parametric design and modeling:

Dimensions such as angle, distance, and diameter control CATIA geometry. It can create relationships that allow parameters to be automatically calculated based on the other parameters. To modify the dimensions, the entire model geometry can update according to the relations which are created.

When a feature is created in a solid model, dimensions are created. These dimensions do more than showing the size of the feature. They define the parameters that control the geometry. Since parameters control geometry, the geometry is said to be dimension driven.

Parameters can be driven by dimension values, or they can be driven by other parameters using relation. For instance, the length parameter of the feature is set so that it is always twice the width parameter. If the width changes the length will also change.

There are other kinds of values that can be used as parameters. A formula that relates specific feature geometry to volume, temperature, stress, weight and other properties can also be used in parametric designs.

When parameters change, other parameters driven by the modification also change. This is the essence of parametric design.

In CATIA the dimensions of an object are required to develop modeling or to create a solid object. All systems of units are available. Here with this feature the drawing of the object shapes in the required dimension is possible and also the conversion from one system of units is possible.

2.1.2 Associative:

CATIA is a fully-associative system. This means that a change in the design made in the development process propagated throughout the design, automatically, updating all engineering deliverables, including assemblies, drawing and manufacturing data. Associatively makes concurrent engineering possible by encountering change, without penalty, at any point in the development cycle. This enables downstream functions to contribute the knowledge and expertise early in the development cycle.

This is an excellent feature in CATIA. It means the modifications of an object after creation is possible without modification of the entire object. A list of operations will appear in the model tree, so the operations, which has been modified or redefined is selected.

2.1.3 Feature-based modeling:

These features have intelligence in that they contain knowledge of their environment and adapt predictably to change. Each feature asks the use for specific information based on feature type.

The most fundamental concept in understanding how to create a solid model is the concept of feature based design. In typical 2D CAD applications, a designer draws a part by adding basic geometric elements such as lines, arcs, circles and spines. Then dimensions are added. In solid modeling a 3D design is created by adding features, one at a time, until one has an accurate and complete representation of parts geometry.

2.2 Modeling Of Vane Spring Using CATIA:

Fig. 2.1 Selecting a Plane, Fig. 2.2 Creating a Point using Coil Diameter, Fig 2.3 Selecting Reference Axis to Generate Helix, Fig 2.4 Creating Helix By Giving Pitch and Free Length, Fig 2.5 Giving Coil Diameter, Fig 2.6 Generation of Solid Model along the Helix, Fig. 2.7 Drafting of Given Spring, Fig. 2.8 Drafting of Modified Spring, Fig. 2.9 Drafting of Alternate spring

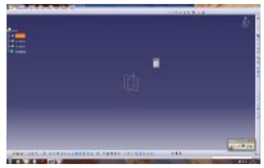



Fig. 2.1 **Fig. 2.2**

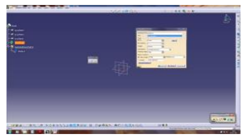
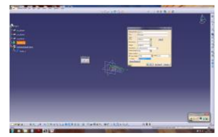



Fig. 2.3 **Fig. 2.4**


Fig. 2.5 **Fig. 2.6**

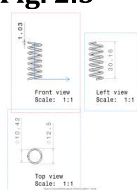
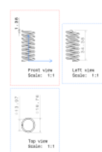



Fig. 2.7 **Fig. 2.8**

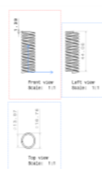


Fig. 2.9

3. INTRODUCTION TO ANSYS

ANSYS is general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user-designated size) called elements. The software implements equations that govern the behavior of these elements and solves them all; creating a comprehensive explanation of how the system acts as a whole. These results can be presented in tabulated or graphical forms. This type of analysis is typically used for the design and optimization of a system for too complex to analyze by hand. Systems that may fit into this category are too complex due to their geometry, scale, or governing equations.

ANSYS provides a cost-effective way to explore the performance of products or processes in a virtual environment. This type of product development is termed virtual prototyping.

