AiMT

Advances in Military Technology
Vol. 14, No. 1 (**2019**), pp. 59-69
ISSN 1802-2308, eISSN 2533-4123
DOI 10.3849/aimt.01254

Investigating T-finned Barrels for Machine Guns: Enhancement in Heat Dissipation and Flexural Rigidity along with Weight Reduction

E. Chaturvedi*

Maulana Azad National Institute of Technology, Bhopal, India

The manuscript was received on 2 Juny 2018 and was accepted after revision for publication on 3 March 2019.

Abstract:

This design innovation work is related to design and comparison of thermo-structural characteristics of a light-weight machine gun barrel. Compared to traditionally used thicker profile counter parts of machine gun barrels, these barrels were concluded to have lighter weight, better heat dissipation capability and better flexural rigidity. These barrels do not have rigidity problems like finned pencil barrels; neither have they extra weight addition, as in case of thicker profile barrels (H-Bars). To conduct the analysis, two same-length models of barrels bored for 5.56 × 45 mm, were modelled using Solidworks 15 software. The subsequent analysis using ANSYS 14.5 multi-physics solver simulated the condition of cook-off, which is an almost steady state reached after sustaining firing of 600 rounds (for 5.56 × 45 mm ammunition). It is a usual technical requirement expected from every durable machine gun, before the barrel is changed. The results showing maximum heat flux were transported to structural analysis workbench to measure the longitudinal deformation in both barrels under the gravity. The results concluded that T-finned barrels even after having less material, thus less weight, portrayed better heat dissipation characteristics and significantly less longitudinal deformation, thus better flexural rigidity and at the same time better accuracy retention than the conventional unfinned H-Bars.

Keywords:

machine gun barrel; high rate of fire; long duration of fire; heat dissipation; pseudo-I section; FEM analysis.

1. Introduction

A prominent difference between a rifle and its machine gun counterpart is their barrel profiles. Due to the suppressive high rate firing capability for long duration, the barrels of machine guns inherently acquire a thicker profile for greater heat dissipation. This thicker

* *Corresponding author: Department of mechanical engineering, Maulana Azad National Institute of Technology, Bhopal, India. Phone:+91-99 81 22 17 09, E-mail: ekanshchat96@gmail.com.*

profile indeed results in weight addition, which is the first problem that this research work intends to deal with.

Along the firing duration, the temperature of the bore surface continuously increases and after firing a specific amount of ammunition, the temperature reaches a value which is high enough to a limit called cook-off temperature. It is the temperature at which the chrome-line would begin to evaporate and the barrel walls get heated to such an extent that as the next round is chambered, the heat flows through the shell and auto ignites the propellant [1]. For the case considered here, the barrel is bored for 5.56×45 mm NATO ammunition and the corresponding average cook-off temperature comes to be around 1 000 K [2, 3]. Thus, the machine gun barrels are required to dissipate heat at a faster rate than their rifle counterparts. Hence, increasing the heat dissipation capability becomes the second concern of the barrel design.

The structural concerns related to barrels and hence the accuracy of rifles, being a dependent parameter, also come into play at higher temperatures. In most of the machine guns, the barrel is conjuncted to the trunnion of the receiver body and the rest of the length suspends as a cantilever. Indeed, there is a very slight bend in the barrel due to gravity which amplifies as the barrel attains higher temperatures. At higher temperatures, the inter-crystalline forces begin to weaken and the material begins to acquire fluidity and thus the flexural rigidity of barrel starts diminishing [4, 5]. Therefore, a slight increment in deformation leads to high accuracy loss and precision loss in the shot-group. Thus retaining high flexural rigidity in order to prevent accuracy loss becomes the third concern of the barrel design [6, 7].

Therefore, a major modification in the machine gun barrel is required in order to deal with all the three aforementioned problems simultaneously. Although the market launch of fluted-pencil barrels attempted to solve the problems of heat dissipation and weight, it was on the cost of rigidity; which hence was dismissed by the MIL-STDs. As a result, the pencil barrels have found usage in civilian arms market only. The proposed solution implements the concept of re-arrangement of material at supportive locations and removal of the extra material, rather than solely employing maximum material removal approach, as used in case of fluted-pencil barrels.

The solution proposes to reduce the base diameter of middle portion of a conventional H-Bar and to add even number of T-cross sectioned fins along that length. Incorporating the even number of fins would ensure that two fins are diametrically opposite to each other and this formation would formulate a pseudo-I section. Since the I section has the greatest section modulus per unit weight, and thus the flexural rigidity, it may be incorporated in a feasible manner to enhance the flexural rigidity of barrels. Moreover, the increment in surface area subjected to atmosphere due to fins would improve the heat dissipation capability of the barrel.

