BASF Corporation

Optimizing Mechanical Performance of Injection Molded Multiple Gated Rotating Thermoplastic Components: *Part 1 – Consideration of Structural Analysis and Knit Line[1] Effects*

ABSTRACT

Engineering thermoplastics were successfully utilized in the design of injection molded rotating parts such as the impellers, wheels, and cooling fans of commercial air-cooled chillers, and gas and diesel engines.

Complex aerodynamic and mechanical performance of impellers and cooling fans are very important for the efficiency of integrated air-movement, climate control and cooling systems of various types of engines of vehicles, cars, heavy-duty tractors and trucks. The transportation and automotive industries have developed a culture of reliability and cost effectiveness, in which high risks and adventures are not encouraged.

Due to the wide and ever increasing application of thermoplastics for the transportation and automotive industries, the performance of the under-the-hood parts depend upon optimized design and processing technology and properties of polymer based materials.

The mechanical properties of the injection-molded thermoplastic components depend on the part and the molding tool design. For injection molding of the multi-blade fans and various rotating plastic parts, the complex of multiple gating injection molding tools were used. Both the design of the various rotating parts (including the industrial and automotive cooling fans), and the molding tool design are very important to get optimum flow patterns and to predict the locations of stress-bearing areas and knit lines (planes)[1].

For non-reinforced or non-filled nylon, the mechanical performance in the knit lines (planes) areas are approximately equal to the mechanical performance of the resin (polyamide) used. Fiber-glass reinforced and fiber-glass/mineral nylons have mechanical properties of plastic in the knit (weld) line (plane) and is different from basic mechanical properties of reinforced plastic due to flow patterns and local fiber-glass re-orientation in the weld plane areas. Due to the above changes, the weld planes (lines) become likely areas of crack initiation and propagation and possible molded part failure or damage.

In this investigation, we are presenting the results of analytical structural analysis and design optimization of various multiple gated rotating thermoplastic parts such as wheels, impellers, and multi-blade fans with an external ring under the influence of mechanical parameters of fiber-glass reinforced plastic in local knit (weld) plane areas. For determination of these mechanical properties in local (weld plane) and bulk (molded part) areas, the influence of molding (melt and mold temperatures, shear rate, etc.), and end-use (strain rate, temperature, moisture, etc.) conditions should be taken in to account.

The results from this study should help designers to accurately interpret the results of structural analysis and complex tensile (as basic) properties, such as strength, deformation and fatigue of nylon based plastics and to utilize these important material parameters at end-use conditions for a part life assessment.

INTRODUCTION

Thermoplastic cooling impellers, fans, and fan shrouds first came [1-3] into general use in the early sixties[2]. French automaker PSA molded from non-reinforced nylon a fan with eight flexible blades. The integral injection molded cooling fan design reduced cost by replacing ten stamped metal parts and also eliminated thread-forming screws as a method for attaching of the blades to the hub. Variations of this injection molded fan design are used in many industrial air-cooled chillers, compact-cars today [3-5]. Injection molded cooling fan was more aerodynamic than stamped metal, which improves engine and air-cooled chiller efficiency, reduces noise, and lowers part fatigue failures.

There were obvious advantages in replacing laboriously manufactured stamped sheet metal multi-parts by one-piece injection molding cooling fan or shroud system. A good example of consolidation is the automotive under-the-hood integrated cooling fan/shroud system or module (panel and two fans) shown in Figure 1.

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[1] Sometimes called the weld line or weld plane [1-2].

[2] The first nylon made fan with flexible blades appeared on a Citroen DS19 at the 1955 Paris Auto Show [2]

Figure 1. Integrated shroud/fan module

An assembly of the front end involving more than twenty steel shaping operation is replaced by a single injection molded thermoplastic part. The choice on type of thermoplastic for the impellers or fans depends on the strength and rigidity/stiffness requirements, design considerations, and the length/time of vehicle/car model run [3, 6-7].

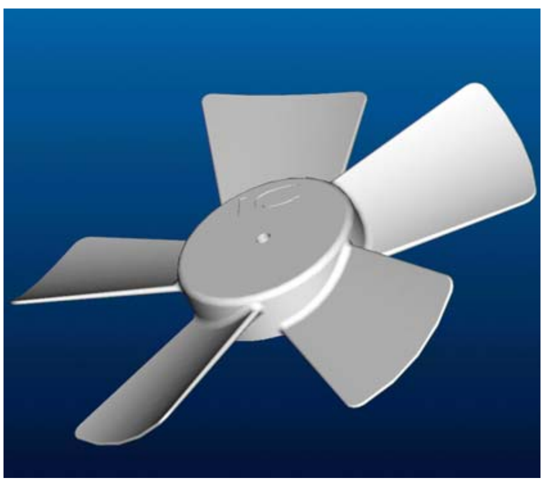


Figure 2. Cooling fan with the five flexible blades

For some of the first metal to thermoplastic conversions, polypropylene (PP) with good moldability and properties at low cost was a favorite for some industrial and automotive cooling fans. Higher ambient temperatures, greater air movement, or higher fatigue and creep requirements at the hub have often necessitated moves to higher specification thermoplastic [8-10].

Second an injection molded five blades flexible nylon (polyamide – PA) made fan was introduced by Mercedes in 1968. Higher temperatures alone would suggest a switch to glass-fiber reinforced PP; higher stress and life specification would suggest glass-fiber reinforced nylon, whereas high impact strength requirements would lead to non-reinforced nylon 6.

Both types of injection molded cooling nylon fans (with flexible blades – Figure 2 and with a ring – Figure 3) are widely used in the design of many automotive and other industrial applications. The structural analysis of these complex geometry rotating parts, requires utilizing data of many mechanical parameters (short-term and long-term) with the influence of end-use conditions[3]. Due to the advantages of injection molding tools design, typically stamped metals fans are not as efficient as plastic fans (with optimized by geometry for the better air-flow).

Figure 3. Cooling fan with the five blades and a ring

Wide experimental and design programs in the application of nylon for under-the-hood components and integrated systems of various vehicles demonstrated nylon's capacity for resisting heat and chemicals at various mechanical loading [6-9].

For automotive cooling fans, the main requirements are airflow, strength (including impact) and fatigue life [10-11]. Fan shrouds main requirements are for rigidity and dimensional stability, often in quite thin sections, rather than impact strength and creep. High performance fiber-glass reinforced and glass / mineral versions of nylon 6 were utilized in various automotive and commercial air-cooled chillers applications.

WELD AND MELD LINE (PLANE) INTEGRITY

The successful design and advanced manufacturing of injection molded thermoplastics components require a correct combination of knowledge in material, design and processing technology. Although the injection molding process offers a wide degree of freedom of design, optimized results can be obtained only if the

[3] Design principles of a large group of the cooling fans of commercial air-cooled chillers are very similar to principles applied in the design of automotive cooling fans [2, 4]. The minor differences for air-cooled chillers are in the operating temperatures (for a running fan can range from - 12 to 70°C).

product and mold designers take the numerous processing factors into account, and they will realize these combined effects in finalizing all processing conditions and molding tool design. Many thermoplastics of an injection mold will influence the final products and assembled parts mechanical performance, dimensions and other end-use characteristics.

These mold parts include the cavity shape, gating, parting line, vents, undercuts, ribs, hinges, etc. Additionally, there are many cases such as parts requiring multiple gates or family mold cavities. The molding tool designer should take all these factors into account.

