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Feasibility Study for a Two-Stage Axial Flow Automotive Cooling Pump*

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Seniors in the Department of Mechanical and Industrial Engineering, University of Illinois at Urbana/Champaign have to select a capstone design project in which student teams (groups of 3–4) under the guidance of senior faculty members carry out a theoretical analysis in preparation for designing, manufacturing and testing a working prototype. The results have to be submitted to the industrial sponsor and the departmental faculty in form of a written report (and in oral presentation) for discussion and critical evaluation. As an example, we selected a feasibility study for an alternative (and unconventional) design of an automotive cooling pump, which required considerable deviations from established guidelines for optimal design parameters.

INTRODUCTION

SENIORS IN THE DEPARTMENT of mechanical and industrial engineering at the University of Illinois at Urbana Champaign, (UIUC), are required to complete a senior capstone design experience. Student design teams composed of 3–4 members conduct these projects, working in close cooperation with an industry client. The project experience exposes the students to the professional practice of engineering with requirements for problem definition, statement of project objectives, definition of project deliverables, and completion of a final written technical report and oral presentation.

Project outreach and case study

The capstone design project simultaneously provides students with a complete professional experience while serving the Outreach obligation of the university. The students' capstone design experience is augmented by the full cooperation of the university's technical faculty and utilization of its facilities. As an example, a case study is presented here that follows the developmental stages of a series of automotive cooling pump designs in cooperation with Airtex Products, Inc. of Fairfield, Illinois.

PROJECT HISTORY

While product reliability and space considerations are of utmost importance for a final production quality coolant pump, efficiency and control improvements are the specific motivational factors for this effort. Convenient division of the project is possible, to allow different teams of students to conduct each phase.

* Accepted 20 December 2000.

Phase I: radial pump improvements

In the initial stages of the project, efforts were directed at increasing the hydraulic efficiency of a conventional, shaft-driven, radial pump design. Results show that the efficiency level of this type of pump could be improved from the usual tolerated levels of ~20% to as high as 50% or more with conventional design approaches. However, these designs proved impractical due to severe increases in size.

Phase II: alternative proposed

As an alternative, a study of the feasibility of an electrically driven axial flow pump was proposed. The second phase of the project, (Spring 1995), involved the design, construction, and testing of an electrically driven axial pump. This pump geometry has the attractive feature of allowing the drive motor to be placed within the hub of the pump, thus eliminating the need for an external shaft seal. In addition, the electric motor can provide a convenient speed control that satisfies the cooling requirements of the engine. The first approach followed conventional pump design methods, (scaling of successful models and inviscid flow analysis), and resulted in a prototype that produced unacceptable test results; failing to satisfy the design criteria of $\Delta P = 4$ psi and $Q = 30$ gpm and exhibiting poor efficiency. Analysis of the test results revealed that the established conventional design methodology was not valid for this application [1]. A detailed study of the inviscid flow over the rotor and stator indicated that it was essential to prevent laminar flow separation before the onset of transition to turbulent boundary layer development.

Phase III: redesign

In Spring 1996, the design of a two-stage axial flow pump took place to incorporate the changes recognized from the observations made in Phase

II. By decreasing the number of rotor vanes from 10 to 5 while increasing their lengths from 26 to 39 mm, it was possible to decrease the adverse pressure gradients that led to flow separation while simultaneously increasing the Reynolds number. These design changes satisfied the requirement of inducing boundary layer transition before the onset of laminar separation. Tests conducted in the M&IE Fluid Mechanics Laboratory pump testing facility produced results that indicated that a two-stage axial coolant pump is a feasible alternative to presently available single-stage radial pumps for automotive cooling applications [2]. The size of the pump, 50 mm diameter and 150 mm in length, appears to be attractive for installation within ever more crowded engine compartments.

Phase IV: Continuous Development

Further attempts to improve the hydraulic efficiency of the two-stage axial pump include the design of twisted rotor and stator vanes [3]. Following this will be a study to increase the size of the pump diameter by applying a linear scale factor of 1.25 to the Phase III geometry.

THEORETICAL BACKGROUND

Dimensional Analysis provides an excellent overview for selecting the proper type of machinery for a given task to be performed by pumps, compressors, gas turbines and water and/or steam turbines. For an operating liquid of given density ρ (kg/m^3), and the physical design variables identified as the volumetric rate of flow Q (m^3/sec), the energy increase $E = \Delta P/\rho$ (m^2/sec^2), the rotor diameter D (m) and the shaft speed ω (sec^{-1}) it can be shown that dimensional analysis predicts a relationship between the dimensionless variables [4]

$$\text{specific speed} = \Omega = \frac{\omega Q^{1/2}}{(\Delta P/\rho)^{3/4}}$$

$$\text{specific diameter} = \Delta = \frac{D(\Delta P/\rho)^{1/4}}{Q^{1/2}}$$

A supplementary consideration involves the effects of viscosity (Scale/Reynolds number) on the hydraulic efficiency.

