Open Access Journal

Journal of Power Technologies 97 (2) (2017) 85–88

journal homepage:papers.itc.pw.edu.pl

Modification of cooling impeller pump in combustion engines driven by electricity

Grzegorz Peczkis*

Politechnika Śląska, ul. Konarskiego 18, 44-100 Gliwice, Poland

Abstract

This paper presents methods of using an electric powered impeller pump to circulate liquid coolant in a combustion engine. The proposed impeller pump is driven differently from pumps typically used in combustion engine cooling systems. The conventional mechanical drive using the mechanical energy produced by the combustion engine is replaced by an electric motor. This solution is not new, but is being given greater consideration due to the improved power efficiency of the combustion engine. Power-smart optimum methods of impeller pump regulation are discussed. The guidelines based on the theory of the impeller pump construction included herein seek to make combustion engine designers more sensitive to issues related to the design of highly efficient flow systems of impeller pumps. The cooling system pumps which are currently used in combustion engines often draw on rather unsophisticated structural solutions which are no longer used in any other industry due to their low efficiency. This is related to the relatively low power consumption of the impeller pump relative to the power output of the entire engine, as well as to its low cost of manufacture.

Keywords: pump, impeller pump, cooling system, electric motor pump

1. Introduction

The use of mechanical power generated by the combustion engine to drive the cooling system pumps included in the engine structure is very common. The circulation system of the liquid coolant consists of ducts and pipelines that connect the combustion engine to the liquid-to-air exchanger (cooler). It is a closed system, where the liquid circulation is forced by the impeller pump. The pump driven by the combustion engine power begins operating the moment the engine is started. The pump is most often driven directly from the crankshaft by means of a gear, enabling the desired range of the operating pump rotational speed to be achieved. After the engine is started, the coolant is heated preliminarily within the engine itself, where it is contained by means of a thermostatic valve. During that time, when the liquid is pumped into a closed system, the effect of the pump operation is dissipated and the power consumed by the pump is irretrievably lost [1–6]. When the cooling liquid reaches the optimum temperature from the point of view of engine operation, the thermostat opens the system connecting the engine to the cooler and the liquid starts circulating between the two components. In the first stage, the cold liquid in the cooler gets mixed with the hot liquid from the engine. The difference in temperature depends on the temperature of the surroundings (the temperature of the liquid in the engine is about 100°C). After a short time of engine operation, the cooling liquid temperature stabilizes according to design assumptions of the given engine type (cooler size, load, size and arrangement of the cooling passages in the engine, cooling system capacity). If the coolant breaches the allowable temperature, a fan system activates, increasing the amount of air passing through the cooler. After the temperature falls, the fans are switched off. A further decrease in coolant temperature causing the thermostatic valve to close does not occur until the engine turns off and cools down. If ambient temperatures are low (below -15°C), (typically electric) heaters are used in the cooling system, making it possible to keep the cooling liquid temperature at a desired level [7–9].

If the combustion engine operates under a steady, constant load (stationary engines), the cooling pump, if appropriately built, driven directly from the combustion engine, performs its task optimally and at the same time – at minimal risk of failure.

If the combustion engine operates at variable load, the temperature of the liquid pumped in the cooling system varies. A change in the engine shaft rotational speed triggers a change in the rotational speed of the pump impeller and, consequently, the pumping parameters [3, 6, 10–12]. Adjustment of the pumping parameters varied by the changeable rotational speed is desirable for reasons of pump efficiency.