With virtual prototyping techniques, users can iterate various scenarios to optimize the product long before the manufacturing is started. This enables a reduction in the level of risk, and in the cost of ineffective designs. The multifaceted nature of ANSYS also provides a means to ensure that users are able to see the effect of a design on the whole behavior of the product, be it electromagnetic, thermal, mechanical etc.

The ANSYS computer program is a large-scale multipurpose Finite Element program that may be used for solving several classes of engineering analysis. The analysis capabilities of ANSYS include the ability to solve static and dynamic structural analysis, steady state and transient heat transfer problems, more frequency and buckling Eigen value problems, static or time varying magnetic analysis, and various types of field and coupled field applications.

The programming contains many special features which allow nonlinearities or secondary effects to be included in the solution, such as plasticity, large strain, hyper elasticity, creep, swelling, large deflections, contact, stress stiffening, temperature dependency, material anisotropy, and radiation.

3.1 Types of analysis

1) Static analysis
2) Non linear analysis
3) Dynamic analysis
4) Buckling analysis
5) Thermal analysis
6) Fatigue analysis
7) Optimization
8) CFD analysis
9) Crash analysis
1 0) Vibration analysis (Modal analysis*).*

For this analysis I am using static analysis and Dynamic analysis.

4. ANALYTICAL SPRING DESIGN

4.1 ANALYTICAL SPRING DESIGN BASED ON STRENGTH

For spring material:

Ultimate shear stress of material (τs) = $\frac{1}{2}$(τs)
= $\frac{1}{2}$(345000)
= 172500 Psi
= 1189345633 Pa

Spring rate (K) = $\frac{\text{Wmax}}{\delta} = \frac{6.421025}{0.7475}$ = 8.59 lb/in
= 1.504 N/mm

Helical spring fails principally due to torsional shear. Hence the maximum torsional shear (τsmax) is calculated as
τsmax =Ks(8WD)/(πd3)

Where Ks – Wahl's Factor
W – Maximum Load
D – Coil Diameter
d – Wire Diameter
C – Spring Index
Ks = Direct Stress effect + Stress concentration effect due to curvature
= $\frac{4C-1}{4C-4} + \frac{0.615}{C}$

For the given spring
C = 11
d = 0.041 in = 1.041 mm
Hence, D = 0.451 in = 11.4554 mm
Ks = 1.1309 and
τsmax =Ks(8WD)/(πd3)
τsmax =Ks (8WC)/ (πd2)
τsmax = $\frac{1.1309*(8*6.421025*11)}{\pi*0.041^2}$
= 121002.3071 Psi
τsmax = 834281538.3 Pa < (τs = 1189345633 Pa)

Hence the other spring parameters are calculated as
Number of active coils (na) :
Deflection (δ) = (8WC3na)/(G.d)
0.7475= (8*6.421025*113*na)/(11600*103*0.041)
na = 5.19 = 6

Since squared and ground total number of turns (nt):
nt = na+2 =8

Free length (lf):

lf = (nt*d) + δ + 0.15δ
= 8*0.041 + 0.7475 +(0.15*0.7475)
= 1.18765 in
lf = 30.166 mm

Pitch of the coil (P):

P = lf/(nt-1)
= 1.18765/(8-1)
= 0.16966 in
P = 4.309 mm

Helix Angle (α):

α = Tan-1(P/πD) = Tan-1(0.16966/π*0.451)
α = 6.82 deg

4.2 ANALYSIS OF SPRING DESIGN UNDER FATIGUE CONDITIONS

Let's assume a factor of safety (F.S) = 1.25
Mean load = Wm = (Wmax + Wmin)/2
= (6.421025 + 1.344)/2
= 3.8825 lb
Wm = 17.27 N
Variable load = Wv = (Wmax - Wmin)/2
= (6.421025 - 1.344)/2
= 2.5385 lb
Wm = 11.29 N
Mean Shear Stress = τm = Ks (8wmD)/ (πd3)

$$= \frac{1.1309*(8*3.8825*11)}{\pi*d^2}$$

τm =123/d2
Variable Shear Stress = τv = K (8wvD)/ (πd3)
Where K = Shear stress factor = 1+1/2C

