Design Report: Multi-plate Clutch Date: 26/04/2018

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1 Overview

1.1 Introduction

This report details the design of a multi-plate clutch for use in motorcycles with a power output of 60 PS. Multi-plate clutches provide radially compact transmissions whilst still providing equivalent traction to larger diameter single-plate clutches. Further advantages to their use are: improved modulation at low speeds (if operated wet) and improved spring compatibility for hand-based actuation. The designed clutch is intended to operate wet and so, dynamic and lubricated frictional effects are predominantly considered.The design consists of drawings, a bill of materials and a general assembly (all produced in SolidWorks). Dynamic, stress, fatigue and thermal analyses are provided, along with the theoretic calculations that underpin the design. These provide justification for particular design features and further demonstrate the feasibility of the design.

1.2 Design Specification

This section outlines the required clutch components and also their dimensions, where necessary:

spring posts x5
centre splines x14
centre dogs x34
outer dogs x13
drive rivets x7
friction plates x7 (1.8mm thickness)
clutch discs x8 (1.2mm thickness)

Outer hub: 76 teeth, Crank pinion gear: 33 teeth
Pitch circle diameter: 150mm, Total primary reduction ratio = 33:76
Engine spec: 60PS = 44.8kW @ 8000rpm

2 Underpinning Theory & Design Configuration

Uniform Wear Theory: The key factor to clutch design is the transmittable torque. The uniform wear theory is used for calculation of this torque as it provides a more conservative estimate. The torque input to the clutch is calculated via the reduction ratio between the engine crankshaft and the clutch outer:

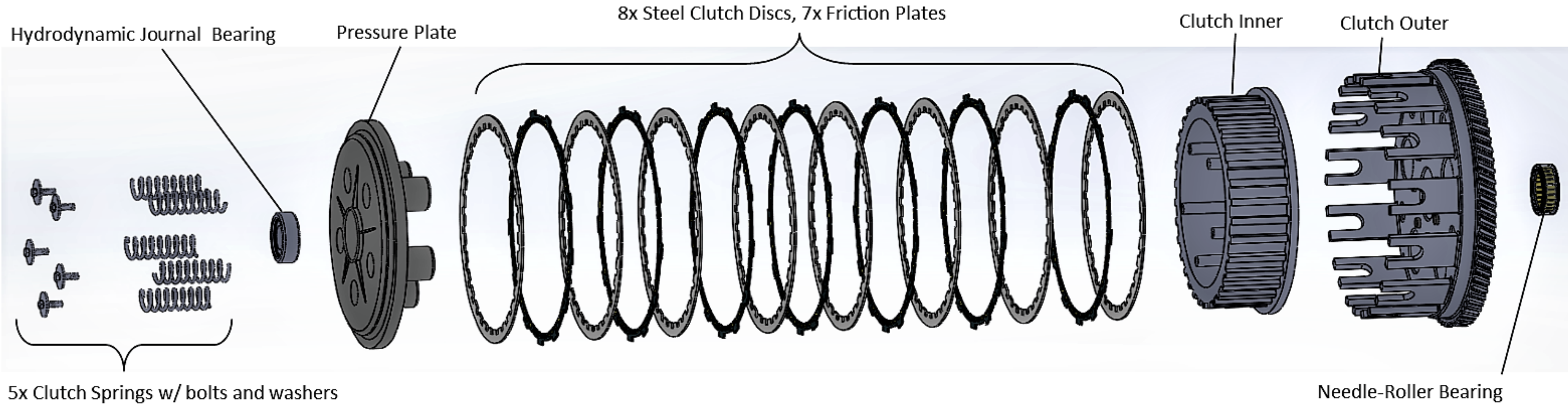
$T = \tau \times Primary\ Reduction\ Ratio\ (\frac{76}{33}) = n \times \mu \times F \times \frac{(R+r)}{2}$

Where the variables involved are as follows:

τ = Crank torque = 53.48 Nm,
T = Clutch Torque = 123.16 Nm,
n = No. contacting surfaces -1 = 14,
R = Outer plate radius = 67.5 mm,
r = Inner plate radius = 59 mm,

F = Axial clutch force $\approx$ Spring Force = ?
μ = Coefficient of friction (laminate phenol resin against steel under lubrication = 0.34)

Figure 1: Exploded Clutch Assembly



As a result of the above calculation, the necessary spring actuation force comes to 410 N, this falls comfortably under average grip strengths (50kg) demonstrating viable hand actuation. The required clutch springs can then be determined with Hooke's law:

$F = 5K \times (x_2 - x_1)$ Where F is as above, K is the desired (individual) spring constant and x_1 and x_2 are the extension limits of the spring in their respective pressure plate slots (43 mm and 28 mm). The required individual spring constant is therefore 5465 Nm^{-1}. Depending on the intended user, further consideration should be made of the handlebar lever point and hydraulic system to allow for the desired clutch lever sensitivity.