*Corresponding author

Email address: grzegorz.peczkis@polsl.pl (Grzegorz Peczkis)

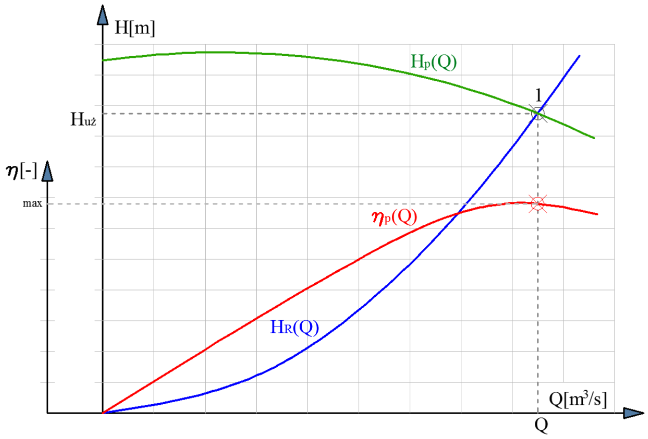

Figure 1: Curves illustrating the pump co-operation with the cooling system

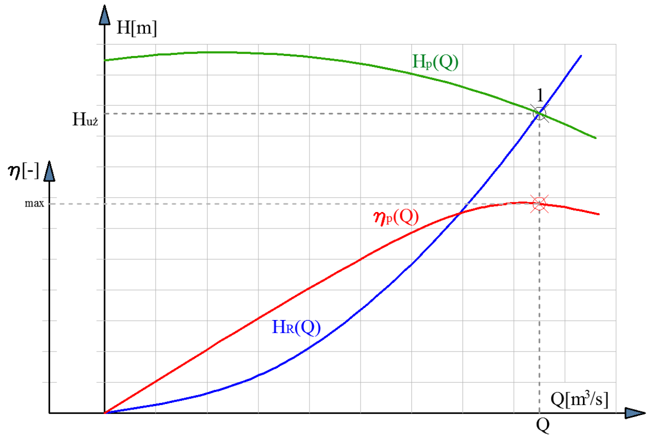

Figure 2: Curves illustrating the pump co-operation with the cooling system

However, a sudden reduction in engine load and rotational speed and, consequently, a drop in pumping parameters do not involve an equally sudden reduction in the demand for carrying away thermal energy. Operation of the cooling system in conditions of high ambient temperatures and low rotational speed of the engine is equally unfavorable. Low pumping parameters make it impossible to reduce the temperature of the coolant effectively. In both these cases the cooling system can be extended to raise the pump parameters. This, however, increases energy consumption of the pump.

2. Impeller pump co-operation with the cooling system

The basic parameters of the impeller pump operation are: the pump flow rate (volumetric discharge rate marked as Q) and the elevation head (marked as H). They occur as couples, making individual points on the pump characteristic curve ($H_p(Q)$ shown in Fig. 1). One impeller pump with a constant shape of the vanes for one rotational speed has one characteristic curve presented in the chart. Each elevation head curve is accompanied by the pump efficiency curve ($\eta_{p(Q)}$ shown in Fig. 1). Another curve illustrating the cooling system (with an open thermostatic valve) $H_p(Q)$ is also shown in the chart. Its parabolic shape depends on the hydraulic perfection of the cooling system (minimized flow resistances). The flattest curve of the cooling system flow resistance curve is desired – the demand for energy needed to pump the same amount of the coolant is then at its lowest. The point (marked as “1” in Fig. 1), where the curve illustrating the cooling system losses intersects with the characteristic curve of the pump operating in this system is called the operation point. If the pump then reaches its highest efficiency, the point is also the optimum point.

A change in rotational speed shifts the curve as presented in the chart (Fig. 2). As the rotational speed increases, the pump parameters rise (the pump flow rate and the elevation head are shifted from Point 1 to Point 2). The rise proceeds along the curve illustrating the cooling system flow resistances. A reduction in the rotational speed involves a drop in the pumping parameters from Point 1 along the resistance curve to Point 3 and further – to Point 4. The impeller pump enjoys maximum efficiency, i.e., the lowest expenditure of energy needed to pump identical amounts of the cooling liquid, only at one point of the pump operation for the entire set of all characteristic curves. From the point of view of its efficiency, the pump should operate at this particular point all the time.