$$= \frac{1.048*(8*2.5385*11)}{\pi*d^2}$$

τv = 74.52/d2
From the design equation
1/FS = (τm – τv)/τs + 2*(τv /τe)
1/FS = (123/d2 – 74.2/d2)/172500 + 2*(74.2/d2/67500)
= (48.48/d2)/172500 + (149.04/d2)/67500
= (2.81*10-4/d2) + (2.208*10-3/d2)
= 2.489*10-3/(0.041)2
1/FS = 1.48
FS = 0.67 < 1.25 and it is unreasonable

4.3 MODIFIED SPRING PARAMETERS

The wire diameter for the factor of safety is calculated as
1/FS = (123/d2 – 74.2/d2)/172500 + 2*(74.2/d2/67500)
= (48.48/d2)/172500 + (149.04/d2)/67500
= (2.81*10-4/d2) + (2.208*10-3/d2)
1/FS = 2.489*10-3/(d)2
d2 = 2.489*10-3*1.25 =3.11125*10-3
d = 0.055 in = 1.397 mm (which is modified spring diameter)
By taking C = 11, the Modified Spring Parameters are
Coil Diameter (D) = 0.055*11 = 0.605 in =15.367 mm
Number of active coils (na):
Deflection (δ) = (8WC3na)/ (G.d)

0.7475 = (8*6.421025*113*na)/ (11600*103*0.055)

na = 6.99 = 7

Since squared and ground total number of turns (nt):

nt = na+2 =9

Free length (lf):

lf = (nt*d) + δ + 0.15δ

= 9*0.055 + 0.7475 + (0.15*0.7475)

= 1.354 in = 34.39 mm

Pitch of the coil (P):

P = lf/ (nt-1)

P = 1.354/ (9-1)= 0.1693 in = 4.30 mm

Helix Angle (α):

α = Tan-1(P/πD)

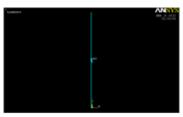
α = Tan-1(0.1693/π*0.605) = 5.09 deg

5. 2D ANALYSIS

5.1 STATIC ANALYSIS USING ANSYS

A spring model was made using the element combination 14 with two nodes as shown below

Fig 5.1 2-D Spring model, Fig 5.2 Constrained Static Condition



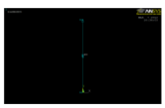


Fig 5.1 **Fig 5.2**

The maximum load that would be on the spring is applied at bottom node with direction upwards as shown below

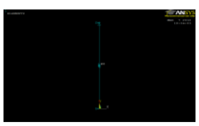


Fig 5.3 Loading Condition in Static Analysis

The model was solved and the deflections were found out as shown below

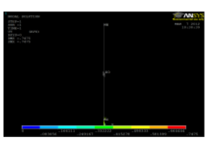


Fig 5.4 Resulting deflections in Static Analysis

The maximum deflection was 18.68 mm (0.7475 in) which is within limits.

5.2 DYNAMIC ANALYSIS

5.2.1 Steady Static Analysis (Harmonic Analysis):

Input Conditions:

Maximum Load = 6.421025 lb = 28.56 N

Operating frequency range = 48.5-50 Hz

Assumed Input Signal = Ramp Signal

Number of Sub-steps = 500

Amplitude Response of the Vane Spring at the node:

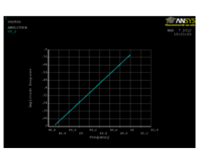


Fig 5.5

Resulting Amplitude in Steady State

The maximum amplitude i.e., spring displacement is almost equal to 18.68 mm (0.7475 in) which is the maximum displacement of design required. Hence the design is safe.

6. ALTERNATE SPRING DESIGN

We can use two springs instead of one. Let's consider two springs in parallel. Let the material wire diameter (d) and coil diameter (D) is the same with the same spring end conditions for this application.