Table 1: Engineering Bill of Materials

Item No.	Part Name	Materials	Quantity
1	Clutch Outer Hub	Sand Cast 355 Al - T6: EN AC-45300	1
2	Friction Plate	C67500 Hot Worked Manganese Bronze	7
3	Friction Pads	Phenol Formaldehyde - Fibreglass laminate	7
4	Clutch Disc	YS500 Hot Forged Steel Plate: BS EN 10149-2	8
5	Clutch Inner	Al 7075 - T6 alloy	1
6	Pressure Plate	Sand Cast 355 Al - T6: EN AC-45300	1
7	Clutch Spring	Steel Helical Compression Spring	5
8	Spring Bolts & Washers	M4 Threaded Stainless Steel	5
9	Needle Roller Bearing	N/A	1
10	Hydrodynamic Bearing	N/A	1

Table 2: Machine and Forming Processes

Item No.	Part Name	Process/Specification
1	Clutch Outer Hub	Sand Cast, Gear Teeth - Precision Ground
2	Friction Plate	Hot Worked, CNC Milled
3	Friction Pads	Phenol resin bonded
4	Clutch Disc	Hot Forged
5	Clutch Inner	CNC Milled dog slots, Ground Splines
6	Pressure Plate	Sand Cast 355 Al - T6: EN AC-45300
7	Clutch Spring	K = 5465 Nm^{-1}
8	Spring Bolts & Washers	M4 Threaded
9	Needle Roller Bearing	N/A
10	Hydrodynamic Bearing	N/A

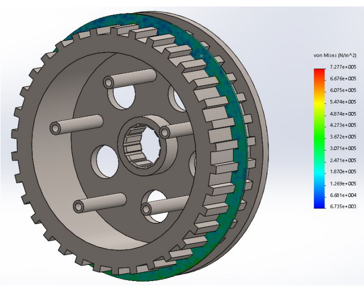
3 Engineering Drawings

4 Analysis

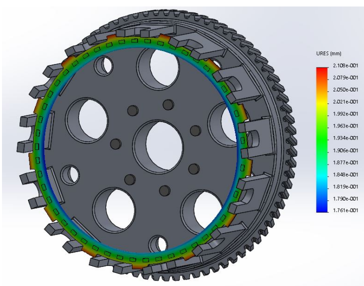
For evaluation of the effectiveness of the proposed clutch, Finite Element Analysis (FEA) was conducted on the assembly and key components. This took the form of stress, fatigue and thermal simulation studies. Further analysis was performed by means of motion studies and dynamic analysis. Design improvements are suggested as a result of these analyses, these are provided below the analysis plots themselves. Ultimately though, further knowledge of the wider vehicle is necessary for a conclusive working design to be implemented.

FE - Stress, Displacement, Thermal Analyses:
The following analyses were conducted so as to determine the robustness of the dogged parts, which experience the greatest impulses throughout the clutch operation

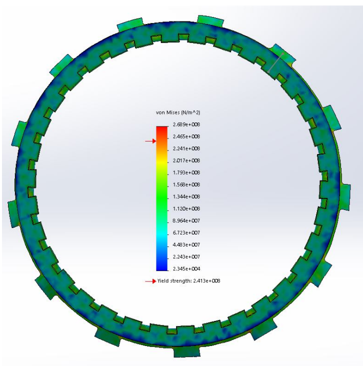
Dynamic Analysis:
A motion study was conducted of the complete clutch assembly for proof of effective rotation and transmission of torque, the following velocity plot indicates a clear transfer of momentum from the inner to outer once engaged.