Change j in the rotational speed in the range dependent on a given pump structure decreases pump efficiency. But it is this particular way of changing the parameters of an individual pump operation by adjusting its rotational speed that is the optimum one in a range higher than several per cent, i.e., in the range of values typical of combustion engines with a variable load.

If the rotational speed is reduced, the efficiency curve is shifted to the left. If the rotational speed is reduced from the optimum value (at which pump efficiency was highest), the maximum of the efficiency curve is reduced. If the rotational speed is increased above the optimum point, the efficiency curve is shifted to the right and the curve maximum is also smaller.

The equations that make it possible to describe the changeability of pumping parameters (the pump elevation head and flow rate) at changes in the rotational speed result from the flow similarity theory. The flow rate values change linearly with change in rotational speed.

$$\frac{Q}{Q_1} = \frac{n}{n_1} \tag{1}$$

The elevation head is square-dependent on the rotational speed.

$$\frac{H}{H_1} = \left(\frac{n}{n_1}\right)^2 \tag{2}$$

Calculations can be performed according to the dependencies mentioned above at low, several per cent changes in efficiency.

Based on similarity theory, regardless of the magnitude of the pump's individual parameters, the shape numbers char-

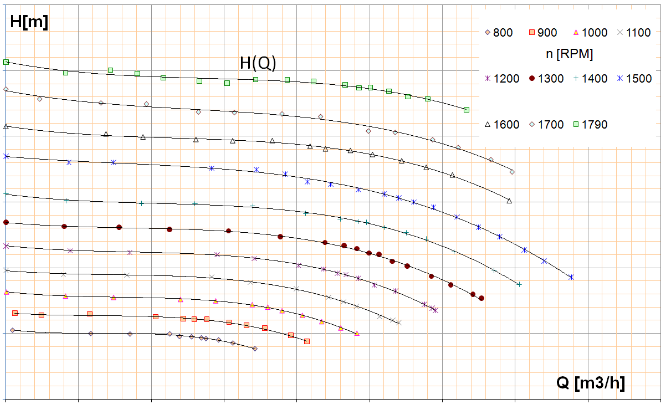

Figure 3: Curves illustrating the pump measured useful elevation head for variable rotational speed

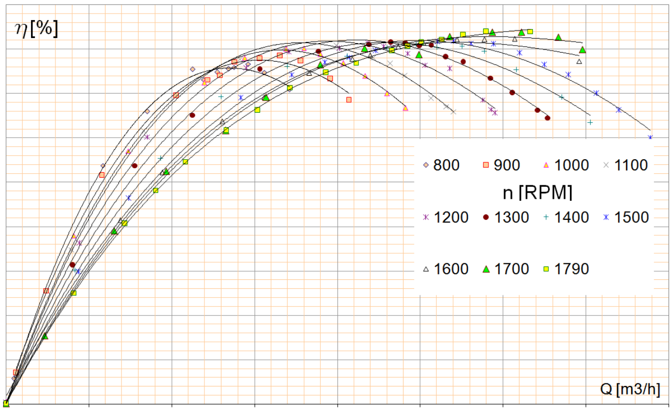

Figure 4: Curves illustrating pump efficiency calculated based on measurements for variable rotational speed

acterizing the pump impeller are defined. In this manner the impeller type is determined in terms of the way in which the liquid flows through it: radial flow, mixed-flow, helicoidal flow or axial flow impeller. Tables can be found in the reference literature that allow a quick identification of the shape of the rotodynamic pump impellers based on their specific speed. The specific speed can also be calculated from the following formula:

$$n_Q = n\frac{\sqrt{Q}}{\sqrt[4]{H_{u\dot{z}}^3}} \quad (3)$$

where: n – the shaft rotational speed [RPM], Q - volumetric flow rate [m^3/h]. $H_{u\dot{z}}$-head [m]

The chart (Fig 3) above presents the measured characteristic curves of an impeller pump with a changeable rotational speed. It can be seen that, although the rotational speed is halved, the flow rate hill drops slightly. The subsequent drop, not presented in the chart, is already more progressive. For economic reasons, if impeller pumps are used on a continuous basis, their operation at an efficiency level that is 20% lower than the highest rated, efficiency is not taken into consideration. After a few months, the total cost incurred due to continuous operation of a pump with a substantially diminished efficiency is often higher than the cost of purchase of a new pump with appropriate parameters [5, 6, 12–14].