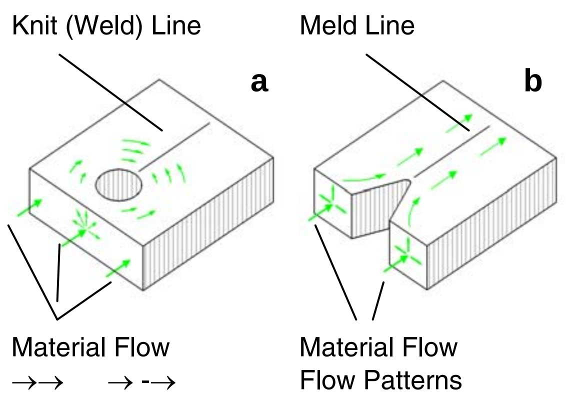


Figure 4. Principles in formation of knit (weld) and meld line and weld & meld flow paths: a – flow around cylindrical hole; b – flow around ribs or long insert.

Weld and meld lines (planes) are created wherever polymer flow' fronts meet from opposite or parallel directions correspondingly (Figures 4-5). They are significant for thermoplastic part performance because the local mechanical properties in the weld and meld plane area differ significantly from those in the rest (total/bulk) areas of the molded parts.

Due to the above described changes, these weld planes (lines) become likely areas of the crack initiation and propagation and possible molded part failure.

Weld lines are formed when more than one gate is used (Figure 5) or wherever a divided stream of thermoplastics joins after flowing around a pin or core (Figure 4a).

These sections are particularly prone to weak weld lines because of possible rapid melt solidification. Knit (weld) lines are created where two flow fronts from opposite direction meet (Figure 4a and 5).

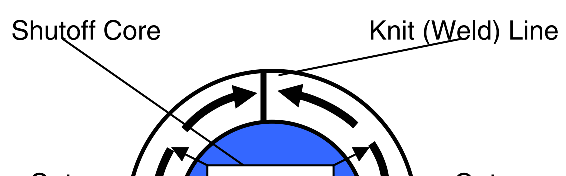


Figure 5. Example of the knit (weld) lines formation

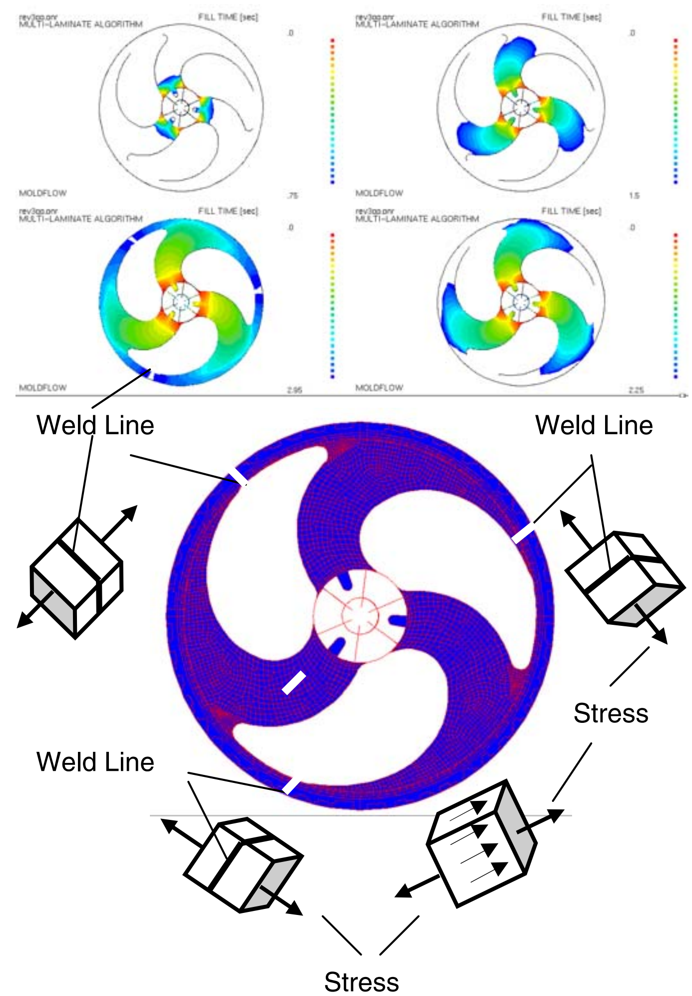


Figure 6. Flow patterns and weld lines formation (MoldFlow® data) in the injection molded thermoplastic part with a ring (nylon 6, 33 wt.% GF)

Meld lines are created where two flow fronts from different but not opposite direction met (Figure 4b). Weld lines are weaker than meld lines.

These weakness change the thermoplastic strength and deformation parameters.

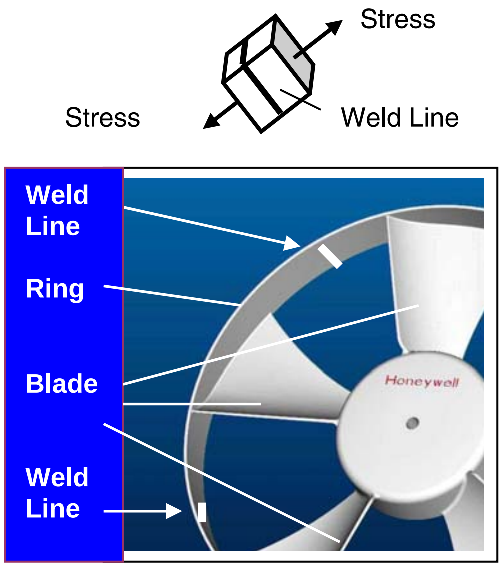


Figure 7. Weld lines formation in a ring of cooling fan (between every pair of the blades)

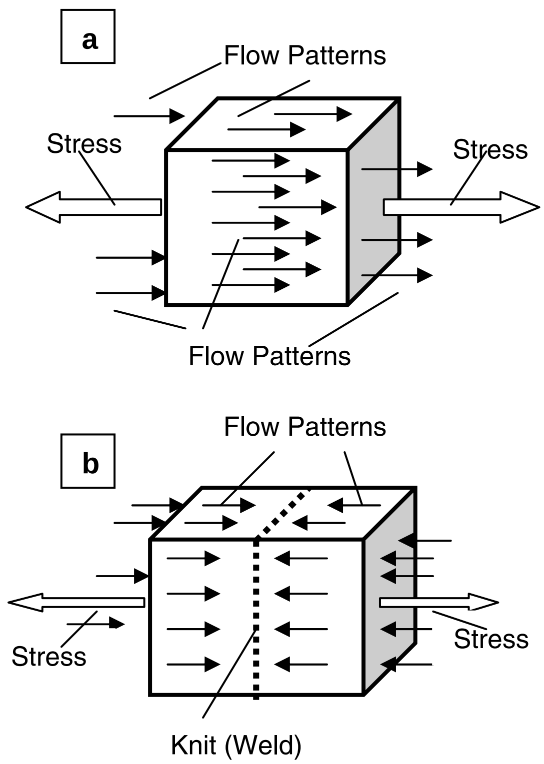


Figure 8. Effect of orientation on mechanical performance of injection molded parts/specimens: a – applied stress is parallel to flow direction (unidirectional fiber-glass orientation); b – applied stress is perpendicular to the knit (weld) line.

Figure 6 shows various flow patterns in multiple gated rotary components such as an injection-molded wheel, which is similar by configuration to automotive cooling fan. These flow patterns are very influential on mechanical performance of various multiple gated injection molded plastic parts.

For the multi-blades cooling fan with a ring, the knit (weld) lines will be created between every two opposite melt flows in a ring area (Figure 7). As a rule, the location of these knit (weld) lines is between every pair of the thin blades. Orientation of these knit (weld) lines, are perpendicular to the tensile stresses, applied to the ring (Figure 6-8). The knit (weld) line is in the direction perpendicular to applied stress. For fiber-glass reinforced thermoplastics, the strength in these local areas (knit/weld and meld lines) is less than in bulk areas, due to the orientation effects (Figure 9).