For single stage turbomachines of successful design (high efficiency and large Reynolds numbers) a unique relationship between specific speed Ω as a function of specific diameter Δ has been empirically established the Cordier Curve (Fig. 1).

The specific speed determines under design conditions the geometry (and type ranges) of a stage. Of special interest for the present task (cooling pump design) are the ranges recommended for radial pumps (single stage) $0.2 < \Omega < 1.1$ and axial pumps (single stage) $2.35 < \Omega < 6$.

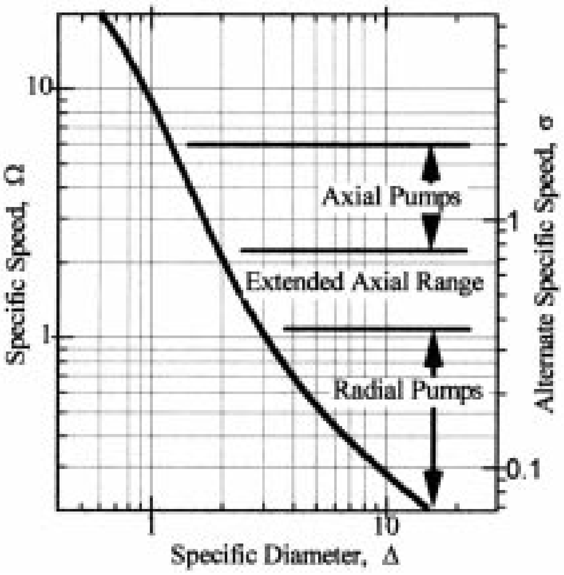

Fig. 1.

It must be observed, however, that a wide variation of design characteristics occurs within each of the ranges identified above, and this can have a profound influence on the expected performance. An illustration of this result for single stage pumps or compressors of optimal design, (axial entry, free vortex flow) [5] occurs in Fig. 2, which depicts the geometry changes over the recommended range. In particular, one notices that at low specific speeds (here represented by the right hand y axis $\sigma = \phi^{1/2}\sqrt{1-\nu^2}\psi^{-3/4} = 0.3355\Omega$), a drastic decline in hydraulic efficiency h_{st} can be expected, attributed to the requirements for large hub diameter ratios ν and its effect on blade tip losses and boundary layer buildup on the walls. In addition, the increase in vane numbers—as indicated qualitatively on top of the figure—and the decrease in vane lengths raise concern about the boundary layer development on the vanes themselves (adverse pressure gradients, laminar separation before turbulent transition) These influences—exacerbated by a small pump size and thus, reduced Reynolds Number level—have been shown to be of importance in evaluating the feasibility of an axial pump in comparison with a single stage radial pump under the specified design operating conditions of

$\Delta P = 4$ psi
$Q = 30$ gpm
$N = 2600$ RPM

Inspection of Fig. 2 for these prescribed design conditions leads to the conclusion that a single stage axial pump appears to be impractical (also see Table 1a). The feasibility of a two stage axial pump (still of marginal position with respect to the Omega range) has to be carefully evaluated (see Table 1b). A 50 mm diameter two stage axial pump at 2603 RPM suggests marginal feasibility, as the specific speed Ω falls into the 'Extended Range' in the Cordier Curve.