(a) Clutch Inner Engagement Stresses



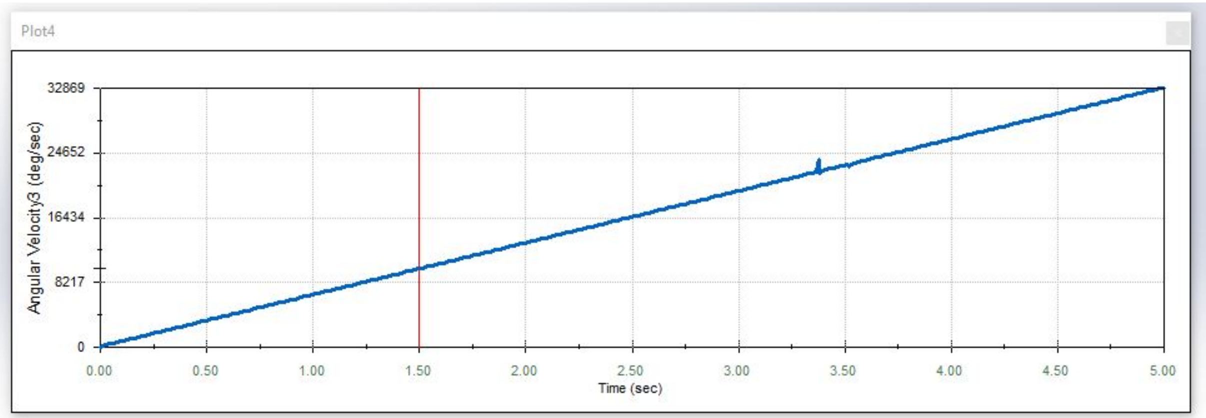
(b) Clutch Outer Engagement Stresses



(c) Combined Thermal-Stress

Figure 2: FEA Results

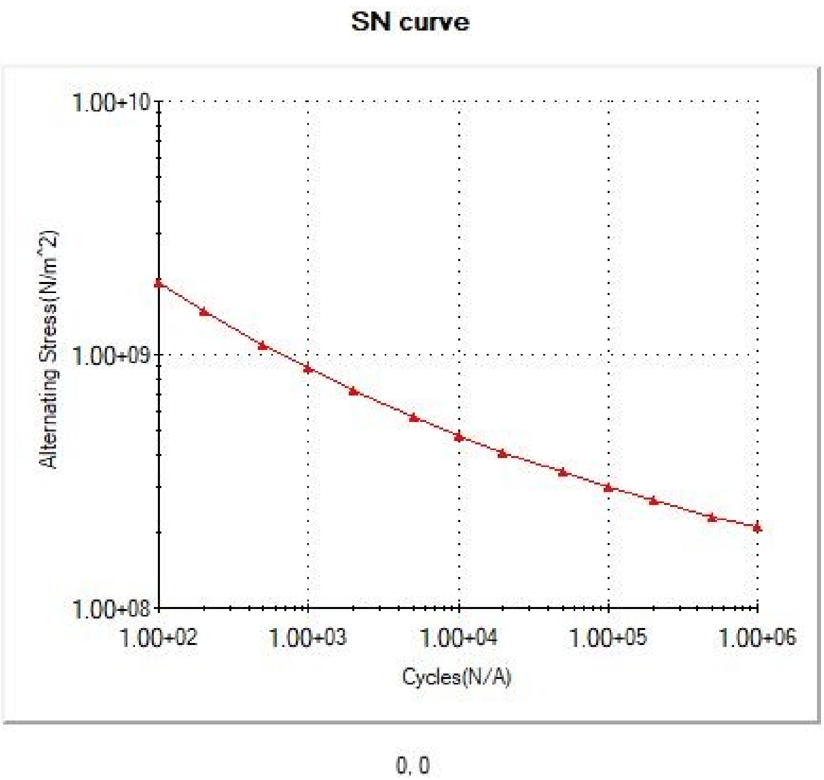
Figure 3: Angular Velocity of Inner w/ respect to Outer vs Time (Clutch Engaged)



Fatigue Analysis:

Fatigue Analysis was conducted through simulated cycling of the assembly engagement, the SN characteristic is displayed in Figure 4.

Figure 4: Clutch Assembly SN Characteristic



5 Design Improvements

The following design features have been implemented to improve the clutch operation by means not considered above:

1. **Helical reduction gear**: Being a key source of Noise, Vibration and Harshness (NVH), the reduction gear on the clutch basket possesses helical gears with a helix angle of 26 °. These are proven to provide reduced noise compared to the equivalent sized spur gears due to an increased overlap ratio and hence more gradual meshing.
2. **Tip relief**: In order to reduce the stresses on gear teeth, a fillet of radius 0.40mm is present on all the teeth that constitute the primary reduction gear. This is in part necessitated by the additional bending loads transmitted by the incline of the teeth on the helical gear, as above.
3. **Tapered roller bearing**: Present on the inner diameter of the clutch outer, the tapered roller bearing aims to accommodate both axial loads and the additional radial loads resulting from the incline of the helical gears.

The further improvements below may be implemented given further specification of the motorcycle's intended usage:

- Potential use of spur gears over helical for primary reduction (if cost, machine or acoustic constraints are present)
- Shot Peening of gears and dogs for up to 2x fatigue life
- Polishing of components for less scattered fatigue characteristics across components.