2. Methodology and Analysis

2.1. Mechanism of Heat Transfer

The simulation attempts to model a situation when, after firing almost 600 rounds in a sustained way, which resembles the situation with machine guns, the bore surface has reached sufficient temperature and can be assumed to be a heat conduction with convection off the outer surface. The mechanism of single dimensional steady state heat transfer at a radial position r in case of cylinders (barrel) is governed by Fourier's law of heat conduction for variable conductivity k of material with density ρ and heat capacity C [8] given as:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(rk\frac{\partial T}{\partial r}\right)+\dot{g}=\rho C\frac{\partial T}{\partial t}, \tag{1}$$

where g represents the heat generated within the cylinder. The Eq. (1) further reduces to a simpler form if conductivity of material is considered constant; independent of the temperature; and is given as:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)+\frac{\dot{g}}{k}=\frac{1}{\alpha}\frac{\partial T}{\partial t}, \tag{2}$$

where, α is the thermal diffusivity of the material.

For steady state condition, the term $\partial T/\partial t$ becomes 0. Thus the equation transforms to:

$$\frac{1}{r}\frac{d}{dr}\left(r\frac{dT}{dr}\right)+\frac{\dot{g}}{k}=0. \tag{3}$$

As further explained in the reference [8], the aforementioned equations converge to a much transformed equation specific to heat transfer in cylinders, which includes radiation and convective dissipations as well. The equation where T_∞ represents the ambient temperature T_2 represents the inner surface temperature, $R_{\text{conv},1}$ is the convective thermal resistance and $R_{\text{cyl}.1}$ is the thermal resistance of cylinder material, is given as:

$$\dot{Q}=\frac{T_\infty - T_2}{R_{\text{conv},1}+R_{\text{cyl}.1}}=\frac{T_\infty - T_2}{\dfrac{1}{h_1(2\pi r_1 L)}+\dfrac{\ln\frac{r_2}{r_1}}{2\pi L k_1}}. \tag{4}$$

2.2. Heat Transfer Mechanism Through Fins

The purpose of fins is to enhance convective heat transfer from surfaces. The primary mechanism behind the operation of fins is to increase the effective heat transfer area of a surface. They are commonly used in situations in which cooling is attained via free (or natural) convection for which the heat transfer coefficients h are relatively small. The mechanism is governed by the following equation, derived by substitution of laws of conduction and convection, taken from reference [9]:

$$-kA_c(x)\left.\frac{\mathrm{d}T}{\mathrm{d}x}\right|_x + kA_c(x+\Delta x)\left.\frac{\mathrm{d}T}{\mathrm{d}x}\right|_{x+\Delta x} - hdA_s(x)(T-T_\infty)=0. \tag{5}$$

To further simplify the problem, the fin of length L was assumed to have an adiabatic tip. The heat transfer is:

$$\dot{Q}=\sqrt{hPkA_c}\,(T_b - T_\infty)\tanh mL, \tag{6}$$

where, x is the distance of element from root of fin, A_c is the area of cross section, A_s is the surface area exposed to ambience and P is the perimeter. Here, m is defined as $\sqrt{hP/kA_c}$ [9].

The steady state thermal workbench solver utilises all the aforementioned equations as a basic algorithm to solve the problem concerned with heat transfer.

2.3. Algorithms Governing Structural Deformation of Materials

By simple shear deformation beam theory [10, 11], the displacement field defined by u and v for homogenous isotropic beams is given as follows:

$$u=-z\frac{\mathrm{d}a}{\mathrm{d}x}+\left(\frac{z}{4}-\frac{5z^3}{3h^2}\right)\frac{\mathrm{d}b}{\mathrm{d}x}, \tag{7}$$

$$v=a+b \tag{8}$$

where a and b denote the bending and shear displacements of a point on the mid-plane of the beam, and h is the thickness of the beam.

In the present displacement field, it can be seen that the in-plane displacement is neglected, since the stretching and bending deformations are uncouple in the cases of homogeneous isotropic beams. Furthermore, the quadratic distribution of transverse shear stress across the thickness of the beam is observed. The non-zero stresses are expressed by constitutive relations below:

$$\begin{aligned} \sigma_x &= E\varepsilon_x \\ \sigma_{xz} &= G\gamma_{xz} \end{aligned} \tag{9}$$

where E and G are Young's modulus and shear modulus, respectively, and the other parameters having usual meanings [10]. The linear strains are given by:

$$\varepsilon_{\mathrm{x}} = -z\frac{\mathrm{d}^2 a}{\mathrm{d}x^2} + \left(\frac{z}{4} - \frac{5z^3}{3h^2}\right)\frac{\mathrm{d}^2 b}{\mathrm{d}x^2}, \tag{10}$$

$$\gamma_{\mathrm{xz}} = \left(\frac{5}{4} - \frac{5z^2}{h^2}\right)\frac{\mathrm{d}b}{\mathrm{d}x}. \tag{11}$$