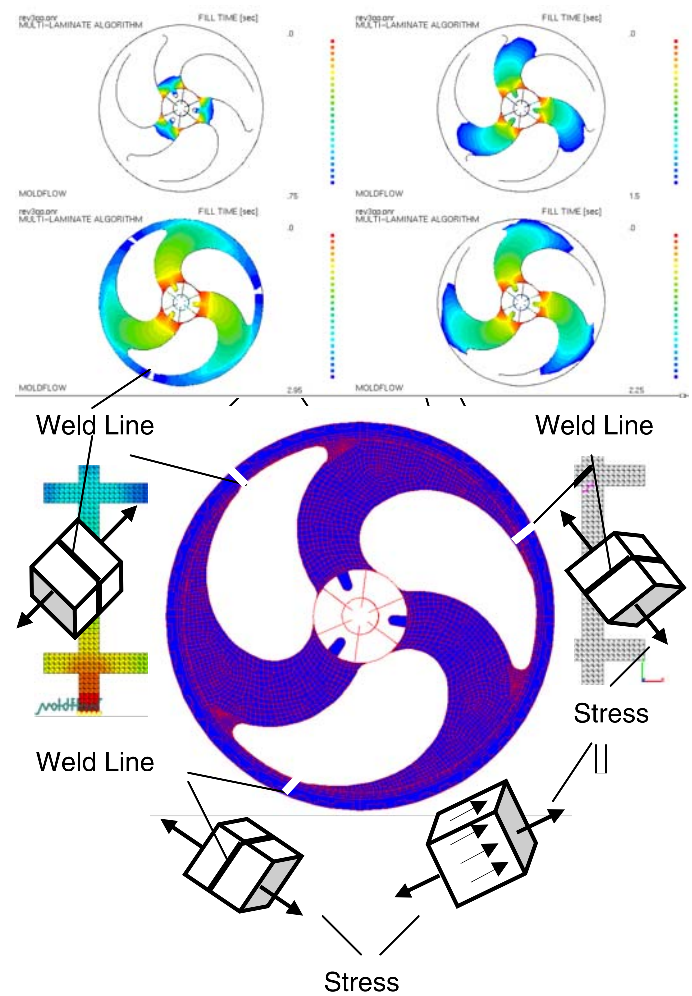


Figure 9. Different plastic flow patters create weld and meld lines (MoldFlow® data)

The maximum of tensile strength of molded plastic is in the direction parallel to flow (Figure 8a).

The extent of physical and mechanical property change depends on the ability of the two melt flows to knit (joint) together homogeneously. In general, the following “key” parameters affect weld plane integrity [1, 12-14]:

- Type of base resin
- Molded part/specimen thickness

- Presents of fillers and reinforcements
- Injection molding process conditions (such as temperature and viscosity of the molten polymer lastic when it meets)
- Orientation and interaction of flow and stress patterns.

The number of knit (weld) lines (planes) may be determinate by equation (1):

$$N = G + C_0 - 1 \qquad (1)$$

Where

N – number of weld planes (lines)

G – number of gates

C_0 – number of shutoff cores (or pins)

A practical thermoplastic part design – melt flow consideration is to combine concepts of full optimization of *mechanical performance of* thermoplastic parts where multiple gating systems were used. The number of gates and internal shutoff cores should be considered as an important aspect of:

- Initial thermoplastic part design
- Prediction of part performance with the influence of location of stress-bearing areas and weld planes / lines.

Standard mold filling and cooling analysis (MoldFlow® data, Figure 9) can provide the designer and technologist with quantitative data on flow patterns and weld lines (planes). Critical plastic flow distance, and the number of gates and internal shutoff cores should be considered an important factor of thermoplastic part design for required uniform mechanical / physical performance. It was assumed [8-9] to use for design purpose the tensile strength of knit (weld) line in range from 50 to 95% of base material[4] strength.

Unfortunately, it was not possible to find in the published literature recommendations on thermoplastic selection and design of impellers and automotive cooling fans with the external ring. The data on mechanical performance of nylon in knit (weld) line is limited also [1, 12, 14]. These recommendations and specific (localized) material properties data is very important for the structural analysis and life assessment of injection molded (using multiple gating system) wheels and automotive cooling fans with a ring.

[4] The definition of the strength of base material is not clear enough in used trade literature and published reports. By our opinion “the strength of base material” is strength of the matrix, but not strength of reinforced plastic.

STRUCTURAL DESIGN ANALYSIS OF INJECTION MOLDED COOLING FAN WITH A RING

INTRODUCTORY REMARKS

Structural analysis of rotating plastic parts such as cooling fan with a ring (Figure 3), wheel (Figure 6) or impeller (Figure 10), is very complicated due to complex geometry of the blades/spokes and a ring/base plate and loading conditions.

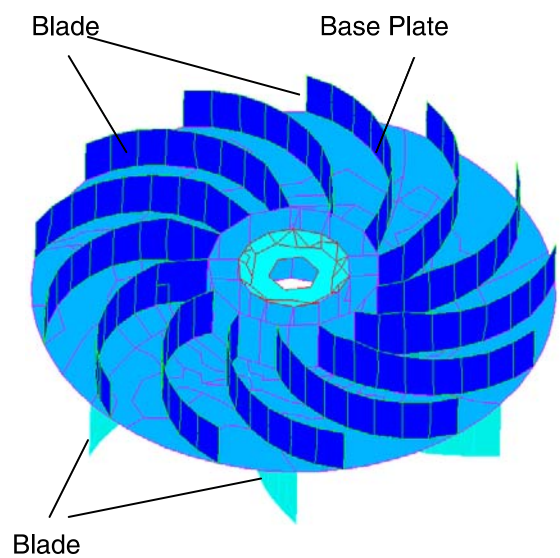


Figure 10: Structural model of injection molded multi-blade impeller

All types of the impellers and automotive cooling fans and shrouds should be optimized for the better airflow and mechanical performance by short-term and long-term strength & life criterion.

At the same time, modeling of these rotating injection molded thermoplastic components is more complicated. Used models should take into account the complex of local (weld plane areas) material properties (with the influence of orthotropic properties of fiber-glass reinforcement thermoplastic) and localized stiffness distribution for the blades and a ring. In some cases, knit (weld) lines may have low resistance to flexural loading also due to formation of local yield effects in weld inter-phase.

Models of the cooling fans with the flexible blades (Figure 2) are more convenient for the structural analysis, and it is possible to analyze mechanical performance of the cooling fan itself by using data developed stress-strain and strength/life data for a separate blade only.

All cooling fans should operate properly during the whole life cycle of a car engine (ten years and more). Fatigue performance of cooling fans with flexible blades depends on the amplitude of tensile and flexural stresses at various loading (fixed rotations, and start-stop cycles) and environmental conditions. Resistance of the flexible blades to fatigue crack initiation and propagation (in the areas close to the hub) is a limited factor in proper design of these fans.

Mechanical performance of the cooling fans with the ring (Figure 3) and wheels (Figure 6) depends from the integrated short-term (including impact) and long-term properties (fatigue) in knit (weld) line (plane) areas.

Qualification of short-term (static and dynamic) tests of the impellers, automotive cooling fans and the cooling fans of commercial air-cooled chillers typically consist of:

- Impact test (at room and minus temperature conditions)
- Running the fans through burst (rotating to failure) at room temperature (23°C) for conditioned (to moisture equilibrium – 50% RH at 23°C).

Some specific qualification long-term/fatigue (cycling) tests of these fans typically consist of running the cooling fans through cycling of temperature (heating-cooling) and start-stop cycles [2, 13].

ANALYTICAL INVESTIGATON

<u>"Model 0"</u>

Uniformly rotating ring – tensile stresses in the uniformly rotating ring and ring extension.

<u>Assumptions</u>

The ring is thin and ratio $t/R_0 \subset\subset 1$ (Figure 10), where

R_0 is radius of restrictive ring (average),
t – thickness of restrictive ring (average).