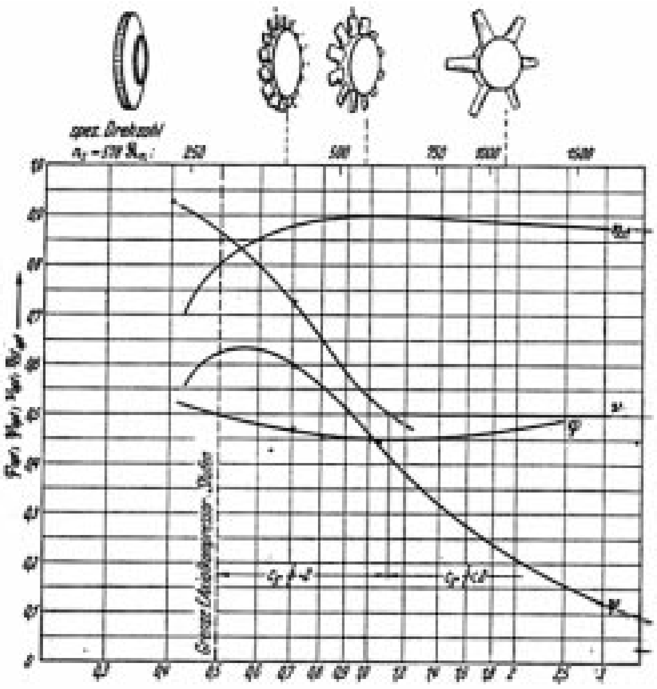

Fig. 2. Geometry changes over the recommended range for single stage pumps or compressors of optimal design.

Table I: Axial pump feasibility based on Cordier Curve (Figure 1) for high Reynolds number and high efficiency. Marginal feasibility possible with 50 mm two stage pump at 2603 RPM.

Table 1a: Single-stage pump				Table 1b: Two-stage pump (values reported for a single stage.)			
Dia.[mm]	Omega	Sigma	N [RPM]	Dia. [mm]	Omega	Sigma	N [RPM]
50	1.232	.413	3255	50	1.657	.556	2603
60	.935	.314	2470	60	1.214	.407	1907

DESIGN CONSIDERATIONS AND PROCEDURES

Attractiveness of an axial pump

After evaluating the performance of single stage radial pumps already in use in automotive applications, the attractiveness of an axial pump (smaller size, the possibility to accommodate an electric motor in the hub and the potential energy savings due to rotor speed control) must be weighed against the limits imposed on its hydraulic efficiency impaired by the design conditions.

Three student teams: Spring 1995, Spring 1996 and Summer 1996, took a pragmatic approach to explore an essentially unknown regime for pumps in three consecutive phases, carried out within the M&IE 280 Capstone Design course.

Two-stage axial pump (Phase II)

In Spring 1995, students based the first design of a Two-Stage Axial Pump on the information provided in Figs 1 & 2, covering an extended range of specific speeds. As can be seen, there is a sharp decline in hydraulic efficiency for values of Omega less than 2. For small pumps, the need for large hub ratios ($\nu > 0.8$) will exacerbate the detrimental effects of tip clearance losses and boundary layer buildup along the hub and housing. The suggested number of vanes (approximately 10) also leads to unfavorable conditions for the flow past rotor and stator vanes due to short cord lengths. In this first approach, no variations in vane shape and angle of attack was considered because of the shortness of the vanes, The test results were, by themselves, not encouraging but led to most important insights to encourage further studies.

Improved axial pump (Phase III)

In Spring 1996, a new team following the recommendations outlined by the previous group, (see Ref. 2), re-designed the two-stage axial pump with a reduced number of rotor (5) and stator (6) vanes. This resulted in larger chord lengths of the vanes. According to Euler's theory for uniform flow conditions, the entrance (axial entry) and exit vector triangles for the rotor tip were chosen for $D = 0.05\,m$, $N = 2600$ (RPM), and an assumed hydraulic efficiency $\eta_h = 0.5$; requiring a rotor tip velocity of $U_{,o} = 6.807$ m/sec (see Fig. 3a).

The design utilized the results of Weinig's lattice theory for thin, circular arcs, [5] to evaluate the overcamber due to finite spacing of the vanes (Programs 'WEINIG.BAS' and 'WEINO.BAS'

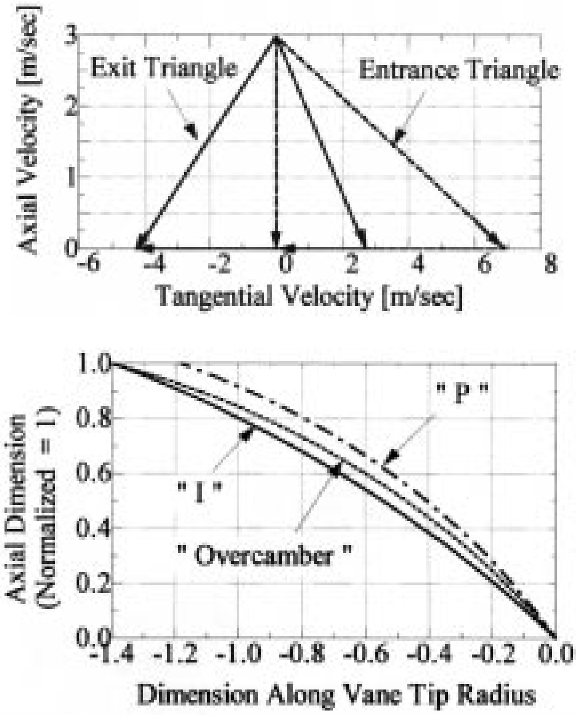

Fig. 3.