6.1 ANALYTICAL SPRING DESIGN BASED ON STRENGTH

From the design equation,

W = W1 + W2

Where,

W = Maximum load on the Spring = 6.421025 lb = 28.56 N

W1 = Maximum load at spring1

W2 = Maximum load at spring2

Load on each spring i.e., W1 = W2 =W/2 = 6.421025/2

= 3.2105125 lb = 14.28 N

Let k1 = Spring constant of spring1

k2 = Spring constant of spring2

From the design equation δ1 = δ2 = δ =0.7475 in = 18.98 mm (Design Deflection)

i.e., δ = W/K = W1/K1 = W2/K2

0.7475 = 3.2105125/K1

K1 = 3.2105125/0.7475 = 4.2951 lb/in = 0.75218 N/mm

The maximum torsion shear (τsmax) is calculated as τsmax = Ks(8WD)/(πd3) = Ks(8WC)/(πd2)

= 1.1309*(8*6.421025*11)/(π*0.0552)

τsmax = 67241.28 Psi = 463612305.5 Pa

<(τs= 172500 Psi =1189345633 Pa)

Then,

Number of active coils (na):

Deflection (δ) = (8WC3na)/ (G*d)

0.7475 = (8*3.2105125*113*na)/(11600*103*0.055)

na = 13.95 = 14

Since squared and ground total number of turns (nt):

nt = na+2 = 16

Free length (lf):

lf = (nt*d) + δ + 0.15δ

= 16*0.055 + 0.7475 + (0.15*0.7475)

= 1.739625 in

lf = 44.186 mm

Pitch of the coil (P):

P = lf/ (nt-1)

= 1.739625/ (16-1)

= 0.115975 in

P = 2.94 mm

Helix Angle (α):

α = Tan-1(P/πD)

= Tan-1(0.115975/π*0.605)

α = 3.48 deg

6.2 ANALYTICAL SPRING DESIGN UNDER FATIGUE CONDITIONS

Let's assume a factor of safety (F.S) = 1.25

Mean load = Wm = (Wmax + Wmin)/2

Variable load = Wv = (Wmax - Wmin)/2

Mean Shear Stress = τm = Ks(8wmD)/(πd3)

$$= \frac{1.1309*(8*3.8825*11)}{\pi * d^2}$$

τm =123/d2
Variable Shear Stress = τv = K(8wvD)/(πd3)
Where K = Shear stress factor = 1+1/2C = 1.048

$$= \frac{1.048*(8*2.5385*11)}{\pi * d^2}$$

τv = 74.52/d2
From the design equation
1/FS = (τm – τv)/τs + 2*(τv /τe)
1/FS = (123/d2 – 74.2/d2)/172500 + 2*(74.2/d2/67500)
= (48.48/d2)/172500 + (149.04/d2)/67500
= (2.81*10-4/d2) + (2.208*10-3/d2)
= 2.489*10-3/(0.055)2
1/FS = 0.8228099174
FS = 1.22 which is nearer to 1.25

6.3 ANALYSIS OF ALTERNATE SPRING MODEL

6.3.1 Static Analysis using Ansys:

A spring model was made using the element combination 14 with two nodes, judging from the acting forces
The model was solved and the deflection were found out as shown below

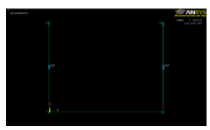


Fig 6.1 Constrained and Loading in Static Analysis

The model was solved and the deflection were found out as shown below

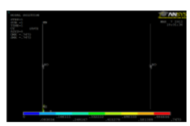


Fig 6.2 Deflection

It was found that the maximum deflection of the system as 18.68 mm (0.7475 in), which is design deflection.

6.3.2 Dynamic Analysis

Steady Static Analysis (Harmonic Analysis):

Input Conditions:
Maximum Load = 6.421025 lb = 28.56 N
Operating frequency range = 48.5-50 Hz
Assumed Input Signal = Ramp Signal
Number of Sub-steps = 500
Amplitude Response of vane spring at the nodal

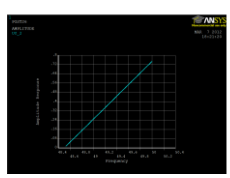


Fig 6.3 Amplitude Response in steady state

The maximum amplitude i.e., spring displacement of alternate spring is almost equal to 18.68 mm (0.7475 in) which is the maximum displacement of design required.