This ring is unsupported (free) and rotates uniformly with angular velocity ω. Tensile stress σ_{R0} caused by the uniform rotation is distributed uniformly through the ring thickness *t*.

<u>Approach 1</u>

The dynamic problem of rotating ring is usually substituted by a quasi-static problem. The ring is considered, as loaded by uniformly distributed radially oriented static load with the intensity p_0 (Figure 10a).

The load found as follows. Consider an element of the ring circumference (Figure 10b), defined by an elementary angle $d\varphi$. Such element has a mass

$$dm = \rho t b R_0 d\varphi\,, \tag{2}$$

where ρ - is thermoplastic density.

During uniform rotation with angular velocity ω an elementary centrifugal force

$$dP_0 = p_0 R_0 d\varphi = dm\omega^2 R_0 \tag{3}$$

From (2), (3) follows expression for load intensity p_0

$$p_0 = tbR_0\omega^2 \tag{4}$$

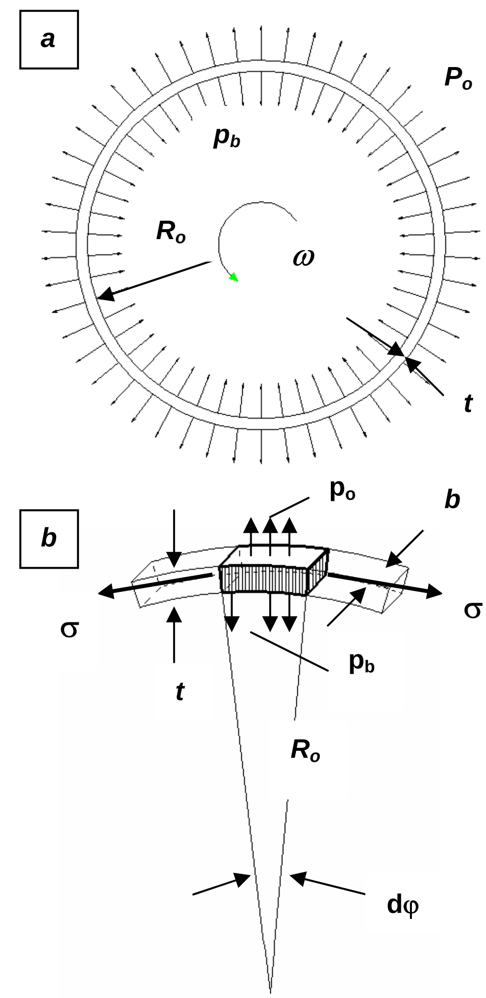


Figure 10. Rotating ring and the loading scheme (Model 0)

Uniformly distributed over the ring cross section tensile stress σ_{R0} (the stress caused by load intensity p_0) can be found considering the balance of a semi-ring generated by an imaginary section of the ring in half. Since projection of all forces (internal and external) should be equal to zero

$$\int_0^{\pi} p_{0y}(\rho) R_0 d\varphi - 2\sigma t b = 0 \qquad (5)$$

Here $p_{0y}(\varphi)$ are Y constituents of p at angle φ.

The integral in (5) is equal $2p_0R_0$. Therefore, tensile stress

$$\sigma_{R0} = p_0 R_0 / bt = \rho R_0^2 \omega^2 \qquad (6)$$

Formula (6) gives the value of tensile stress σ_{R0} in a thin, freely rotating ring. According to the model the ring is under simple tension. The value of elastic strain is $e = \sigma / E$. Circumference extension is

$$\Delta_c = 2\pi R_0 / bt \qquad (7)$$

The radial dimension of the ring is increased by

$$\Delta_R = eR_0 = \sigma R_0 / E \qquad (8)$$

Considering (6) and (4) radial displacement

$$\Delta_R = \rho \omega^2 R_0^3 / E \qquad (9)$$

Approach 2

Figure 11 shows a ring supported by a number of blades (spokes) on the metallic hub. This is a two-dimensional problem, i.e. the spokes (fan blades) are molded as rectangular plates of the same thickness b as the ring.

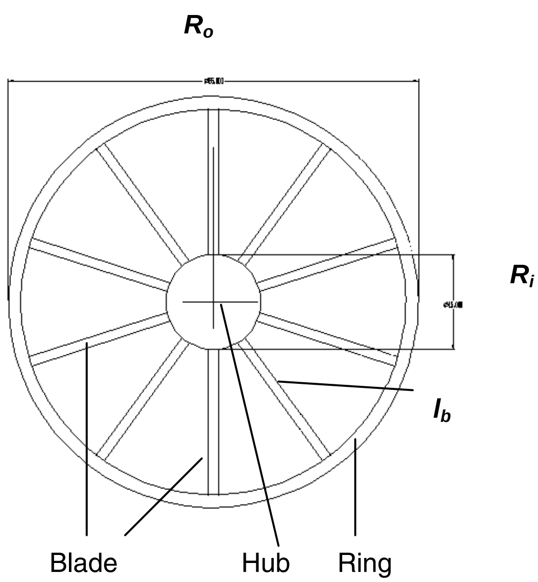


Figure 11. Ring with the blades and a rigid metallic hub

One can expect two opposite effects in a situation. In one case, the ring can restrict extension of the massive fan blades the extension caused by the centrifugal forces of their mass; in another case, when the fan blades have a smaller mass and are rather rigid, they can restrict the ring extension. Let us evaluate what is going on for some important cooling fan with a ring application of the model.

Before proceeding to the next model we must evaluate the radial displacement Δ_b the outer end of the blade. The blade is attached to the ring hub only (there is no ring) and the fan rotates with the angular velocity ω (figure 12). The hub radius is R_i, The fan blade has constant thickness t_b and width b. The blade length is $l_b = R_0 - R_i$.

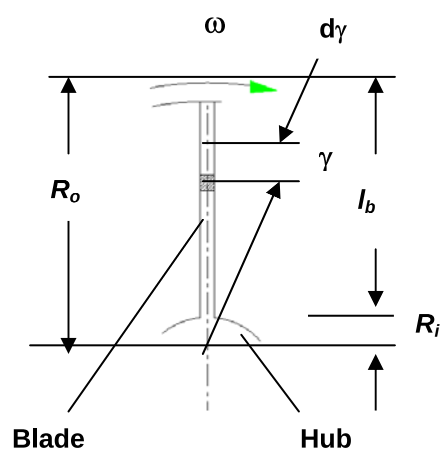


Figure 12. Blade rotating on a rigid metallic hub

The metallic hub is considered to as absolutely rigid and radial displacement of the blade Δ_b is assumed consisting of elementary extension $e(y)dy$ from element dy defined by the radial coordinate y :

$$\Delta_b = \int_{R_i}^{R_0} e(y) dy \qquad (10)$$

$$\Delta_b = (1/E) \int_{R_i}^{R_0} \sigma(y) dy \qquad (11)$$

Stress $\sigma(y)$ at the cross section *y* is equal

$$\sigma(y) = p(y) / bt_b \quad (12)$$

where

$$p(y) = \int_y^{R_0} y' \omega^2 dm = \int_y^{R_0} t_b b \rho \omega^2 y' dy = \quad (13)$$
$$= t_b b \rho \omega^2 (R_0 - y^2) / 2$$

the normal centrifugal force in the cross section.

Then

$$\sigma(y) = \rho\omega^2 (R_0^2 - y^2)/2 \quad (14)$$

and the radial displacement of the blade

$$\Delta_b = \rho\omega^2 / 2E \int_{R_i}^{R_0} (R_0^2 - y^2) dy =$$
$$= \rho\omega^2 / 2E(2R_0^3/3 - R_0^2 R_i + R_i^3) \quad (15)$$

Note, that the radial displacement of a blade Δ_b does not depend on blade cross section. The ratio Δ_b / Δ_R is equal

$$\Delta_b / \Delta_R = (2/3 - R_i / R_0 + (R_i / R_0)^3) / 2 \quad (16)$$

We can compare values Δ_b (15) and Δ_R (9) for a typical fan dimensions: R_0 = 0.26m; R_i = 0.12m.