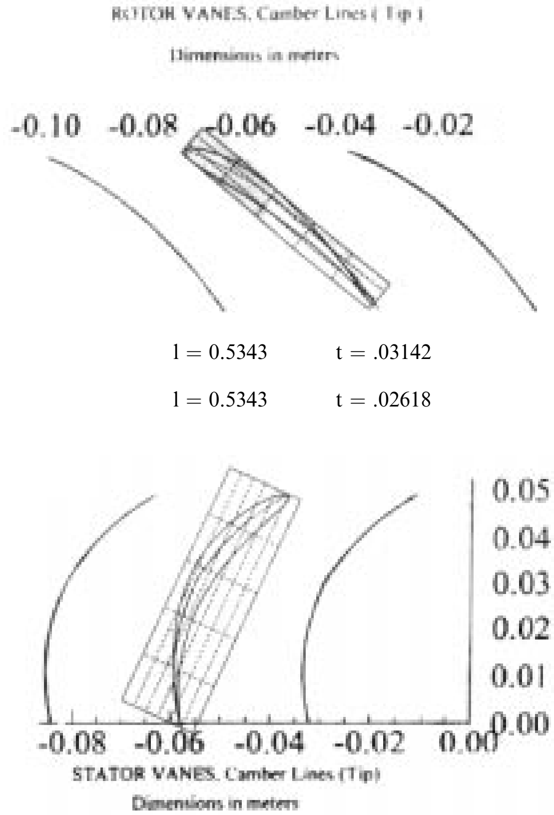

Fig. 4.

The BASIC programs are available in PC format from the author) (Fig. 3b). Note that the data for the Vane Tips presented in Figs 3, 4 and 5 are taken from [3].

Calculations for the thin vane profiles were on the basis of the thin airfoil theory, but supplemented by the Riegels Factor. Thus, not only did students determine the lattice arrangement for the profile camber lines and the shapes of the vanes, but they also obtained velocity and pressure distributions over the profiles. Here for a camber ratio $H = 0.08045$, thickness ratio $B = 0.1$ and an angle of attack $= 4.2$ degrees [4] (Program 'SHAPEP.BAS') (Figs 4a & 5). The students followed the same procedures to generate camber lines and vane profiles for the stator (Fig. 4b).

Factors such as:

1. Calculations carried out for laminar,incompressible boundary layers using the approximate method due to V. Karman and K. Pohlhausen [6], yielding information on the 'point of instability'.
2. Transition delay due to free stream turbulence (Program 'LBL12F.BAS').

confirmed that, for these longer vanes, transition to turbulence can be expected before the occurrence of laminar separation on the suction side of the rotor profiles (see Fig. 5). This promised improved pump performance. Indeed, hydraulic efficiency increased to the level established for the single stage radial pump (Airtex CDT).

Further Developments (Phase IV)

The Summer 1996 team, trying to improve the efficiency of the 2-stage pump without changing the overall dimensions, gave special attention to 'twisting' the vanes on both the rotors and the stators. While the intention was to accommodate an improved potential flow solution, the persistent effects of viscous layers along the hub and outside wall overshadowed this intent, cancelling out any improvement in efficiency. It was concluded that a moderate increase in dimensions (linear scale factor of 1.25) could alleviate to some extent the detrimental influences of viscous effects, while consideration of a decreased hub ratio $\nu < .8$ should also be pursued. Justification exists for further efforts in this direction.

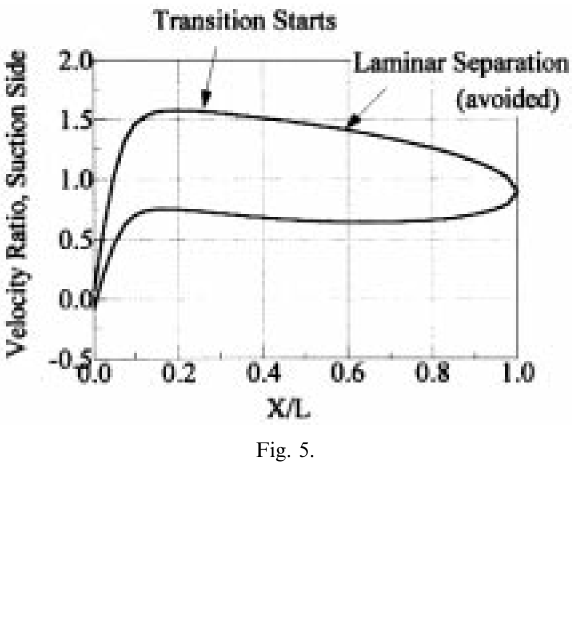

Fig. 5.