The other well-known methods of infinite adjustment of pumping parameters: throttle regulation and shunt regulation are less effective or, as in the case of regulation through a change in vane angle, are not commonly used in impeller pumps due to the structural complexity and related costs.

The pump under analysis, for which characteristic curves are presented in this paper, had the specific speed $n_Q = 30$. The impeller pump should be operated at a slight change in the flow rate and elevation head parameters. In the case of regulation through a change in rotational speed, the parameters that still deliver high pump efficiency can be varied across a wide range.

The impeller pump under analysis, tested in the laboratory, was driven by an electric motor with a variable rotational

Figure 5: Two impellers with a different structure but similar pumping parameters

speed. The systems of monitoring and adjusting the rotational speed of direct- and alternating-current motors make use of them possible across the entire range of powers under discussion. A potential problem is the additional cost related to their installation.

The option of not activating the cooling cycle pump when firing up the combustion engine means zero energy consumption by the pump. The pump will activate only when it is necessary to pump the coolant in the cooling system. A cooling system with an isolated impeller pump driven by an electric motor with variable rotational speed means pumping parameters can be raised exactly when they are needed to retain the thermal balance of the combustion engine, at only a slight loss of operating pump efficiency. The advantage of such a system is the possibility of controlling its flow rate depending on actual momentary needs. The downside is the need to incorporate into the combustion engine system an additional electric motor fitted with a rotational speed adjustment module to drive the cooling system pump.

Fig. 5 shows two impellers with a different structure but similar coolant pumping parameters. Both impellers operate at a rotational speed of up to 8000 rpm. The one on the left is referred to as a channel impeller; the one on the right is a classic radial impeller. The two impellers were made using rapid prototyping technology (3D printing), which means the optimum structure can be obtained in an effective and relatively cheap way. Owing to an increase in pump rotational speed, it is possible to achieve the same pumping parameters with smaller pump dimensions. The downside of increasing rotational speed is the occurrence of cavitation in the flow system. This phenomenon grows with speed. A common design error of the cooling cycle pump systems in use is the very advanced structural simplification of the flow system (impellers, inflow systems on the sucking side and outflow systems on the pumping side). The impellers have straight vanes with excessive thickness and with dull ends at the inlet (on the sucking side). The inflow and outflow systems are made under the engine structure regime without regard for the liquid flow. This results in considerable pressure losses, which translates into the need to raise the pump parameters and increase its power consumption. If pumps are made as units isolated from the engine system, it is possible to improve their hydraulic structure in compliance with the principles of pump building.

Another important issue is the impeller pump sealing. In particular – the dynamic seal between the stationary casing and the rotating shaft on which the impeller is mounted. It is defects in or wear of this seal that most often causes damage to pumps that start leaking. In classic systems of radial impellers (like in Fig. 5) mechanical contact seals are used. In them, rings with a very smooth surface are pressed against each other with a spring, which prevents leaks of coolant. However, this solution requires that the pairs of ring materials that operate in partial face contact to each other should be hard or soft. Different material pairs are used. Graphites are soft materials; ceramics, alloy sinters or metals subjected to surface hardening are hard materials. A damage to the seal results from one of the interacting sealing rings (the soft one made of graphite) becoming worn out due to overly strong pressure from the spring. In the case of sudden failures, the cause is most often the use of ceramics in the production of slip-rings. High hardness is then achieved and the surfaces are very smooth, but the materials are brittle and sensitive to mechanical impact. If they crack, the sealing cannot operate appropriately. Sintered carbides are more resistant to impact, but they are more expensive. The problem could be solved by using double-suction impeller systems or a hermetic separation of the flow systems from the environment. The application of a double-suction impeller involves substantially smaller requirements concerning its sealing during pump operation. The downside of pumps with a double-suction impeller is the more complex structure of the casings (usually divided in the pump axis). Hermetic systems, e.g. driving impellers levitated by electromagnetic couplings, are not commonly used because such structures involve higher costs. The use of drives operating partially in the pumped liquid (wet drives) causes a drop in drive efficiency.