Rojacz et al. [5] investigated deformation mechanisms in solid bodies at elevated temperatures. The results indicated a very significant discovery that the deformation strongly depends on the microstructure, consequently their hardness. Ductile materials behaviour can be found for carbon steels, even at harder microstructures, such as martensites, but will change with elevated temperatures. Due to softening and microstructural changes at elevated temperatures, the impact behaviour of steels at a given chemical compositions is different. This explains the core purpose for conducting the analysis, which is to check the maximum sagging that would result in the conventional machine gun barrel and modified T-finned machine gun barrel under the influence of gravity when subjected to same firing conditions. This would validate the true significance of the structural modification suggested for barrel profiles.

2.4. Respective Geometries of Conventional and Modified Barrels

Fig. 1 shows the section view drawing of a conventional machine gun barrel chambered in 5.56×45 mm, which has been slightly modified from that of the archetypal reference [12]. The outer diameter of the mid- portion is 21.7 mm as shown.

Fig. 2 shown as follows presents the intricate drawings of a modified machine gun barrel with a reduced base diameter of 19 mm, of the mid-portion and addition of T-cross section fins over that surface, along the mid-length of the barrel. The drawing shown below hence further clarifies the concept of rearrangement of material, over the concept of removal of material.

2.5. Assumptions, Boundary Conditions and Mesh Specifications for Thermal Analysis

An inevitably important parameter is the selection and application of suitable material for both of the solid bodies. The material chosen for the thermo-structural FEM (Finite Element Modelling) analysis was AISI-4340 annealed steel whose material properties are shown in Tab. 1.

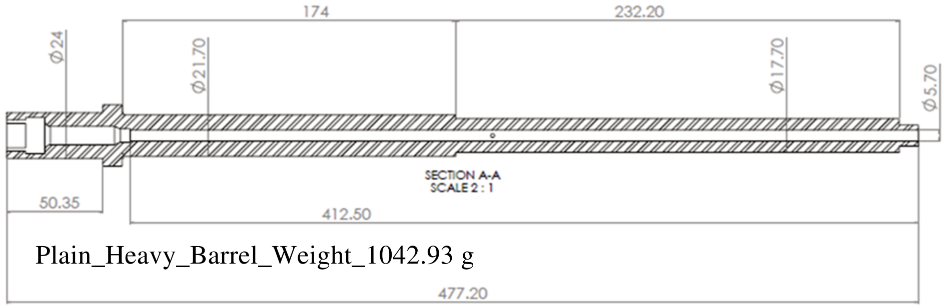

Fig. 1 Conventional machine gun barrel with thick mid-portion

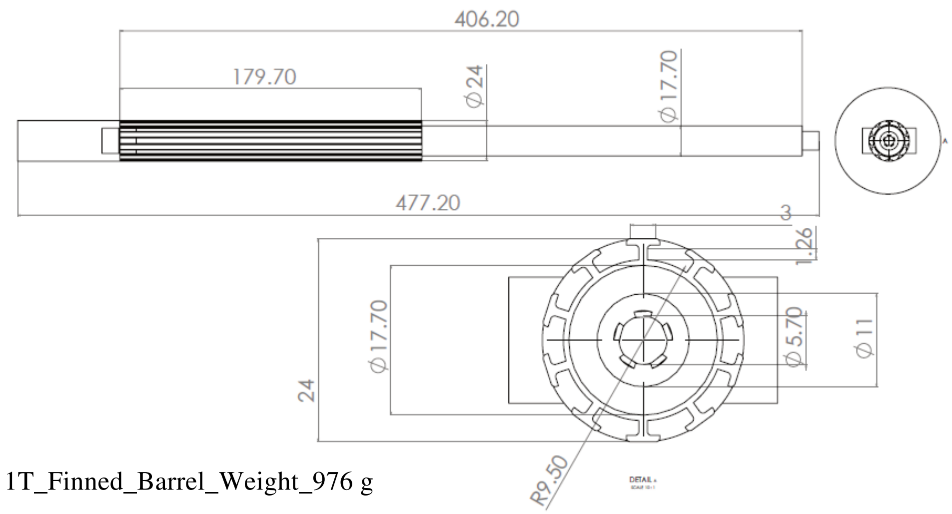

Fig. 2 Modified barrel with T-fins along the mid-length of the barrel

Tab. 1 Properties of AISI 4340 annealed steel

Properties	**Values**	**Units**
Elastic modulus	2.05×10^{11}	$[\mathrm{N\,m^{-2}}]$
Poisson's ratio	0.285	[1]
Shear modulus	8×10^{10}	$[\mathrm{N\,m^{-2}}]$
Mass density	7 850	$[\mathrm{kg\,m^{-3}}]$
Tensile strength	7.45×10^{8}	$[\mathrm{N\,m^{-2}}]$
Yield strength	4.70×10^{8}	$[\mathrm{N\,m^{-2}}]$
Thermal expansion coefficient	1.23×10^{-5}	$[\mathrm{K^{-1}}]$
Thermal conductivity	44.50	$[\mathrm{W\,m^{-1}K^{-1}}]$
Specific heat	475	$[\mathrm{J\,kg^{-1}K^{-1}}]$