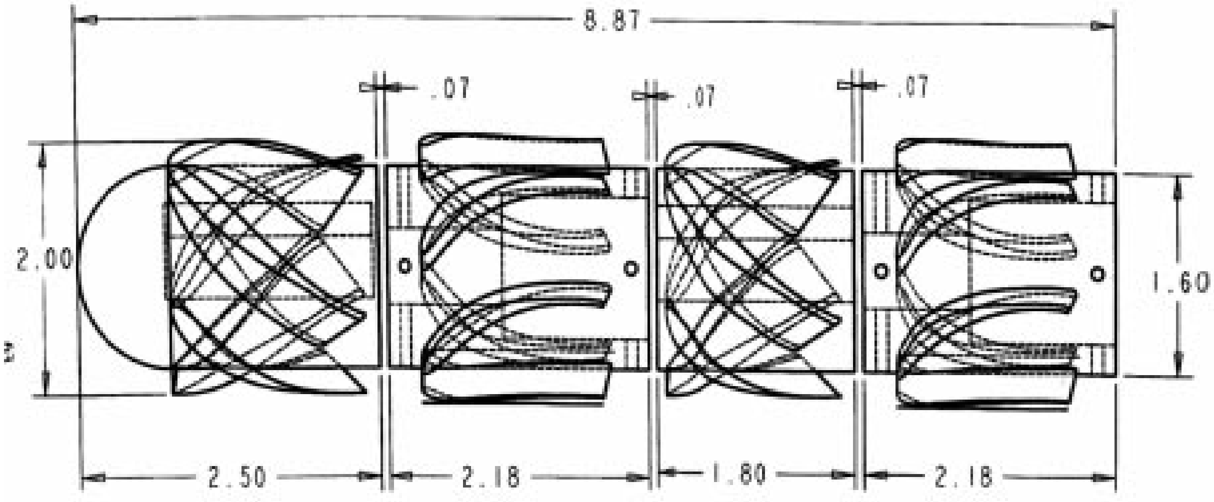

Fig. 6. Assembly drawing.

PROTOTYPE MODELING AND FABRICATION

Rotor and stator fabrication

Stereolithography, a rapid prototyping procedure, enabled fast production of the rotors and stators and these parts proved to be quite robust in the testing facility. The rotors and stators were built using a plastic resin, Exactomer 2202 SF, with a tolerance of about 0.01″. Manufacture of each part required approximately 4 to 5 hours.

Solid models for each part were first created on PRO/Engineer 3-D Modeling software, which provides a direct interface with the University's SLA-190 stereolithography machine. The PRO/Engineer program was used to transform the two-dimensional vane shaped profiles mentioned above into a cylindrical system of coordinates.It became possible to thicken the trailing edges of the vane profiles after consideration of them as being submerged into the boundary layer and its separation wake.

The pump assembly was contained in a 2″sch40 PVC housing, with the components mounted on a $\frac{1}{2}$″ shaft that extended out through a spring loaded seal from a standard automotive cooling pump. A 1/8″ Woodruff key secured each rotor on the shaft, while the pump housing held the stator stationary relative to shaft and rotors. Figure 6 shows the assembly drawing (the second stator was optional) and Fig. 7 is a cut-away photo of the pump built by the Spring 1996 team.

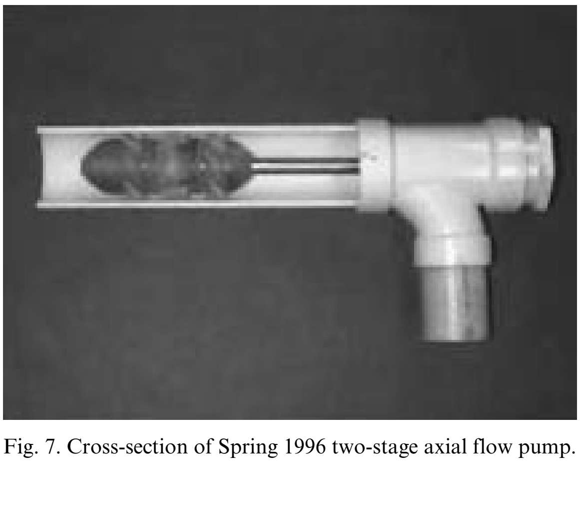
Fig. 7. Cross-section of Spring 1996 two-stage axial flow pump.