7. 3D ANALYSIS

7.1 GIVEN SPRING

A model of given spring from the spring parameters i.e., the wire diameter, coil diameter, free length in CATIA V5R18 version as show

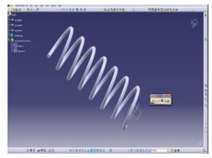


Fig 7.1 CATIA Model of Given Spring

The solid model was converted into iges format. From the Ansys workbench the iges format of the model was imported. After adding Brick 8 node 45 element and applying material properties i.e., the young's modulus, poission's ratio and density, the model was meshed. The meshed model was fully constrained on the first turn from the top and the other turns were horizontally constrained. The maximum load of 28.56 N (6.421025 lb) was applied at the bottom node.

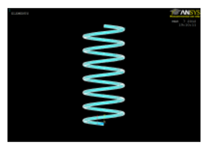


Fig 7.2 Loading Condition

Then software was run to solve, which produced the following results.

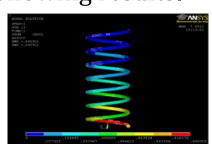


Fig 7.3 Deflection

The maximum deflection shown was 17.67 mm (0.695901 in). Hence from deflection point of view design is safe.

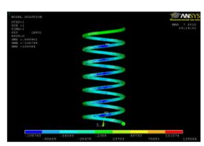


Fig 7.4 XY Shear Stress

Maximum shear stress in XY plane = 128064 Psi = 882970197.6 Pa

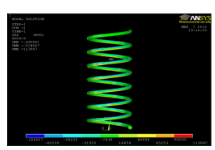


Fig 7.5 XZ Shear Stress

Maximum shear stress in XY plane = 113847 Psi = 784947433.2 Pa

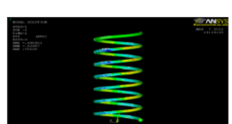


Fig 7.6 YZ Shear Stress

Maximum shear stress in YZ plane = 75100 Psi = 517796272.5 Pa

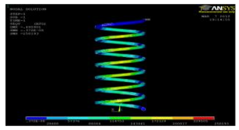


Fig 7.7 Von mises Stress

From the simulation it was found that
Maximum shear stress in XY plane = 128064 Psi = 882970197.6 Pa
Maximum shear stress in XY plane = 113847 Psi = 784947433.2 Pa
Maximum shear stress in YZ plane = 75100 Psi = 517796272.5 Pa
Von mises stress = 258193 Psi = 1780178069 Pa
Maximum shear allowable is 1189346100 Pa (172500 Psi). But the value of Vonmises stress calculated is much higher hence the spring can fail as the design was made on basis of maximum torsional shear.

7.2 MODIFIED SPRING

A model of modified spring from the spring parameters i.e., the wire diameter, coil diameter, free length in CATIA V5R18 version as shown below

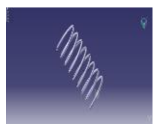


Fig 7.8 CATIA model of Modified Spring

This solid model was converted into iges format. From the Ansys workbench the iges format of the model was imported. After adding Brick 8 node 45 element and applying material properties i.e., the young's modulus, poission's ratio and density, the model was meshed. The meshed model was fully constrained on the first turn from the top and the other turns were horizontally constrained. The maximum load of 28.56 N (6.421025 lb) was applied at the bottom node.

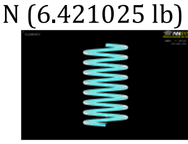


Fig 7.9 Loading Condition

The maximum load of 28.56 N(6.421025 lb) was applied at the bottom node.

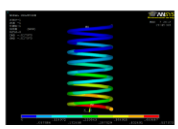


Fig 7.10 Deflection

The maximum deflection shown was 13.14 mm (0.517373 in). Hence from deflection point of view design is safe.