We obtain ratio $\Delta_b / \Delta_R = 0.15$,

which shows that radial displacement of ring $\Delta_R \supset\supset \Delta_b$. This result indicated the role, which the fan blades and the ring play when connected namely the fan blade support the ring at the points of connection.

If the number of the fan blades are big, a model can be considered with the blades supporting the rotating ring uniformly. This supporting effect can be represented by a uniformly distributed force of interaction with a load intensity p_b.

Since the radial displacement of a blade Δ_b is substantially less than the radial displacement of a ring Δ_R, we can first neglect the extension of the blades under their centrifugal forces and evaluate the effects of blades support on the ring with radial displacement of the blade $\Delta_b = 0$.

"Model 1"

A model of a rotating ring with radial displacement restricted by a uniformly distributed multitude of elastic spokes (blades). The purpose of such model is to evaluate in a simplified way the interaction between the ring and the cooling fan blades.

Assumptions:

Assume first that the parameters of the fan blades (spokes) are such that can neglect their elongation (radial displacement) from inertia and their mass. Such assumption would be reasonable for the thin (low mass) and rigid thermoplastic fan blades.

Assume also that the metallic hub is rigid and the number of the fan blades (spokes) connecting the ring to the rigid metallic hub is sufficiently large, and the restrictive forces are distributed uniformly over the ring length and can be represented by some restrictive load intensity p_b.

Let n – is the number of the blades, N_b – is a normal force applied to the fan blade at the point of blade-ring interaction.

Assuming the cooling fan blade has connected cross section (t_b - fan's blade thickness), the outer end of the fan blade extension (displacement) Δ_b is equal to:

$$\Delta_b = N_b l_b / Ebt_b = \sigma_b l_b / E \quad (17)$$

For our case radial displacement of the ring

$$\Delta_R = \Delta_b, \quad (18)$$

- a displacement compatibility equation.

Since the radial displacement of a fan blade:

$$\Delta_b = e_b\, l_b;\ \Delta_R = e_R l R_0; \quad (19)$$

$$\sigma_b l_b / E = \sigma_R R_0 / E; \text{ and } \sigma_b = \sigma_R R / l_b \quad (20)$$

results from compatibility equation.

Expression for load intensity,

$$p = p_0 - p_b \quad (21)$$

for the resulting distributed load on the ring, where load intensity p_b can be defined as

$$p_b = nN_b \,/\, 2\pi R_0 = n\sigma_b t_b b \,/\, 2\pi R_0 , \qquad (22)$$

- n is the number of the fan blades.

From (6) normal stress in the ring

$$\sigma_{R1} = (p_0 - p_b) R_0 \,/\, bt \qquad (23)$$

By substituting load intensity p_b from (22) we obtain normal stress in the ring

$$\sigma_{R1} = \rho R^2 \omega^2 \,/ (1 + nRt_b \,/\, 2\pi l_b t) \qquad (24)$$

Table 1. Effect of the number n of blades and parameter $R_0 \,/\, l_b$ on stress parameter $\sigma_{R1} \,/\, \sigma_{R0}$

n – number of blades	**R/l_b =1.5**	**2.0**	**4.0**
8	0.340	0.282	0.164
10	0.295	0.238	0.135
12	0.258	0.207	0.116

The formula (24) the fan blade supported the ring and in formula (6), the unsupported ring allowed for an interesting comparative analysis of effects of ring and fan blade parameters, such as $R_0 \,/\, l_b$ ratio, number of blades n, thermoplastic density ρ, on the stress level in the ring.

The normal stress in a ring depends on the thermoplastic density (specific gravity) ρ, which may vary from 1.13 to 1.74 for fiber-glass reinforced (0 – 63 wt.%) nylon based thermoplastics (Table 2).

Table 2. Influence of fiber-glass reinforcement (GF, wt.%) on the density (ρ, g/cm^3) for nylon 6 based plastics

GF, wt.%	0	15	33	45	50	63
ρ, g/cm^3	1.13	1.24	1.38	1.49	1.56	1.74

The normal stress σ_{R0} in non-supported freely rotating ring (6) and normal stress in the supported ring σ_{R1} (24) will increase proportionally to the density increase. Figure 13 shows supporting effect (parameter $\sigma_{R1} \,/\, \sigma_{R0}$, where σ_{R1} - stress in supported ring).

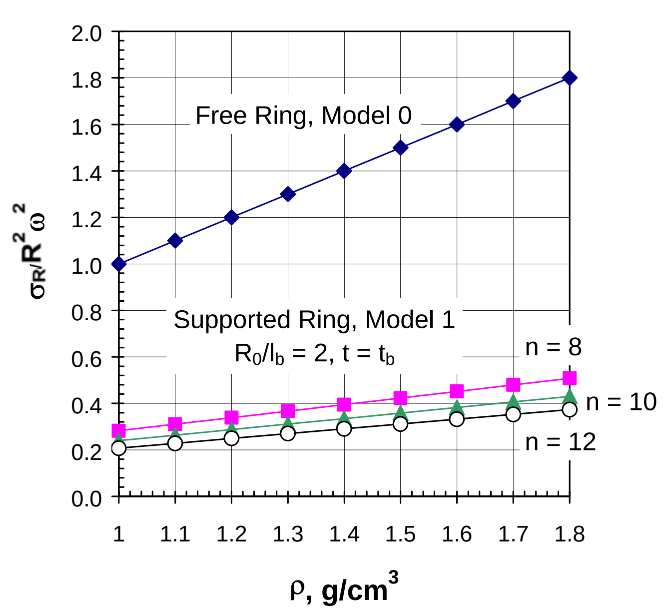


Figure 13. Normalized stress ($\sigma_R/R^2\omega^2$) as a function of thermoplastic density ρ

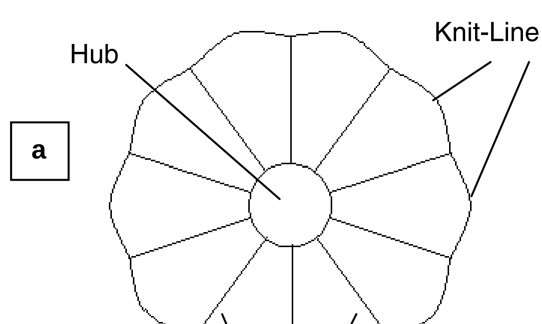
“Model 2”

Assumptions and Approaches

On Model 1, we assumed that the support from the blades are distributed uniformly along the ring, which leads to reduction of the stress level. The loading scheme, however, remains the same as for the unsupported ring (“Model 0”), - the ring remained uniformly extended.

A more realistic model should consider support from the blades at the discrete points of blade-ring connections Figure 14a,b.

The space between the points remains unsupported, which leads to a different mode of ring deformation, namely the parts of the ring between the points of support are bent.



Hub

Blade

$\mathbf{p_o}$

Figure 14. Bending of supported ring (Model 2)

The loading model for this case is a beam length l_s – with ends fixed, as shown in Fig 14c. Solution for the problem is given elsewhere {see (12)}. The maximum moment M_2 at the middle of the specimen (the knit line area) is:

$$M_2 = p_0 l_s^2 / 24 , \qquad (25)$$

where p_0 is the uniformly distributed load.

The maximum bending stress at this point is:

$$\sigma_{R2} = 6M_2 / bt^2 \qquad (26)$$

The width of the blade

$$l_s = 2\pi R_0 / n - b_{bl} ,$$

where b_{bl} - circumferential projection of the fan blade width.