PUMP TESTING FACILITY

A closed loop arrangement allowed steady state testing of the pump. Figure 8 provides a schematic of the test stand. A 3 HP variable speed electric motor, controlled by a frequency converter, drives the pump. Power delivered to the pump is measured directly by the torque meter. Ambient temperature water circulates through the system, and a standard VDI/ASME nozzle measures the flow rate by utilizing a special computer program to cover an extended range of Reynolds Numbers. Pressure taps connected to a mercury manometer monitor static pressures at the pump entrance and at the exit. The throttling valve regulates the system resistance of the loop, allowing testing of a wide variation of flow rates from closed to open conditions.

Test procedure

Measurement of the hydraulic output of the pump, (flowrate Q, pressure rise ΔP), and the mechanical energy input, (torque and shaft speed), determined the pump performance characteristics. Performance curves (ΔP as a function of Q) for constant shaft speeds N (RPM) were obtained by varying the throttle valve setting. In preliminary tests we attempted to map the energy losses due to friction in the pump seal by running the pump without water, thus allowing us to extract both mechanical and hydraulic efficiencies.

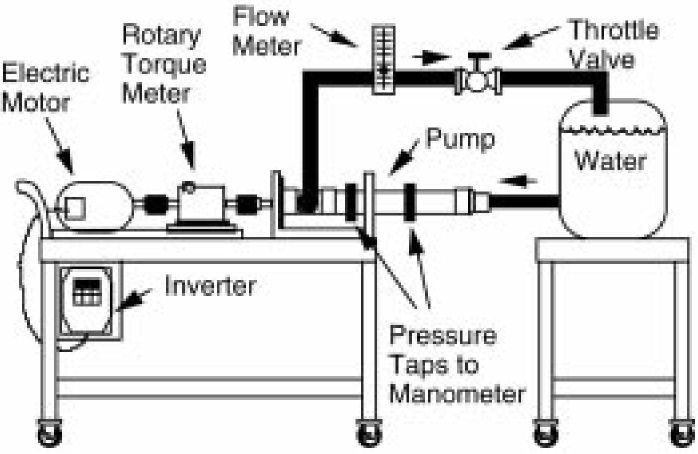

Fig. 8. UIUC M&IE Dept. pump testing axial flow pump facility.

Test results

Figure 9 shows the results obtained by testing the pumps designed by the teams in Spring 1995 and Spring 1996 in dimensionless form. It clearly shows, that the use of conventional geometries by the Spring 1995 team was not successful. By contrast, recognition of the significant effects of boundary layer transition for small scale pumps led to increased chord lengths, fewer vanes and notable improvements in the Spring 1996 design. Note also, that the set of dimensionless coordinates

$$\phi = \frac{4Q}{\pi D^2(1-\nu^2)U_{,o}}$$

and

$$\psi = \frac{2\Delta P}{\rho U_{,o}^2}$$

is more widely accepted for the analysis of axial pump performance. Also, the alternate form of the specific speed

$$\sigma = \phi^{1/2}\sqrt{1-\nu^2}\psi^{-3/4} = 0.3355\Omega$$

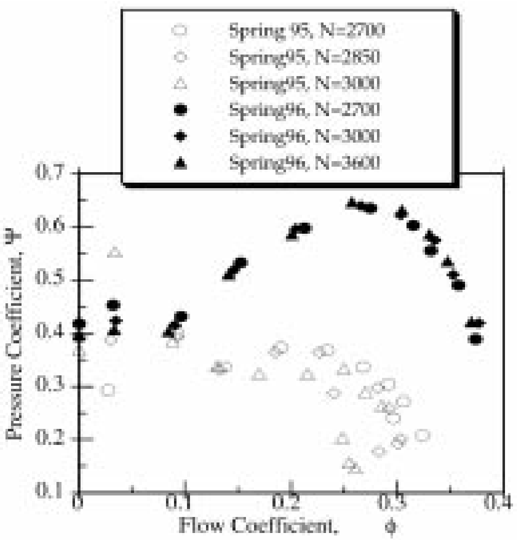

Fig. 9. Spring 1995/Spring 1996 pump.

already shown as second axes in Figs 1 and 2, is preferred in identifying axial pump design criteria. Note that ΔP represents here the pressure rise across the 2 stages of the pump.