The material has been assumed to be isotropic and homogenous in structural properties with a constant temperature independent thermal conductivity [13]. Another important point to be mentioned here is that the bore surface was subjected to a uniform temperature of 1 000 K along the length, instead of a gradient. It has been done to emulate a hypothetical condition where we assumed that; firstly, after prolonged high firing rate, the heat transfer process has almost come to a steady state, for it would be going to take a long time to come into a transient one; and secondly, there will not exist much of a difference in temperatures at the bore surface along the length of barrel [2, 3]. Although usually the preferred operating temperature limits lie between 623 K to 723 K (350 °C to 450 °C) on slow rate firing [14] before there arises a requirement of changing the barrel, but 1 000 K has been chosen for assessing the exaggerated effects over the end sag of the barrel under gravity after sustained firing.

Also, the heat dissipation to the atmosphere by both the barrels was considered to be purely free convective in nature with stagnant ambient air being at 300 K [15] having heat transfer coefficient values varying with film temperature (built-in files in ANSYS software package) shown as follows in Fig. 3.

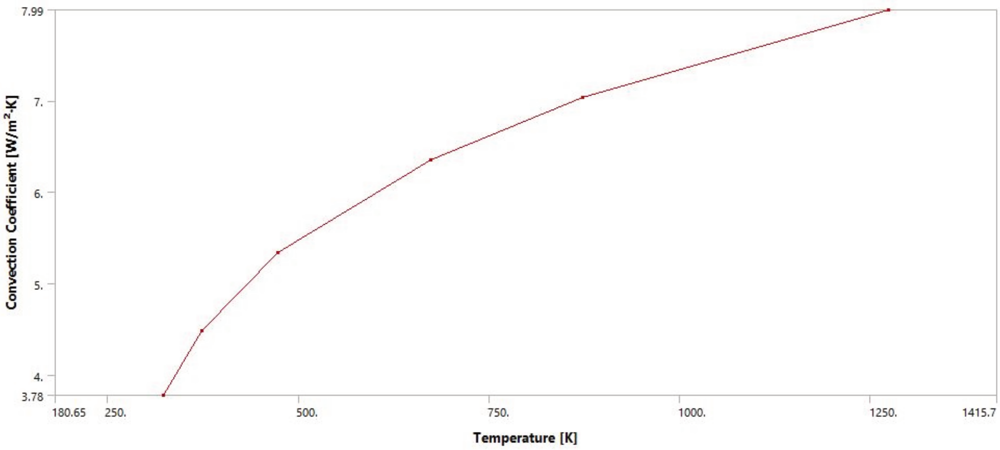

Fig. 3 Atmospheric heat transfer coefficient variation with film temperature

Fig. 4 represents mesh settings adopted for conducting the FEM analysis. The mesh hence generated had fine relevance centre and medium level smoothing with minimum edge length of 0.45 μm [15, 16]. Fig. 4 shows the meshing file of conventional barrel.

2.6. Boundary Conditions for Structural Analysis

The results obtained from the steady state thermal analysis were intrinsically exported to the static structural workbench, thus keeping all the initial mesh settings same and intact.

The protruding lugs at the sides of the breech end of barrel were subjected to fixed supports as shown in Fig. 5, thus emulating the kind of support provided by the front trunnion of a machine gun. Further, the gravitational influence was kept activated in vertical plane to treat the barrel body as a cantilever beam subjected to a uniformly distributed load of its own weight [17, 18]. All the aforementioned conditions and settings were kept absolutely same for both the barrels. The solver was run at automatically set maximum 1 000 iterations setting.