Assume $b_{bl} = k_{bl} l(R_0 - R_i)$ and $R_i = k_b R_0$.

Then,

$$l_s / (R_0 - R_i) = 2\pi / n - k_{bl} k_b \qquad (27)$$

Consider the effect of the number of blades **n** on normal stress σ increase caused by bending as compared to the normal stress in the unsupported ring (4).
The ratio:

$$\sigma_{R2} / \sigma_{R0} = (l_s / R_0)^2 (R_0 / t) / 4 \qquad (28).$$

Table 3 shows the effect of the number of blades in this ratio, for a case $k_{bl} = 0.5$ and $k_b = 0.5$, and for value of parameter $R_0 / t = 65$.

Table 3. Effect of the number of the fan blades **n** on maximum stress ratio $\sigma_{R2} / \sigma_{R0}$ at the knit line.

n – number of blades	**8**	**10**	**12**
Stress ratio = $\sigma_{R2} / \sigma_{R0}$	4.65	2.32	1.21

Note, that the total value of the maximal stress at the knit line area according to the models 1and 2 is

$$\sigma_R = \sigma_{R1} + \sigma_{R2} \qquad (29)$$

A developed family of simple analytical models provide realistic estimate of stresses at the knit line / plane of rotating thermoplastic parts with a ring such as the wheels, cooling fans, etc. The models give a better understanding of the sources for the stresses and indicate ways to control them.

MOLDFILLING AND LINEAR FEA OF THE COOLING FAN WITH EXTERNAL RING

An analytical investigation of structural design has a lot of the advantages (linear and non-linear modes with consideration of time-temperature effects, design optimization) and limitations (difficult to take into account a sharp changes of a part shape and sizes, variations in local material properties).

Several various complex MoldFlow and FEA analysis and design optimization study were performed for the cooling fans with the ring. These fans were different by the design, used thermoplastics, applications, and performance requirements [6, 10, 17], and we present the results of this evaluation for fan with five blades and a ring (Figures 3 and 15-17).

Orientation and distribution of the knit (weld) lines, limiting mechanical performance at ring areas, depends on gating conditions (Figures 15-16).

For central gate molding conditions, the knit (weld) lines were created at a ring preferably between every pair of blades (Figure 15). These knit-lines are long, and the length of these knit-lines is equal to the width of a ring.

The short knit-lines may be developed at area of a blade attachment to the hub.

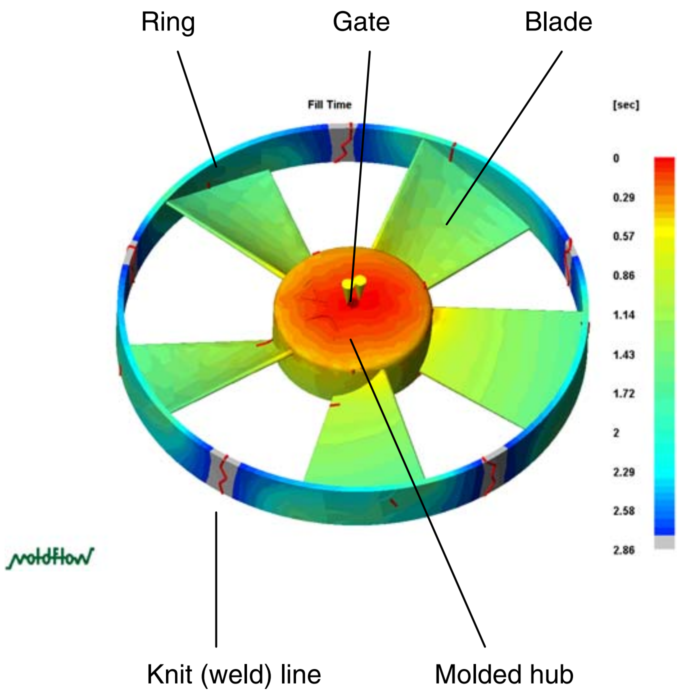


Figure 15. An example of the knit (weld) line formation at central gated cooling fan with a ring (MoldFlow® data).

For multiple gating molding conditions, the knit (weld) lines were at the ring and hub areas between every pair of blades (Figure 16). Both families of developed knit-lines (at ring and at hub) are long and are equal to the width of a ring or 0.5 of a hub diameter correspondingly.

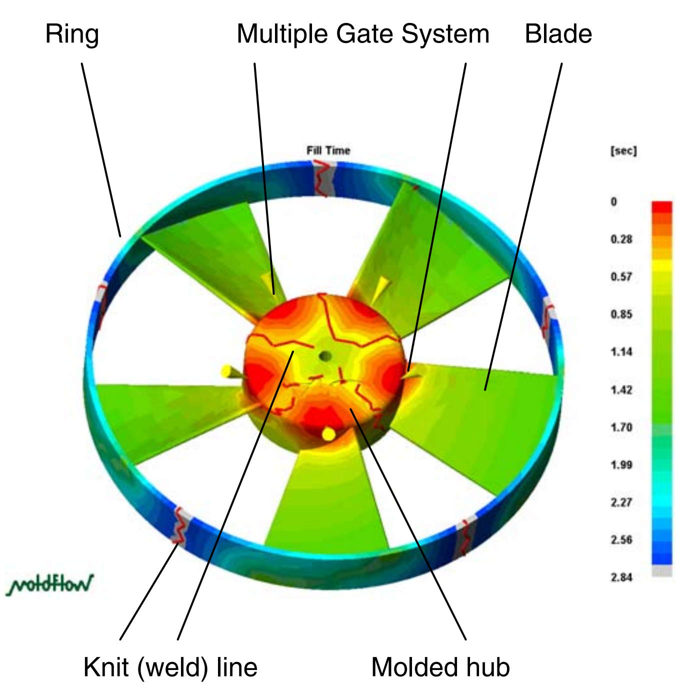


Figure 16. An example of the knit (weld) line formation at multiple gated cooling fan a ring and hub areas (MoldFlow® data)

Two versions of the "Optistruct" (an automatic topology optimization program) supported mass distribution and mass location study at the fan blades, ring and at hub. Because the "Optistruct" does not output actual optimized dimensions, additional FEA models were utilized for design optimization. Some results of FEA for five blades cooling fan with a ring are presented in Figures 17-19.

Distribution of the displacements/deflections at a ring areas is similar for an analytical investigation (Figure 15 a) and for linear FEA (Figure 17). Maximum of deflection is between every pair of the blades (at locations of the knit-lines).

An analytical investigation permits to calculate the radial ring displacement at middle section of the ring, using equation (14) and (19). The results of FEA for displacement analysis (Figure 17) are more comprehensive for the design optimization study. For an example, large displacements were detected at the side of the ring (section A). These large deflections may change airflow and NVH performance at high rotation speed conditions.

Results of the structural analysis of cooling fans with a ring were highly compatible. Differences in values of maximum stresses were in the range 4-6%.

EXPERIMENTAL EVALUATION OF MECHANICAL PERFORMANCE OF ROTATING FAN WITH A RING

For multiple gated thermoplastic parts, designed for automotive and other applications with the high volume of production, both the analytical structural analysis and FEA will not replace complex testing and evaluation. Rather all three (FEA, analytical and experimental mechanics/test) are complementary in nature. Burst test of a molded part(s) and all types of destructive testing of the prototype(s) or specimen(s) supply only one basic answer about material-design-processing technology – either "pass" or "fail".