Mapping the design point performances into the Cordier Curve can provide an interesting illustration of the different design forms, as considered in the present tasks. The three points located to the right of the Cordier Curve in Fig. 10 indicate the need for larger scale geometries (interpreted by Δ, or Ω values), or higher speeds (W, or RPM). For practical purposes, the performance curves for the Spring 96 model are presented in Fig. 11 in the form of ΔP (psi) versus. Q (gpm), for various values of N (RPM). Figure 12 shows a comparison between the performance of the Airtex CDT single stage radial pump at 2000 RPM and the 2-stage axial pump, Spring 96 model at 3600 RPM. At the design point, both pumps have the same efficiency, but the performance curves reflect the typical difference between the two types of pumps.

Finally, Fig. 13 addresses the system requirements of automotive cooling by comparing the usual method of thermostat throttling with the pump speed control approach. As can be seen,

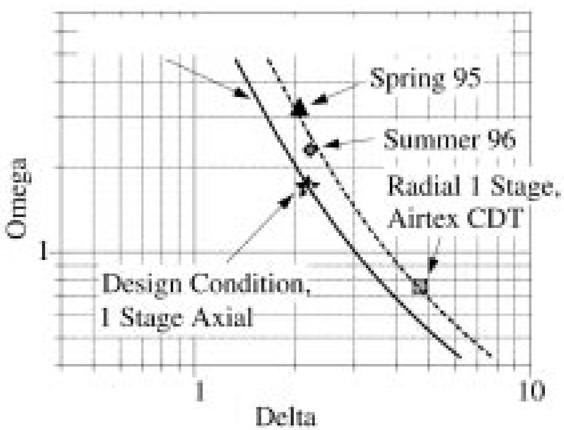

Fig. 10. Pump development, UIUC Senior performance comparison. Capstone design project. Performance presented in the Cordier diagram.

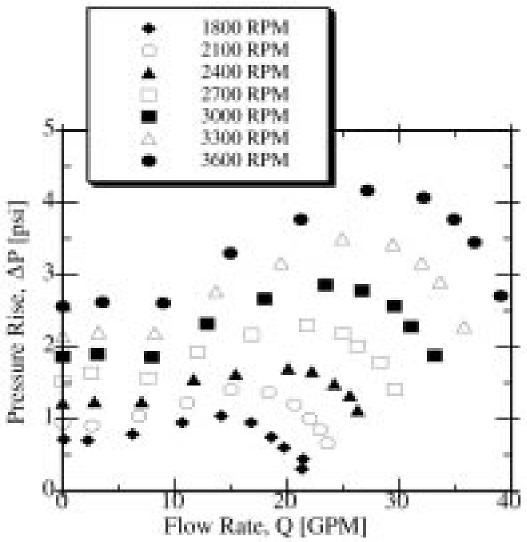

Fig. 11. Spring 1996 2-stage axial pump.

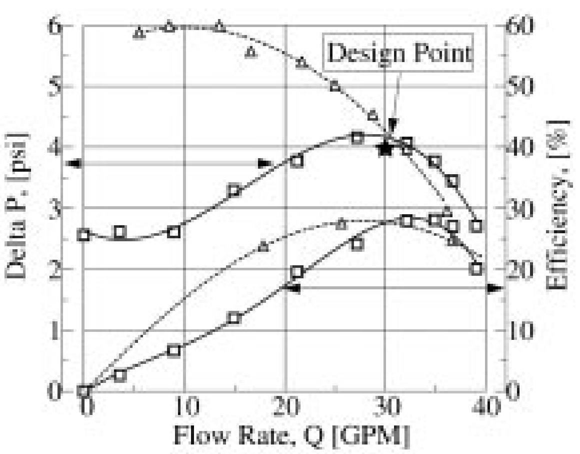

Fig. 12. Performance comparison of radial performance data. Single stage pump (Airtex CDT) N = 2000 RPM (dashed lines), and Spring 96 2-stage axialpump N = 3600 RPM (solid lines).

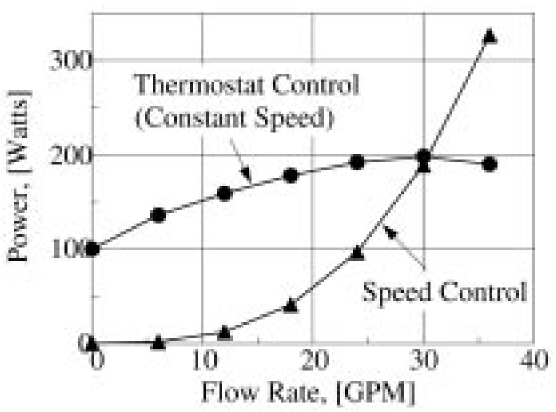

Fig. 13. Power consumption comparison of Airtex CDT pump, N=2000 RPM, thermostat control vs. Spring 96 2-stage axial pump with variable speed.

due to the dramatic reduction in power requirements for an automotive cooling pump using variable rotor speed control when compared to thermostat throttling, it is possible to make a strong case for speed control. In particular, a two stage axial pump that is driven by an electric motor with efficient speed control (preferably housed in the hub of the pump) provides for an attractive alternative due to its small dimensions, the elimination of shaft seals and the potential of further improvements in hydraulic efficiency.