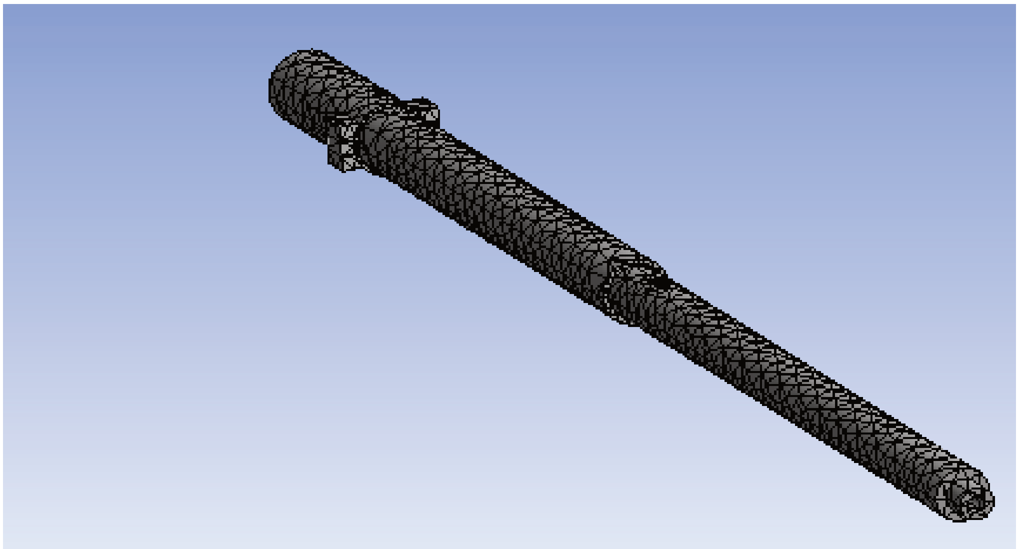

Fig. 4 Mesh representation of the conventional barrel model

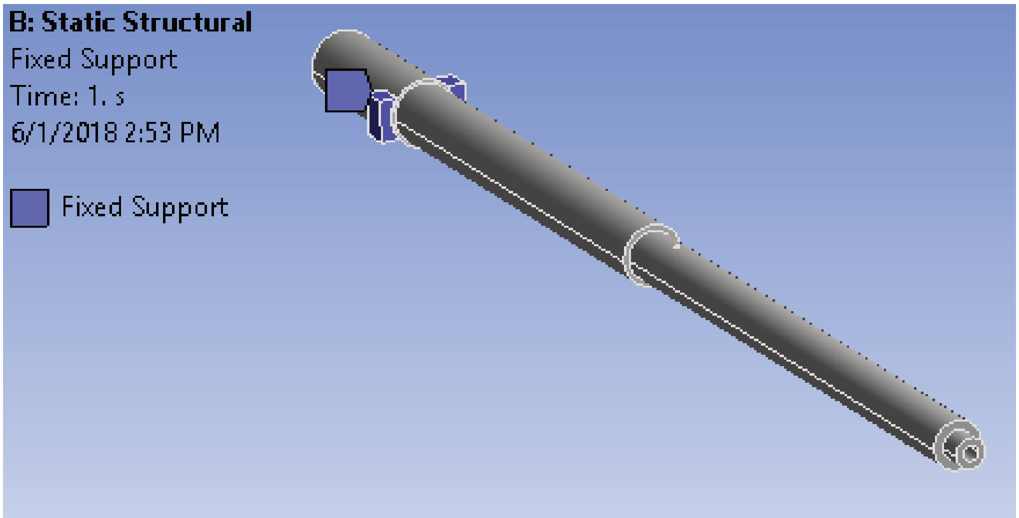

Fig. 5 Fixed supports applied at the protruding lugs

3. Results

Figs 6 and 7 show the total dissipated heat flux from the surfaces of plain H-bar and modified finned barrel respectively. A significant enhancement in heat dissipation can be observed by comparing the colour contours and the corresponding values from the shown scale in the respective figures.

Figs 8 and 9 show the results of total lateral deformation under the influence of gravity at elevated temperatures in respective cases of unfinned and finned barrels. The difference in directional deformation at the muzzle end can be used to find the quantitative value expressing the degree of improvement, which can measure the worth of design modification proposed through this study.

The percentage reduction in deformation is calculable as follows:

$$\frac{\text{difference in the muzzle end deformation}}{\text{deformation at the end of unfinned barrel}} \times 100$$

This proceeds to be:

$$\frac{7.7435 \times 10^{-5} - 5.3355 \times 10^{-6}}{7.7435 \times 10^{-5}} \times 100 = 93\ \%$$

Such a large observable difference expresses a significant improvement in the structural properties of the barrel and expresses the worth of this modified design, if realized.