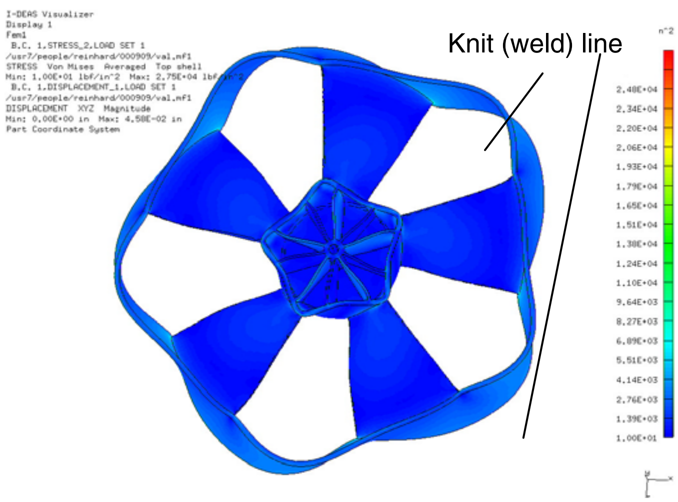


Figure 17. Structural design optimization study: An example of the radial displacements/deflections distributions for cooling fan with a ring (FEA data)

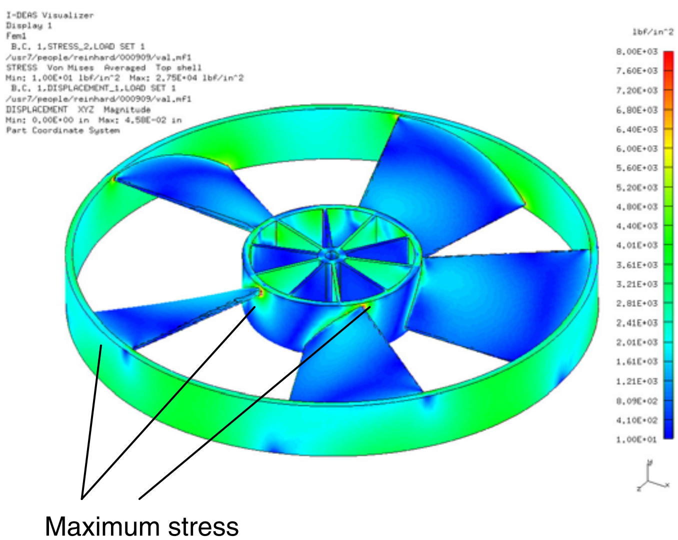


Figure 18. Structural design optimization study: An example of the principles stresses distributions for cooling fan with a ring (FEA data)

Non-destructive testing and evaluation (NDTE), specific advanced methods of the fracture & deformation mechanics and structural analysis (FEA and analytical) provide information with which it is possible to quantify mechanical performance of plastic cooling fans under the end-use conditions (time-temperature, environment, various loads, etc.).

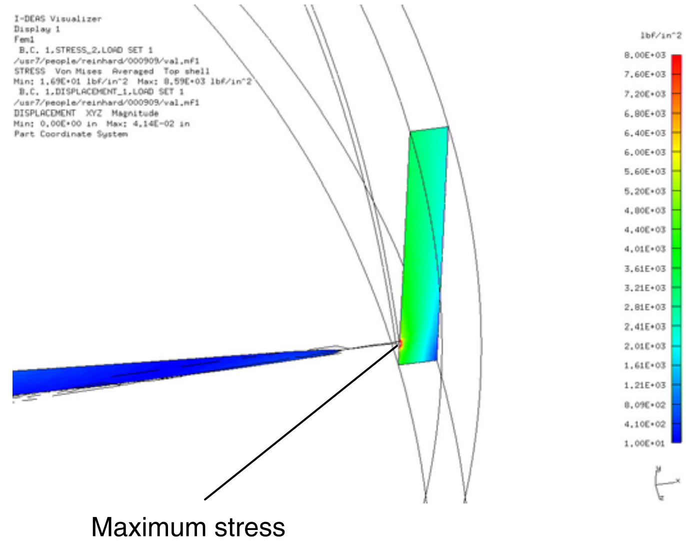


Figure 19. Structural design optimization study: An example of the principles stresses distributions for cooling fan with a ring at blades-ring intersection area (FEA data)

The following test and evaluation results were used in this study for experimental evaluation of mechanical performance of the injection molded fans with a ring:

- Test data obtained from the universal test specimens (Figures 20 and 21) cut from a ring
- Burst speed tests data for rotating fans (with the influence of time-temperature, etc.) conditions.

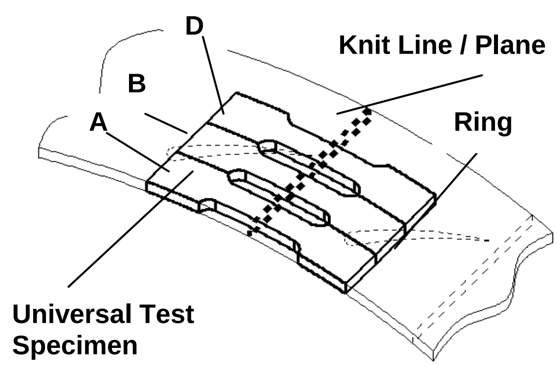


Figure 20. Experimental evaluation of tensile strength at the knit line/plane on the universal test specimens cut from a ring

The similar universal test specimens, as shown in Figures 20 and 21 may be cut from a blade also. As a rule, geometry of a ring and the blades are optimized for better mechanical or airflow performance. Due to these efforts, geometry of the test specimens cut from a ring or blade(s) are not uniform because the thickness of optimized cross-section may vary in wide range. The dimensional analysis of the tested specimens should be taken into account prior to evaluation of the test data.

An experimental evaluation of the mechanical performance of a ring can be performed on the specimens (strips) cut out from the ring in the circumferential direction. To fabricate the specimens, first the attached blades are sliced out flash of the ring, then the ring sliced on a number of circumferential strips (suggested dimensions are similar to recommended by ISO/ASTM).

The knit line is positioned in the middle of the specimen within the gage area of the specimen. Use of long test specimens is preferable to reduce presence of bending during tensile set up. Since the expected strength of the fiber-glass reinforced plastic is lower at the knit line than elsewhere in the specimen, the conventional dog-bone shape is not required but preferable.

By limiting the specimen width, a more accurate strength evaluation can be achieved, especially considering that the thickness of the ring is not uniform across the ring width **b**. More than that, the strength of the knit line can vary across the ring width due to non-optimized injection molding conditions. Thus, more valuable and specific experimental results are obtained.

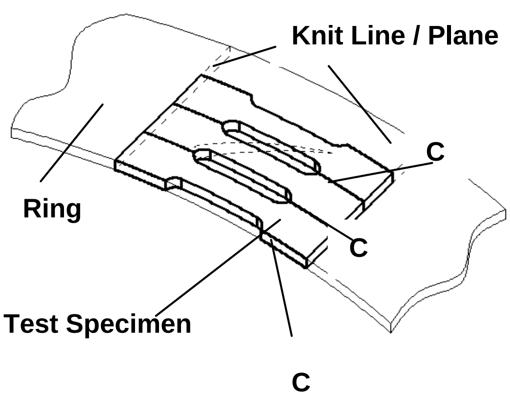


Figure 21. Experimental evaluation of tensile strength at the bulk area (out of the knit-lines) on the universal test specimens from a ring

Universal test specimens cut out similarly but avoiding presence of the knit line (Figure 21) in the test gate area are control specimens (the specimens for evaluating material strength out of the immediate knit line/plane areas).

The cross section area at the test gage length has to be reduced in this case – a conventional requirement (dog-bone shape) for a homogeneous test specimen. At the first approximation, the test can be interpreted as a conventional tensile test, where the stresses in the cross- section sections are distributed uniformly.

Set A - The strips are immediately to the edge, set B - the next, etc. Set C- is control specimens. The ratios $\sigma_A / \sigma_C, \sigma_B / \sigma_C$, etc. indicate the levels of stress retained in the knit line as a fraction of the strength elsewhere in the ring. Typical variation along the width is shown in Table 5.

