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Thermostatic Bypass Valve for Temperature Regulation with Pressure Relief Feature

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ABSTRACT

A thermostatic bypass valve regulates the fluid temperature and also acts as the pressure relief valve. These valves are used to control the fluid temperatures in engines, compressors, radiators and hydraulic power packs. These study aims to design a reliable valve actuating mechanism for thermostatic bypass valve used in engine oil cooling system and hydraulic systems (Hydraulic Oil 68 Widely used in hydraulic power packs of CNC machines)). Pressure, Temperature rating, Body wall thickness was considered as per ASME B16.34-2020 CL150 and CL300. The mechanism of the valve was designed to overcome the cold flow starvation, overheating of the oil, which results in reduced efficiency and to reduce the pressure spikes in the flow circuit. As a result of study the temperature sensitive fluid flow control valve was designed with a bypass passage defined in the valve boy. The bypass passage was connected to heated oil source, oil cooler inlet and outlet and the oil return line.

Novelty and Area of Impact –

1. A temperature sensitive fluid flow control valve comprising of said Body with Seat housing at closing end provided with O-Ring, Thermostatic wax actuator including a valve element which extends to support the closing member Plug and the pressure relief functional actuated with compression spring (1).
2. A temperature sensitive fluid control valve where the thermostatic wax actuator and seat housing forms a resilient seat with bonded rubber.

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KEYWORDS: *Bypass Passage, Plug, Spring, Thermostatic Bypass Valve, Valve Element, Wax Actuator*

I. INTRODUCTION

The thermostatic bypass valve monitors the inlet flow and automatically diverts the flow of fluid based on its temperature. Fluid above the valves set point temperature is sent to heat exchanger for cooling and the fluid below the valves set point temperature bypasses the heat exchanger and enters into the reservoir or the return line. Thermostatic bypass valves are used to control the temperature of oil in hydraulic systems, lubricating systems, engine oil refrigeration units in Aerospace and Defense industries.

The main components of Thermostatic Bypass Valve are Body, Wax Actuator, Element, Seat Housing, Plug and Spring. The Wax actuators are thermally responsive flow control devices in fluid circulation systems. Wax actuators also known as Wax Motors are used as temperature sensitive actuators for valves used in fluid cooling systems to control the flow direction of the fluid during cooling process. The wax actuator is a device that converts the thermal energy into mechanical energy by the usage of phase change property of the Paraffin Wax. The operating principle is, as the wax temperature increases or decreases there is a significant change in the volume of paraffin wax as it goes through a phase change from solid state to liquid state and vice versa.

II. MARKET SURVEY (BACKGROUND)

There are several problems associated with the By-Pass valve used in the past.

1. The problem associated with the externally powered actuator valve like solenoid valve causes pressure spikes in the flow circuit due to sudden open and close of the valve.
2. Location of the actuator, it is either out of the flow path which results in delayed response time

of the valve and inefficient working of the cooling system, or over-exposed to the flow path in this the actuator is exposed to extreme low and high temperatures of the fluid and this can damage the valve seating and reduces the life span of the actuator

3. Externally powered solenoid valves have complex wiring of components like temperature sensors, electronic control unit, coil and power sources.
4. The Cold starvation problems at start of the startup conditions of the engine which reduces the heat exchange efficiency of the system.
5. Complex and inefficient operating mechanism of existing wax actuated bypass valve

III. DETAILED DESCRIPTION

Figures:

FIG. 1

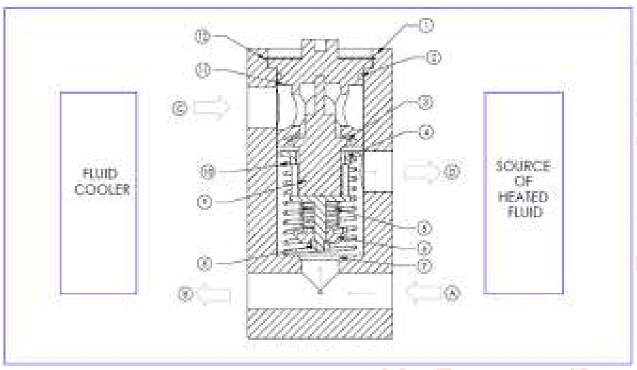
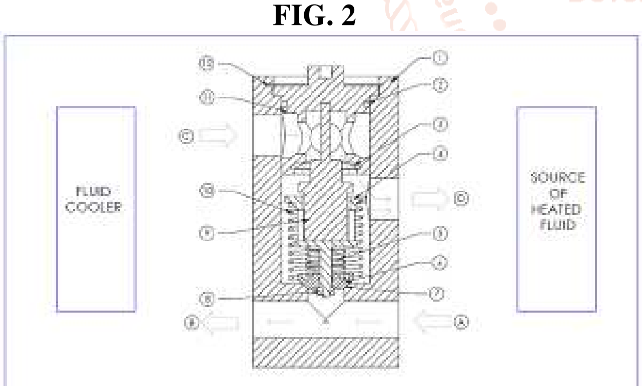