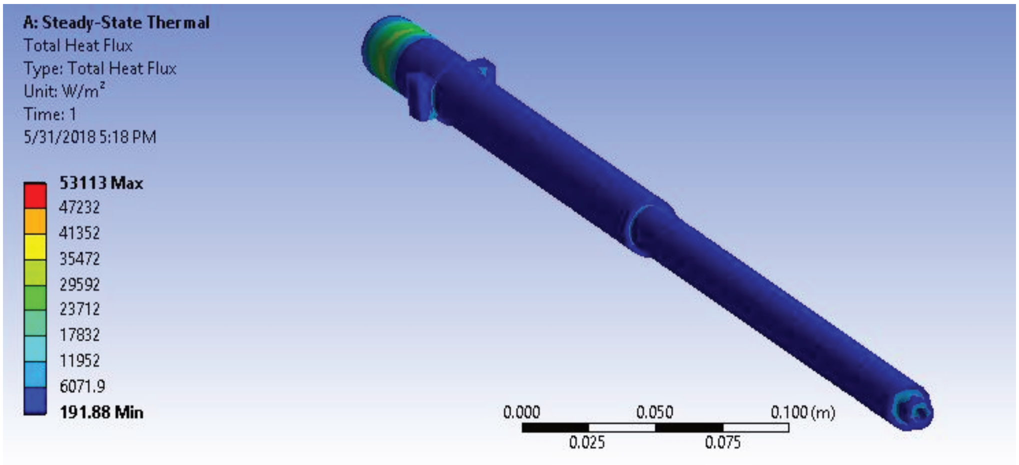

Fig. 6 Heat dissipation in plain thick profile barrel

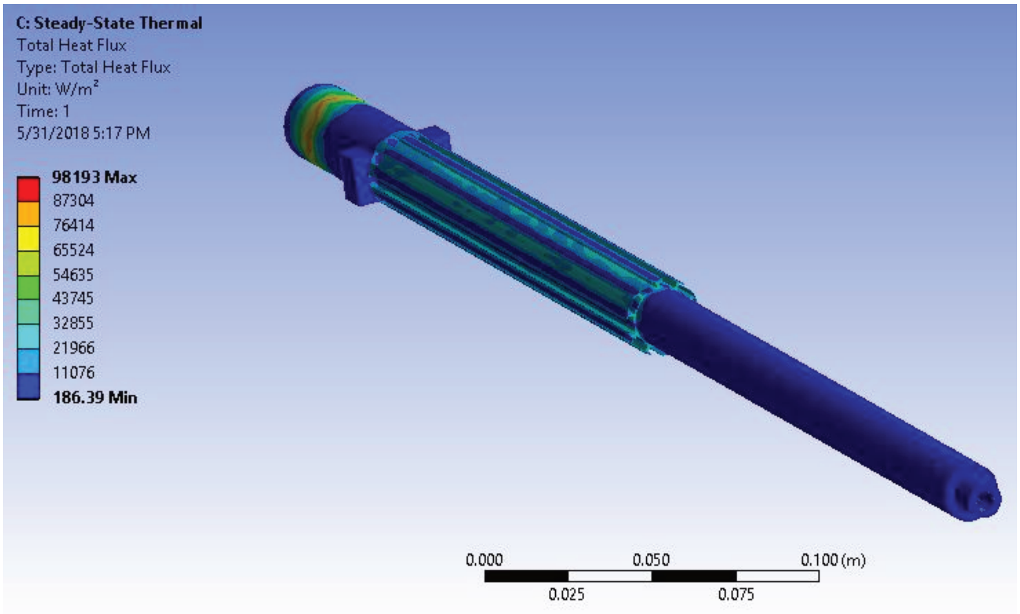

Fig. 7 Enhancement in heat dissipation by T section fins

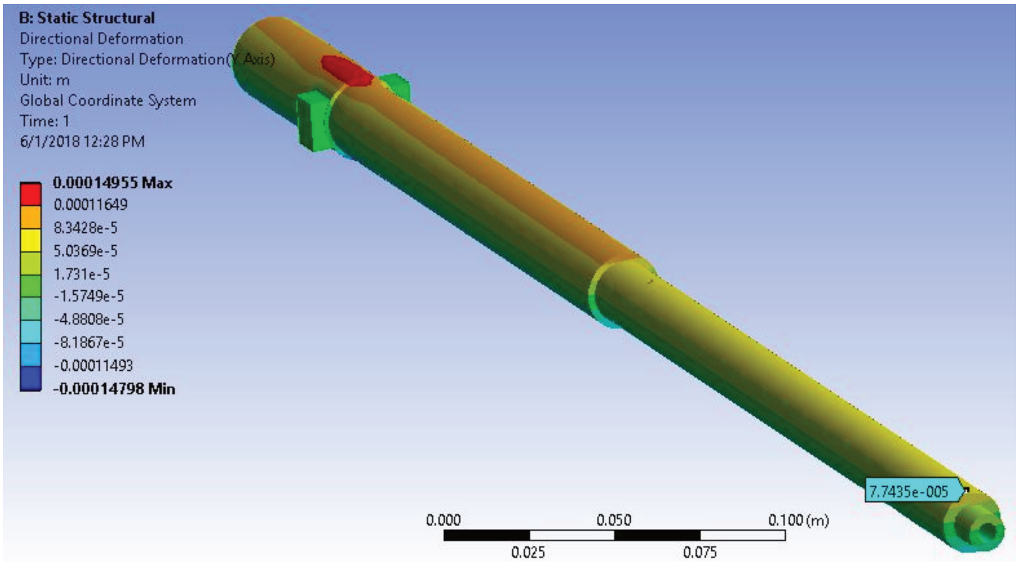

Fig. 8 Directional deformation towards the ground due to gravity in plain barrel. The probe at the muzzle end shows the numerical value for the maximum deflection at the muzzle