Table 5. Tensile stress retention variation across the width of the ring of the cooling fan

Stress ratio σ_A / σ_C	**Stress ratio** σ_B / σ_C	**Stress ratio** σ_D / σ_C
0.7	0.8	0.9

In order to obtain a more accurate interpretation of the test measurement, the test should be interpreted taking the ring curvature into the account.

A model for the curved test specimen

Figure 22 illustrates our approach to the problem. Indeed, by clamping the ends of a curved specimen (specimen curvature radius R_0, specimen length L_s) in the grips of a tensile tester, they are turned by angle θ.

This turning is equivalent to applying bending moments - M to both ends. The loading scheme generates a constant bending moment along the length L_s, and therefore the bending curvature $1/R_o$ is constant. Angle θ is defined by cut specimen geometry from Figure 20 as follows:

$$\sin\theta = L_s / 2R_0 \qquad (30)$$

The relation between bending moment M and angle θ - is

$$M = 2\theta EI / L_s \qquad (31)$$

The maximum tensile stress at the internal surface of the ring consists of two components and is equal

$$\sigma = P / bt + 6M / bt^2 \qquad (32)$$

The analysis of the equation (32) for variety of design parameters (ring diameter, ring thickness and width) and a length of the test specimens is showing an importance of correct data interpretation because the additional bending effects are in the range of 20 – 45% from nominal tensile stresses.

Recommendations and Suggestions

We recommend using the analytical models for stress-strain analysis presented in this paper for the initial (conceptual) stage of a cooling fan with a ring design:

- The model for uniformly rotating ring (Model 0) gives the first estimate for the thickness of the ring.

- The model for the rotating blade gives an estimate of the blade thickness required to withstand the stresses from the centrifugal force.

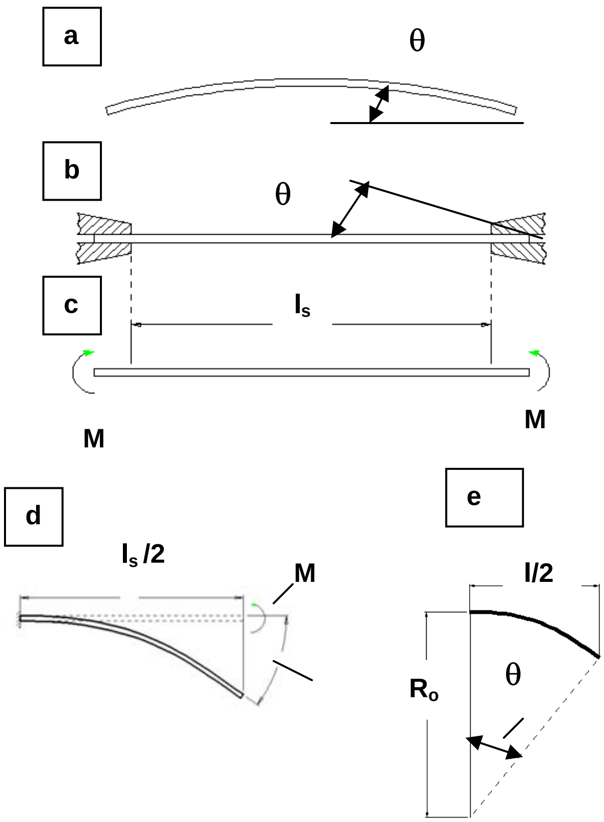


Figure 22. Experimental evaluation of strength at knit-line (plane) – testing of cut specimen with initial curvature.

- Next, by selecting the number of the blades and by using the results from these two models the designer can proceed to the Model 1 to evaluate tensile stresses in the knit line, the stresses reduced by supporting effect from the blades.

- Model 2 should be used next to estimate the bending stresses caused also by the centrifugal forces.

- The first step in optimization of the rotating part design can be achieved by modifying the ring cross section and the number of the blades. The ring cross-section considered be compatible with the expected airflow and other aspects of overall design.

- Furthermore, the rotating part design optimization can be achieved by using FEA.

CONCLUDING REMARKS

Non-reinforced and short glass-fiber reinforced (15 to 35 wt.% GF) nylon based plastics are materials of choice for various injection molded multiple gated components including an automotive cooling fans.

Non-reinforced or moderate reinforced (up 20 wt.% GF) nylon in design of the fans with the flexible blades for non harsh elevated temperature conditions. Glass-fiber reinforced nylon based plastics shall be recommended for the design of the cooling fans with the flexible blades for high temperature conditions and for the cooling fans with a ring.

For fiber-glass reinforced and fiber-glass/mineral nylons, the mechanical properties of plastic in knit (weld) line (plane) are different from the basic mechanical properties of reinforced plastic due to flow patterns and local fiber-glass re-orientation in weld plane areas.

Knit lines/planes become likely areas of a crack initiation and propagation and possible multiple gated injection molded part failure (or damage).

For non-reinforced (or non-filled) nylon, the mechanical performance in knit (weld) plane areas is approximately equal to mechanical performance of used resin (polyamide).

Developed simple analytical models provide realistic estimate of stresses at the knit line/plane of rotating part rings and can assist in designing new cooling injection molded plastic fans with a ring. The models give a better understanding of the sources for the stresses and indicate ways to control them.

For a typical fan the blades support the ring at the points of their ring connection. Bending stresses at the knit line/plane are significant. The support from the blades actually increases stresses there. Increasing the number of the blades reduces this effect.

Curvature of the specimen taken out of the molded ring for testing the strength at the knit line / plane should be taken into consideration because it is substantial.

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KEYWORDS

Nylon; polyamide; plastic; thermoplastic; knit line; weld line; weld plane; weld inter-phase; stress; strain; displacement; tensile; flexural; moisture; temperature; relative humidity; conditioning; fiber-glass; reinforcement; automotive; cooling fan; blade; ring.

DEFINITIONS, ACRONYMS, ABBREVIATIONS

ASTM: American Society of Testing and Materials

BK: carbon black

C_0: number of shutoff cores (or pins)

DAM: dry as molded

G: number of gates

GF: fiber-glass reinforcement

FEA: finite element analysis

ISO: International Organization for Standardization

MF: mineral filled

NDTE: non-destructive testing and evaluation

N – number of weld planes (lines)

PA: polyamide

PP: polypropylene

RH: relative humidity

SAE: The Engineering Society for Advancing Mobility Land Sea Air and Space

SPE: Society of Plastic Engineers

b: width of restricting ring

E – Young's modulus of the fan material / plastic

e – strain

l_b: blade length

n: number of fan's blades (spokes)

p_s : intensity of the restrictive forces of spokes on the ring

p: intensity of centrifugal force uniformly distributed over the ring circumference

wt.: level of plastic reinforcement or filled by weight in %

σ: stress over the ring circumference

dm: mass of the elementary section of the ring (ring element)

dP: centrifugal force on the ring element

$d\varphi$: elementary angle: defines elementary section of the ring

Δ_R : radial displacement of free ring (no restrictions from the blades)

Δ_b: radial displacement of the outer end of the rotary blade (without the ring)

ω: angular velocity of the ring (fan)

R_i : hub radius

R_0 : radius of restricting ring (average)

t: thickness of restricting ring

t_b: fan's blade thickness

ρ: density of the fan plastic

Terms:
Conditioning: the whole series of operations intended to bring a plastic part / sample into a state of equilibrium with regard to temperature and humidity.

Meld line: the mark visible on the finished molded part/specimens, where two plastic flow fronts from different but not opposite direction met during molding.

Weld line: the mark visible on a finished molded part/specimen made by the meeting of two flow fronts of thermoplastic material during molding. Also called *weld mark, flow line or knit line.*

Weld plane: the area or plane in which two polymer/plastic flow fronts meet as the cavity is filled is commonly called a weld plane or weld line (see above please).

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