3. Summary/Conclusions

This paper concerns methods of pumping liquid into combustion engine cooling system by means of an impeller pump driven by an electric motor. The main issue is to provide proper regulation, enabling to cool the engine at maximum pump efficiency. The problem is that a sudden reduction in engine load, rotational speed and consequently pumping parameters, does not involve an equally sudden reduction in the demand for carrying away thermal energy. As the pump achieves maximum efficiency at only one specific set of conditions, adjustment of the pumping parameters varied by the changeable rotational speed is desirable. In presented paper an electric motor drive is proposed for the pump, as the possibility of not activating the cooling cycle pump when starting the combustion engine means energy savings. The pump may be activated only when required.

Acknowledgments

References

[1] R. Cipollone, D. Di Battista, A. Gualtieri, A novel engine cooling system with two circuits operating at different temperatures, Energy Conversion and Management 75 (2013) 581–592.

[2] Y. H. Shin, S. C. Kim, M. S. Kim, Use of electromagnetic clutch water pumps in vehicle engine cooling systems to reduce fuel consumption, Energy 57 (2013) 624–631.

[3] E. G. Ribeiro, A. P. de Andrade Filho, J. L. de Carvalho Meira, Electric water pump for engine cooling, Tech. rep., SAE Technical Paper (2007).

[4] A. Poullikkas, Optimization analysis for pumped energy storage systems in small isolated power systems, Journal of Power Technologies 93 (2) (2013) 78.

[5] J. Dobriański, M. Wesołowski, Ocena techniczno-ekonomiczna zastosowania samoczynnego obiegu cyrkulacyjnego w słonecznej instalacji grzewczej, Problemy inżynierii rolniczej 11 (3) (2003) 71–78.

[6] T. Hu, J. Zhu, W. Zhang, Experimental investigation on system with combination of ground-source heat pump and solar collector, Transactions of Tianjin University 19 (3) (2013) 157–167.

[7] E. Cortona, C. H. Onder, Engine thermal management with electric cooling pump, Tech. rep., SAE Technical Paper (2000).

[8] C. Wu, L. Chen, F. Sun, S. Cao, Optimal collector temperature for solar-driven heat pumps, Energy conversion and management 39 (1) (1998) 143–147.

[9] A. Salij, M. Poniewski, J. C. Stepien, Operation of pumps in a district heating system supplying a distant major industrial user, Journal of Power Technologies 95 (5) (2015) 68.

[10] B. Jawad, K. Zellner, C. Riedel, Small engine cooling and the electric water pump, Tech. rep., SAE Technical Paper (2004).

[11] Foit H. Zastosowanie odnawialnych źródeł ciepła w ogrzewnictwie i wentylacji. Wydawnictwo Politechniki Śląskiej. Gliwice, rok 2013.

[12] E. Różycka, Analiza opłacalności zastosowania niekonwencjonalnych źródeł energii w projektowanym budynku jednorodzinnym. kolektory słoneczne, pompy ciepła, Rocznik Ochrona Środowiska (Tom 11) (2009) 1351–1371.

[13] P. Omojaro, C. Breitkopf, Direct expansion solar assisted heat pumps: A review of applications and recent research, Renewable and Sustainable Energy Reviews 22 (2013) 33–45.

[14] J. Dobriański, J. Fieducik, Urządzenie zastępujące pompę cyrkulacyjną w instalacji słonecznej, Zeszyty Naukowe Politechniki Rzeszowskiej. Budownictwo i Inżynieria Środowiska (2006) 105–112.