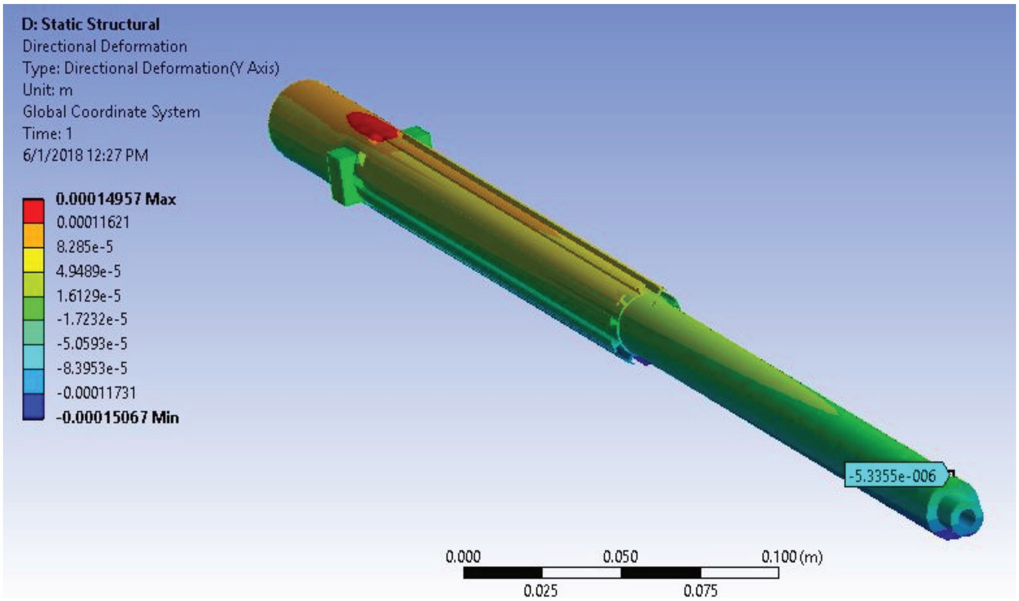

Fig 9 Directional deformation towards the ground due to gravity in modified T-section finned barrel. The probe at the muzzle end shows the numerical value for the maximum deflection at the muzzle

4. Conclusions

The results found were in agreement with the expected outcomes, which shows enhancement in the heat dissipation capability and flexural rigidity along with weight reduction in the modified T-finned barrel against the conventional barrel with thicker profiles and heavier weights.

1. ANSYS 14.5 version was used to generate the mathematical model and simulations. Only the barrel bodies were selected for geometry, excluding all the interacting components such as muzzle devices, gas blocks, front sight posts and trunnion as in actual case.
2. Tetragonal mesh with fine size and medium relevance were applied to achieve best results out of the FEM analysis. The weight reduction was observed at the very first place during the solid modelling. As shown in Fig. 1 and Fig. 2, the weight of the unfinned H-bar was 1 042.93 g while that of the finned barrel was 976 g. The T-section fins forming a pseudo-I section, indeed reduced down the deformation values at the elevated temperatures under the influence of gravity by a considerable difference. This significantly large value of 93 % reduction in gravity-governed deflection indicates a proportional amelioration in the flexural rigidity of the barrel body.
3. Due to factors affecting manufacturability, the extension of fins was halted at the mid length, i.e. where the gas block attaches (in case of a gas operated weapon). Results may indeed vary with different lengths of fin extension.
4. Due to the increment in flexural rigidity, it may be initially assumed that this modification could also result in scaled up spectrum of natural vibrational frequency of the barrel, which would indeed be conducive to guns having high rate of firing. There remains a further scope for specialized focused vibrational and harmonic analysis for both barrels to identify the difference in the resonance frequencies and subsequent deflections under the same loading conditions. Hence, through the conducted analysis, the barrel having T-fins was concluded to have achieved the solution of all the stated problems. This modified design further requires going through a detailed DFMA (Design for Manufacturing and Assembly) study and vibrational study in order to become a commercial success.

Acknowledgements

This work was supported by the Department of mechanical engineering, Maulana Azad National Institute of Technology, Bhopal. The authors express their deep gratitude to Department of mechanical engineering, MANIT for the benign support.

References

[1] HAMEED, A., AZAVEDO M. and PITCHER, P. Experimental Investigation of a Cook-off Temperature in a Hot Barrel. *Defence Technology,* 2014, vol. 10, no. 2, p. 86-91. DOI 10.1016/j.dt.2014.05.006.