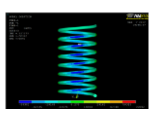


Fig 7.11 XY Shear Stress

Maximum shear stress in XY plane = 72892 Psi = 502572648.4 Pa

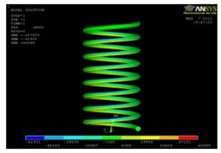


Fig 7.12 XZ Shear Stress

Maximum shear stress in XZ plane = 60899 Psi = 419883824.2 Pa

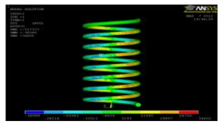


Fig 7.13 YZ Shear Stress

Maximum shear stress in XY plane = 34602 Psi = 238572391.7 Pa

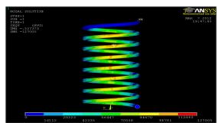


Fig 7.14 Vonmises Stress

From the simulation it was found that
Maximum shear stress in XY plane = 72892 Psi = 502572648.4 Pa
Maximum shear stress in XY plane = 60899 Psi = 419883824.2 Pa
Maximum shear stress in XY plane = 34602 Psi = 238572391.7 Pa
Von mises stress = 127005 Psi = 875668649.6 Pa
Maximum torsional shear allowable is 1189346100 Pa (172500 Psi). All the stress values from the simulation were found to be less than that. Hence the design is safe.

7.3 ALTERNATE DESIGN

A model of alternate spring from the spring parameters i.e., the wire diameter, coil diameter, free length in CATIA V5R18 version as shown below

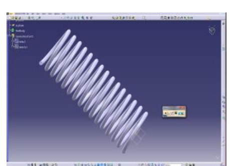


Fig 7.15 CATIA model of Alternate Spring

This solid model was converted into iges format. From the Ansys workbench the iges format of the model was imported. After adding Brick 8 node 45 element and applying material properties i.e., the young's modulus, poission's ratio and density, the model was meshed. The meshed model was fully constrained on the first turn from the top and the other turns were horizontally constrained. The maximum load of 14.28 N (3.2105125 lb) was applied at the bottom node.

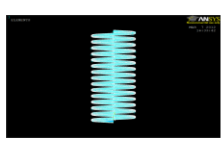


Fig 7.16 Meshed Model

Then software was run to solve, which produced the following results.

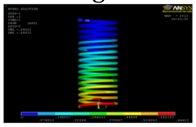


Fig 7.17 Deflection

The maximum deflection shown was 16.96 mm (0.66612 in). Hence from deflection point of view design is safe.

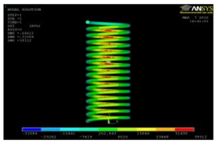


Fig 7.18 XY Shear Stress

Maximum shear stress in XY plane = 39312 Psi = 271046698.6 Pa

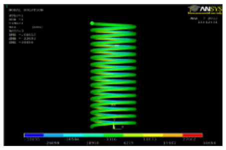


Fig 7.19 XZ Shear Stress

Maximum shear stress in XZ plane = 34656 Psi = 238944708.6 Pa

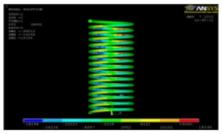


Fig 7.20 YZ Shear Stress

Maximum shear stress in XY plane = 18330 Psi = 126380901.1 Pa

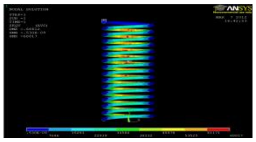


Fig 7.21 Vonmises Stress

From the simulation it was found that
Maximum shear stress in XY plane = 39312 Psi = 271046698.6 Pa
Maximum shear stress in XY plane = 34656 Psi = 238944708.6 Pa
Maximum shear stress in XY plane = 18330 Psi = 126380901.1 Pa
Von mises stress = 68817 Psi = 474476512.4 Pa
Maximum torsional shear allowable is 1189346100 Pa (172500 Psi). All the stress values from the simulation were found to be less than that. Hence the design is safe.