FIG. 2



Brief Description of Drawings

FIG. 1 is the sectional view of the valve in Bypass Position

FIG. 2 is the sectional view of the valve in Closed (Cooling) Position

Nomenclature

1. Body
2. O-Ring
3. Soft Seat
4. Valve Element
5. Compression Spring 1
6. Plug
7. Locking Nut
8. Seat
9. Thermal Element
10. Compression Spring 2
11. Seat Housing
12. Circlip

The Fig. 1 and 2 shows the cross-sectional view of the valve arranged between a source of heated fluid and the fluid cooler such as heat exchangers, radiators. The source of heated fluid could be oil from the combustion engines, hydraulic pumps, transmissions etc. The valve body has the fluid flow passages for receiving the heated fluid from heated source, transmitting it to cooler, receiving the cooled fluid from the cooler and then returning the cooled fluid to the system.

Referring to the Fig. 1 and 2 A, B, C and D are the four fluid flow openings. Opening **A** receives the fluid from the heated source, opening **B** is the connection between valve and the cooler input, opening **C** is the connection between the cooler output and the valve, which receives the cooled fluid from the cooler, opening **D** returns the cooled fluid back into the system.

The disclosed fluid flow control valve has two alternatively opening valves. The first valve is formed by the actuator (9) closing the flow passage by the soft seat (3). The valve seating is formed by the radially projecting surface of thermal element and the soft seat (As shown in Fig.1). The second valve is formed by the plug mounted on the valve element and the valve seat (as shown in Fig.3)

When the fluid temperature is below the predetermined set temperature, the fluid flow control valve is in bypass position as shown in Fig. 1. Due to low temperature the thermal actuator is not activated and it is in the home position, the radially projecting surface of thermal element is seated against the soft seat (3), this combination of valve prevents the fluid flow between fluid cooler and the fluid return line which connects to the heated source. And when the fluid control valve is in bypass position the plug (6) is separated from the valve seat (7) defined by the valve body (1). As a result of above combination of valves the fluid flow through the cooler is obstructed and fluid flows through the bypass passage and into the fluid return line which is connected to the heated source.

When the fluid gets warmed up in the heated source and the temperature of fluid exceeds the predetermined set temperature the thermal actuator starts expanding and exerts the force F and moves the actuator away from the home position to position shown in fig. 2. In this position the valve plug (6) is now in contact with valve seat (7) defined by the valve body (1) thus closing the bypass flow of the fluid. Apparently the thermal actuator has moved away from the soft seat (3) thus opening the passage between the fluid cooler and the fluid return line, which is input of cooled oil to the heated source.

When the bypass is closed and fluid starts flowing through the fluid cooler, the flow may get choked due to excess flow of the fluid and there may be chances of bursting the cooler due to increase in pressure. In order to prevent choking of flow and damage to the fluid cooler the valve is provided by special function of pressure relief. The coil spring (5) and the plug (6) acts as the pressure relief valve. By selecting the appropriate spring the pressure at which the plug is moved away from the valve seat (7) can be adjusted.

IV. CALCULATIONS

Consideration:

The available end connection sizes in markets are ½", 1" & 2" BSP. For calculation purpose we will select ½" BSP as the end connection, with a corresponding tap drill diameter of 19mm, leading us to choose a valve bore diameter of 19mm.

Body MOC: ASTM A276 SS316

1	Bore Diameter "d"	19	Mm
2	Body Pressure Rating (Class 150, ASME B16.34)	21 (2.1)	Bar (MPa)
3	Yield Strength of SS316 (From ASTM A276 SS316)	515	MPa

1. Body Wall Thickness Calculation

For Bore Diameter = 19 mm

Tm = 0.064*d+2.34 = **3.56 mm**

Considering manufacturing feasibility we can consider the wall thickness as **5mm.**

2. Compression Spring

Considerations for **Compression Spring 1.**

1. Max operating pressure of oil cooler = 7 Bar (Note: This may vary from manufacture to manufactures & on the size of oil cooler)
2. Spring Type: Square End
3. Crack Opening Pressure = 0.7 Mpa
4. Initial Deflection δ = 10 mm
5. Spring Index 'C' = 5
6. Bore Diameter 'd' = 19 mm
7. Spring Wire Diameter 'd1' = 2 mm
8. Material: SS316
9. Yield Strength (Ssy) = 1480 MPa
10. Modulus of rigidity = 69000 MPa
11. Assume the gap between adjacent coils = 1mm

Considerations for **Compression Spring 2**

1. Considering return force 'P' = 200 N
2. Spring Type: Square End
3. Initial Deflection δ = 5 mm
4. Spring Index 'C' = 5
5. Bore Diameter 'd' = 45 mm
6. Spring Wire Diameter 'd1' = 4 mm
7. Material: SS316
8. Yield Strength (Ssy) = 1240 MPa
9. Modulus of rigidity = 69000 MPa
10. Assume the gap between adjacent coils = 2 mm