[2] BANNISTER, E.L., JONES, R.N. and BAGWELL, D.W. *Heat Transfer, Barrel Temperatures and Thermal Strains in Guns* [Report]. Maryland: Army Ballistic Research Laboratories Aberdeen Proving Ground, 1963, 61 p.

[3] ABHILASH, P. and CHOPADE, M.R. Analysis of Heat Transfer Coefficient inside Gun Barrel. *International Journal of Current Engineering and Technology*, 2016, no. 5. p. 460-463. ISSN 2277-4106.

[4] YONGHAI, W.U. Analysis of Thick-Walled Cylinder Temperature Field based on the Thermal-Fluid-Solid Coupling. *Research Journal of Applied Sciences, Engineering and Technology*, 2013, vol. 5, no. 16, p. 4094-4100. ISSN 2040-7459.

[5] ROJACZ, H., VARGA, M. and WINKELMANN, H. Deformation Mechanisms at Elevated Temperatures: Influence of Momenta and Energy in the Single Impact Test. *International Journal of Mechanical and Mechatronics Engineering,* 2013, vol. 7, no. 5, p. 877-883. DOI 10.5281/zenodo.1061864.

[6] FIŠER, M., PROCHÁZKA, S. and ŠKVAREK, J. *Weapon Barrels* (in Czech) [Textbook]. Brno: University of Defence, 2006, 200 p.

[7] SEMAN, P. *Deformation Analysis of Muzzle Part of Barrels* (in Czech) [PhD Thesis]. Brno: University of Defence, 2011. 133 p.

[8] CENGEL, Y.A. *Heat Transfer: A practical Approach.* [Textbook], [on-line], 2nd edition, 874 p. [cited 2018-02-12]. Available from: http://www.uotechnology.edu.iq/dep-materials/lecture/secondclass/HeatTransfer14.pdf.

[9] MACKOWSKI, D.W. *Conduction of Heat Transfer Notes for Mech 720* [Textbook], [on-line]. 242 p. [cited 2018-02-06] Available from: http://www.eng.auburn.edu/~dmckwski/mech7210/condbook.pdf.

[10] SHIMPI, R.P., SHETTY, R.A. and GUHA, A. A Simple Single Variable Shear Deformation Theory for a Rectangular Beam. *Journal of Mechanical Engineering Science,* 2016, vol. 231, no. 24, p. 4576-4591. DOI 10.1177/0954406216670682.

[11] THAI, S., THAI, H.T., THUC, P.Vo. and PATEL, V.I. A Simple Shear Deformation Theory for Nonlocal Beams. *Composite Structures,* 2018, vol. 183, p. 262-270. DOI 10.1016/j.compstruct.2017.03.022.

[12] LILJA PRECISION RIFLE BARRELS, Inc. Drawing no. AR-10_740_18_mod. [cited 2018-01-10]. Available from: www.riflebarrels.com.

[13] ABAQUS Analysis User's Manual Version 6.7. SIMULIA Dassault Systemes, 2007.

[14] DEBSKI, A., KONIORCZYK, P., LECIEJEWSKI, Z., PREISKORN, M., SURMA, Z. and ZMYWACZYK, J. Analysis of Heat Transfer in a 35 mm Barrel of an Anti-Air Craft Cannon. *Problems of Mechatronics: Armament, Aviation and Safety Engineering,* 2016, vol. 7, no. 3, p. 71-86. DOI 10.5604/01.3001.0009.2983.

[15] ANSYS Workbench Strain Life Approach Lectures [on-line], 2017. [cited 2018-01-13]. Available from: https://mycourses.aalto.fi/mod/resource/view.php?id=88051.

[16] THOMPSON, M.K. and THOMPSON, J.M. Introduction to ANSYS and Finite Element Modelling. In: Eds THOMPSON, M.K. and THOMPSON, J.M. *ANSYS Mechanical APDL for Finite Element Analysis*. Oxford: Butterworth-Heinemann, 2017, p. 1-9. ISBN 978-0-12-812981-4, DOI 10.1016/B978-0-12-812981-4.00001-0.

[17] BACHOO, R. and BRIDGE, J. The Modal Density of Composite Beams Incorporating the Effects of Shear Deformation and Rotary Inertia. *Journal of Sound and Vibration,* 2018, vol. 423, p. 459-471. DOI 10.1016/j.jsv.2018.01.013.

[18] WANG, S., TONG, G. and ZHANG, L. Reduced Stiffness of Composite Beams Considering Slip and Shear Deformation of Steel. *Journal of Constructional Steel Research,* 2017, vol. 131, p. 19-29. DOI 10.1016/j.jcsr.2016.12.003.