8. COMPARISION OF RESULTS

8.1 GIVEN SPRING

Shear Stress	D_{MX}(in)	S_{MN}(Psi)	S_{MX} (Psi)
Deflection	0.695901	-	0.695901
XY Plane	0.695901	-106749	128064
XZ Plane	0.695901	-104827	113847
YZ Plane	0.695901	-62497	75100
Vonmises stress	0.695901	-	258193

8.2 MODIFIED SPRING

Shear Stress	D_{MX}(in)	S_{MN}(Psi)	S_{MX} (Psi)
Deflection	0.517373	-	0.517373
XY Plane	0.517373	-58302	72892
XZ Plane	0.517373	-61931	60899
YZ Plane	0.517373	-36068	34602
Vonmises stress	0.517373	-	127005

8.3 ALTERNATE DESIGN

Shear Stress	D_{MX}(in)	S_{MN}(Psi)	S_{MX} (Psi)
Deflection	0.66612	-	0.66612
XY Plane	0.66612	-31084	39312
XZ Plane	0.66612	-33693	34656
YZ Plane	0.66612	-18296	18330
Vonmises Stress	0.66612	-	68817

9.CONCLUSION

- Analytical design of the spring was carried on the basis of static condition and fatigue conditions on given springs.
- The given spring was not suitable for fatigue conditions as per the factor of safety was found to be low. Hence the spring dimension was modified to improve the facto of safety to 1.25.
- An alternate design was made.
- 2d static and 2d dynamic analysis
    - Steady state and
    - Transient analysis was carried out.

- The amplitude response was cross checked with limiting values.
- 3d analyses were carried out and the results were compared.
- In static for “given spring”, it was found to have higher von mises stress compare to the torsional shear. Hence there is a possibility of failure of the “given spring”.
- All the results are compared.
- Modified spring is recommended for this application.
- Alternate design option can be recommended for this application depending on the requirement.

10.FUTURE SCOPE

- The optimization of the spring can be done by bringing the balance between the starting torque and the vane contact force required.
- There is also a scope for deep vibrational analysis.
- Alternate materials for the application can be worked out; there by a cost analysis can be made for finding the right material for the right cost for this application.
- By changing the limiting conditions extreme load cases can be worked out for evaluation of better design.

REFERENCES:

- [1] Harries ADH;Bissell KE;, Translating operational research into policy and practice, Abstract Book of the 2~(nd) Conference of the International Union Against Tuberculosis and Lung Disease(The Union),Asia Pacific Region, 2009
- [2] 马必生;, Research Scale of ICWM Perl-operational Period and Its Prospect, Abstracts of the 2nd World Integrative Medicine Congress, 2002
- [3] Knuuttila,K.;Alakoski,E.;J?msén,M.;, Ensure the Best Value of Woody Biomass with the Novel Operational Model, Abstracts of the 23rd European Biomass Conference and Exhibition, 2015
- [4] Susu Guo;Zijing Zhou;Jianming Li;Qiao Xiang;Zhonghua Li;, Applications of Soft Computing in RFID System:A Review, Proceedings of 27th Chinese Control and Decision Conference(CCDC), 2015
- [5] QINGPEI ZENG;CHUNXIA YU;XIANCHU ZENG;, A Risk Integrated Supply Chain Network Design Model, Proceedings of 2013 International Conference on Materials,Architecture and Engineering Technology(ICMAET 2013), 2013
- [6] MD REDZUAN Zoolfakar;ROSE Norman;EHSAN Mesbahi;, A Study of the Effect of Ship Size on Components Selection for an LNG carrier, Proceedings of 2012 International Conference on Mechanical Engineering and Materials(ICMEM 2012 (Part 3)), 2012
- [7] S.J.Fu;Y.M.Qin;, Curriculum development for logistic operational research: a case-based instruction, Proceedings of 2014 International Conference on Computer Science and Systems Engineering（CSSE2014）, 2014
- [8] Alfonsius Ariawan;, USING NON-PAPER-BASED SYSTEM AS AN EFFECTIVE GUIDE FOR OPERATIONAL RISK MANAGEMENT, Proceedings of 4th CCPS China Conference on Process Safety, 2016