Computing for Spring Dimensions

Eqn No.	Equation	Compression Spring 1 outcomes	Compression Spring 2 outcomes
A	Thrust due to cracking pressure $P = \frac{\pi}{4} d^2$x(Crack Opening Pressure)	= 198.47 N	Refer point 1 from Considerations for Compression Spring 2
B	Max Permissible Torsion Yield Strength T = $\frac{S_{SY}}{1.5}$	= 986.67 N	= 826.67 N
C	Wahl Factor K = $\frac{(4C-1)}{4C-4} + \frac{0.615}{C}$	= 1.31	= 1.31
D	Check for Actual Torsional stress for selected diameter T(Actual) = K $[\frac{8FC}{\pi d1^2}]$	= 827.90 N Comparing B & D As T(Actual) **<** **T**, selected diameter of spring is **Safe**	= 208.57 N Comparing A & D As T(Actual) **<** **T**, selected diameter of spring is **Safe**
E	Mean Coil Diameter D = C*d1	= 10 mm	= 20 mm
F	Determine the No. of active Coils N = $\frac{\delta G d1}{8FD^3}$	= 6.95 Nos	= 6.90 Nos

G	Total No. of Coils Nt = N + 2	= 8.95	= 8.90
H	Solid Length SL = Nt x d1	= 17.90 mm	= 35.60 mm
I	Total Gap = (Nt-1) x Gap Between adjacent coils	= 7.95 mm	= 15.80 mm
J	Free Length = Solid Length + Deflection + Total Gap	= 35.85 mm	= 56.40 mm
K	Pitch of the Coil $P = \frac{Free\ Length}{Nt-1}$	= 4.51 mm	= 7.1 mm
L	Spring Strength $K = \frac{Gd1^4}{8D^3N}$	= 19.85	= 40

3. Thrust Calculations for selection of Thermostatic Wax

Considerations

1. Valve orifice diameter 'D' = 19 mm
2. Outside diameter of seating face 'd' = 24 mm
3. Stem Diameter 'ds' = 6 mm
4. Design Pressure 'P' = 0.7 N/mm2
5. Width of the seating face 'w' = 3.54 mm
6. Coefficient of friction between packing and stem 'ms' = 0.16
7. Coefficient of friction between guide bush and stem' mb' = 0.16
8. Plug Length = 7 mm

Eqn No.	Equation	Results
A)	The force required to resist the line pressure up thrust on the valve member $FC = \frac{\pi}{4}\, d^2$x(Crack Opening Pressure)	198.47 N
B)	The frictional resistance imposed by the embracing effect of the guide bush on stem Fb = π X mb X ds X Lb X p	14.77 N
C)	Upward Spring Force	200 N
D)	Total Thrust = FC + Fb + Upward springForce	412.24 N
E)	Thrust considering the FOS of 1.25	515.30 N

4. Valve Flow Parameters

Considerations

1. Fluid = Hydraulic Oil 68 (Widely used hydraulic oil in power packs of CNC machines)
2. Density = 880 Kg/m3
3. Flow Velocity = 3 m/s
4. Dynamic Viscosity = 0.05984 Kg/m-s
5. Specific Gravity = 0.88

Eqn No.	Equation	Results
AA)	Volumetric Flow Rate Q = Flow Area x Velocity	= 13.48 gal (US)/min
BB)	Reynolds's Number $Re = \frac{Desnsity\ X\ Velocity\ X\ Bore\ Diameter}{Dynamic\ Viscosity}$	= 762 (Flow is laminar)
CC)	Flow co-efficient (Cv) $Cv = \frac{Q\ X \sqrt{SpecificGravity}}{\sqrt{\Delta P}}$	= 12.64 US Gallons/min

V. Conclusion

The design calculations for various functions and parts have provided a comprehensive overview of the valve's construction details, ensuring optimal performance and reliability. The basic construction details are tabulated blow.

Sr. No	Parameter	Dimensions & Units
1	Valve Size (End Connection)	½” BSP
2	Valve Bore Diameter	19.00 mm
3	Wall Thickness	5.00 mm
3	Compression Spring 1 wire Diameter	2.00 mm
4	Compression Spring 1 Total no. of coils	8.95 nos
5	Compression Spring 1 Mean Coil Diameter	10.00 mm
6	Compression Spring 1 Free Length	35.85 mm
7	Compression Spring 2 wire Diameter	4.00 mm
8	Compression Spring 2 Total no. of coils	8.90 nos
9	Compression Spring 2 Mean Coil Diameter	20.00 mm
10	Compression Spring 2 Free Length	5.40 mm
11	Flow Co-efficient	15.56 US Gallons/min
12	Thrust required to actuate the valve	515.30 Newton

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