CONCLUSIONS

Educational goals reached

All the Senior Capstone Projects provide for continuous interaction between the student teams, Senior Faculty Members, Industrial Sponsors and up-to-date Departmental support facilities. However, the specific effort as described in the text also stresses the continuity of product development in a sequence of 3 semesters [1, 2, 3]. Such an approach is typical for industrial research projects and, therefor, is of special value for students ready to make the transition from school to the real world. In particular, the use of (fluid machinery) analysis, design (Pro-Engineer), manufacturing (Stereo-Lithography) and product (prototype) testing required individual members of the student team(s) to concentrate on such special aspects of the project. Finally, a synthesis had to be cast into a final report for submission, (and for oral presentation using PowerPoint), to the scrutiny of both University Faculty and the Technical Management of the Industrial Sponsor.

Pump design and performance

As the point of departure for the first group [1], they were able to use the well-established empirical relationship for identifying operating ranges for turbomachinery of high efficiency (Cordier Diagram). This led to the conclusion that a two-stage axial pump could be a marginally promising candidate to replace a currently used single stage radial pump for automotive cooling. The design of the first prototype utilized well-defined design parameters for axial pump stages of high efficiency (and high Reynolds Number—see Fig. 2), but the test results turned out to be unacceptable.It was easy to conclude that the adverse pressure gradients and low Reynolds Numbers encountered by the vanes (high number and short chord lengths) was one reason for poor performance.

As a remedy, the second group decided to reduce the vane number as boundary layer calculations indicated that this would allow transition from laminar to turbulent flow on the suction side of the rotor vanes before laminar boundary separation and stall. Indeed, the performance of the re-designed two-stage axial pump [2] became immediately competitive to a radial pump. Yet, the resulting increase in overall pump length resulted in increased shear layer buildup along the hub and the housing, noticeably affecting the efficiency of the second stage. Twisting of the vanes near the hub did not improve the efficiency [3].

The overall conclusion for the feasibility of a two- stage axial pump replacing the radial type appears to hinge on two key issues:

1. design changes (reducing the hub diameter ratio)
2. the possibility to accommodate electric motors in the hub, (especially in the expectation of highly increased voltage levels in projected automotive systems).

Acknowledgments—This project represents the collective efforts of numerous individuals within the Department of Mechanical and Industrial Engineering and the corporate sponsor Airtex Products, Inc. of Fairfield, Illinois. Starting with the first design team in the Spring 1993 semester, a total of seven teams involving 22 undergraduate students working under the direction of the senior author have contributed to the continued evolutionary development of pump, housing and control design. The results achieved would not have been possible without the continuous coordination of Mr. Phil Hall, Manager, Design and Development, Airtex Products and his staff. In addition, the availability of support resources within the M&IE Department was instrumental:

- Bob Coverdill, Senior Research Engineer, who supervised the construction of the test stand and instrumentation selection
- M&IE Shop personnel, who worked on the machining of pump components
- Prof. Michael Philpott who allowed access to his Product Realization Laboratory and advised teams on the design and manufacture of pump prototype components
- graduate assistant Ho-Joon Lee, who provided consulting assistance regarding use of solid modeling, file translation, and production of stereolithographic component models.

The contributions of these individuals ensured the success of this continuous and continuing development project.

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Helmut H. Korst is an Emeritus Professor of Mechanical Engineering at the University of Illinois at Urbana-Champaign. He is a Fellow of ASME and AIAA, recipient of the ASEE Centennial Medallion. He received the Daniel Guggenheim Medal 'for his pioneering contributions in research in high-speed turbulent flows and for his life- long inspiring technical leadership to students and colleagues'.

Todd M. Leicht completed performance evaluations of the prototype as an Undergraduate Senior, and he continued his involvement with the project as a Graduate Assistant to subsequent teams. He graduated from UIUC in December 1996 with a BSME, and he is presently employed as a project engineer for Webcor Builders in the San Francisco Bay Area.